

18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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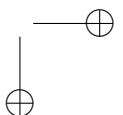
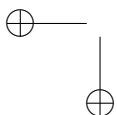
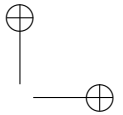
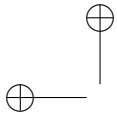
**PROCEEDINGS OF THE
18TH INTERNATIONAL
SHIP AND OFFSHORE STRUCTURES CONGRESS**

Volume 2

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PROCEEDINGS OF THE 18TH INTERNATIONAL SHIP AND OFFSHORE STRUCTURES CONGRESS

Volume 2

Edited by

Wolfgang Fricke

and

Robert Bronsart



Universität
Rostock



Schiffbautechnische
Gesellschaft e. V.

Schiffahrts-Verlag "Hansa" GmbH & Co. KG, Hamburg

Schiffbautechnische Gesellschaft e.V.
Bramfelder Str. 164
22305 Hamburg, Germany

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First published in 2012

ISBN 978-3-87700-132-5
2012 Schifffahrtsverlag "Hansa" GmbH & Co. KG, Hamburg

Designed and set by Lutz Kleinsorge with L^AT_EX 2_ε
Produktionsmanagement: impress media GmbH, Mönchengladbach
Printed in Germany

PREFACE

This volume contains the eight Specialist Committee reports presented and discussed at the 18th International Ship and Offshore Structures Congress (ISSC 2012) in Rostock, Germany, 09-13 September 2012.

Volume 1 contains the reports of the eight Technical Committees whilst Volume 3 contains the report on the congress, the keynote lecture and the discussions of all the reports together with the replies by the committees.

The Standing Committee of the 18th International Ship and Offshore Structures Congress comprised:

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Secretary: Robert Bronsart

On behalf of the Standing Committee, we would like to thank Germanischer Lloyd, Det Norske Veritas, American Bureau of Shipping, Lloyd's Register, Nippon Kaiji Kyokai and Bureau Veritas for sponsoring ISSC 2012.

Wolfgang Fricke
Chairman

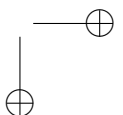
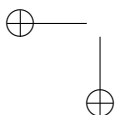
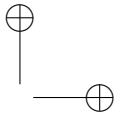
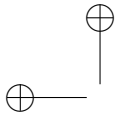
Robert Bronsart
Secretary

Rostock, September 2012

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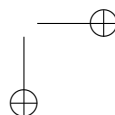
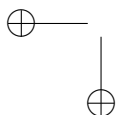
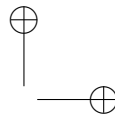
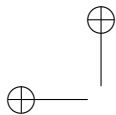
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18th INTERNATIONAL SHIP AND
OFFSHORE STRUCTURES CONGRESS
09-13 SEPTEMBER 2012
ROSTOCK, GERMANY
VOLUME 2



COMMITTEE V.1
**DAMAGE ASSESSMENT FOLLOWING
ACCIDENTS**

COMMITTEE MANDATE

Concern for the structural integrity of offshore structures exposed to hazards. Assessment of risk associated with damage, range of repair required and the effects of temporary repairs and mitigating actions following the damage. The hazards to be considered include hydrocarbon explosions and fires, wave impact, water-in-deck, dropped objects, ship impacts, earthquakes, abnormal environmental actions and possible illegal activities like the use of explosives and projectiles.

COMMITTEE MEMBERS

Chairman: Jerzy Czujko
Natacha Buannic
Sören Ehlers
Chen-Far Hung
Frank Klæbo
Spiro J. Pahos
Mark Riley
Wenyong Tang
Alex Vredeveldt
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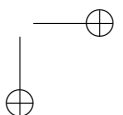
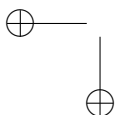
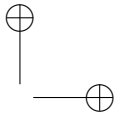
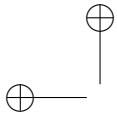
KEYWORDS

HC Explosions, HC Fires, underwater explosions, wave impact, wave-in-deck, dropped objects, ship impact on offshore structures, earthquakes, abnormal environmental actions, ice and icebergs, flooding, explosives and projectiles

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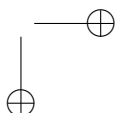
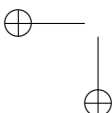
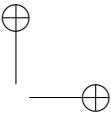
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1 INTRODUCTION

Structural integrity of offshore installations exposed to hazards is one of the challenges offshore industry has faced. Like all kinds of installations, offshore structures can be damaged or collapse as a consequence of a number of possible incidents. For the first time in the history of ISSC activity, a committee has been established to perform a systematic review of offshore installations with respect to their exposure to accidents. The following will be considered:

- The assessment of risk associated with damage,
- The range of repair required,
- The effects of temporary repairs and mitigating actions following the damage.

Therefore, the aims of the committee are to:

- Assess the loads acting on structures during various types of accidents,
- Investigate the consequences of those loads,
- Suggest practical implementation of these findings by establishing balance between structural design and structural safety.

The first step in the project is to identify types of accidents which might occur. Although when it comes to accidents not everything can be predicted, based on statistics and experience, the work is focused on a number of the most probable situations. As expected in the environment where oil and gas are processed, hydrocarbon explosions and fires constitute the most severe hazard for offshore installations. Other extreme conditions this report investigates are: underwater explosions, wave impact, water-in-deck, dropped objects, ship impacts, earthquakes, abnormal environmental actions, ice and icebergs, flooding, as well as illegal activities like the use of explosives and projectiles.

Having established possible threats, the committee's work is to review and recommend best practice in the offshore industry. Ensuring structural safety in the design process of offshore installations requires established procedures for the assessment of loads acting on these structures, as well as procedures for the assessment of the consequences of these loads. Therefore on this level of the project, the focus of the committee is on the following issues:

1. First, the safety measures to be taken during the design phase,
2. Secondly, in case of accidents, the assessment of the level of damage and of the structure's residual strength.

In order to fully investigate the issue of damage assessment following accidents on offshore installations, the committee review selected technical reports and papers worldwide, presenting state-of-the-art research and development achievements in the field.

As part of this Committee's work, a benchmark study has been carried out, aiming at the prediction of structural response of typical offshore installation components subjected to hydrocarbon explosions. The study is based on full scale experiments with hydrocarbon explosions. Its objective is to document how accurately the use of existing software and advanced structural modelling can predict behaviour of structures when subjected to this type of loads.

2 HAZARDS ON OFFSHORE FACILITIES

Quantifying risk in offshore facilities is a multifaceted task as different dynamic effects can arise from various hazards. As deeper oil fields are being discovered, complex

Table 1: Pertinent types of hazards for various offshore facilities

Type of structures	Types of Hazards											
	Explosions	Fires	Underwater Explosions	Wave Impact	Water in Deck	Dropped Objects	Ship Impact	Earthquake	Ice Iceberg	Flooding	Abnormal Environmental Loads	Illegal Activities
Fixed platform	x	x	x	x	x	x	x	x	x		x	x
Semi-Submersible	x	x	x	x		x	x		x	x	x	x
FPSO	x	x	x	x		x	x			x	x	x
TLP	x	x	x	x		x	x				x	x
Spar	x	x	x	x		x	x			x	x	x
Wind turbine, foundation			x			x	x	x	x		x	x

methods of analyses are required. Table 1 is a brief description of a broad range of hazards in offshore facilities and it is by all means not exhaustive. Therefore, engineering judgement must be applied in the design process to select credible hazards.

Identifying the hazards in a tabular format as shown in Table 1 is barely the first step. Predicting the failure modes and any coupled effects likely to arise from secondary or tertiary effects is a challenging task (dropped object > explosion > fire > loss of structural integrity > flooding). Ultimately, any identification will help in preventing and/or mitigating each hazard separately. Feedback from historical data can be of tremendous value during a Hazard Identification brainstorming session (HAZID) when analysing event frequencies and consequences. It is during such sessions that deviations from normal operations, unlikely events, and human factors come into play. Hazards related to offshore facilities are identified by a team of experts and users, who also work jointly on risk assessment, risk reduction and emergency preparedness. These three tasks are separate entities where in most cases the implementation of a risk control option is to be decreed by regulations.

Existing technology can couple numerous scenarios together in a multi-physics analysis where thermal, impulsive, ultimate limit state and hydrodynamic analyses are linked in one common system with the capability of parametric design. Studying various scenarios with advanced techniques can help someone to understand the consequences after the initiating event, and engineer a system against target safety levels.

3 HYDROCARBON EXPLOSIONS

Hydrocarbon explosions and fires are extremely hazardous in offshore installations, Czujko (2010). They involve extreme explosion actions and heat, which have serious consequences for health, safety and the surrounding environment. Since the Piper Alpha accident, a substantial amount of effort has been directed towards the management of explosions and fires in offshore installations. The event scenarios leading to major accidents are generally unpredictable as the calculated frequencies of such accidents are often very low. The consequences of such accidents are however directly related to the inventories of flammable or toxic substances present. To prevent escalation, effective barriers should be in place for the most likely events and a good technical

standard is required for safe operation. Risk-based approaches, rather than traditional prescriptive approaches, have begun to be more extensively applied in offshore designs.

3.1 Explosion Load Assessment

There are no simple calculation methods for determining blast loads for offshore structures. A number of predictive approaches are currently being applied to generate blast overpressure from explosions in congested volumes. However, it is the Computer Fluid Dynamics (CFD) models that are most frequently used for practical offshore development projects. These models solve the underlying equations describing gas flow, turbulence and combustion process topological in precise representation of offshore topsides. Explosion simulations using CFD models have the potential for providing high predictive accuracy and a greater potential of addressing any complex blast scenario.

Other models, such as empirical and phenomenological models, are reviewed and compared in Czujko (2001) and Czujko (2010).

Paik and Czujko (2011b) give state-of-the-art review of technologies used in assessing the risk of hydrocarbon explosions and fires in offshore installations. Both qualitative and quantitative risk assessment approaches are described, and the modelling techniques employed in the quantitative assessment of explosions and fires are presented.

Application of CFD models for calculation of explosion and fire action for FPSO topsides is presented by Paik *et al.* (2011). The existing test data on a methane gas explosion and propane gas jet fire was reanalysed using the ANSYS CFX code. It was concluded that the CFD simulations proposed in this study were in good agreement with the experimental results.

3.2 Load Definition for Design

3.2.1 General

Explosion generates different type of loads depending on the size and shape of structures and equipment. The following types of explosion loads have to be considered in design:

1. Explosion overpressure, p_o , dynamic load generated on large surfaces.
2. Drag force, p_d , dynamic load generated on small equipment items and piping.
3. Differential pressure, p_{diff} , global dynamic load generated on large equipment items or enclosures located within the explosion area by explosion wave passing the object.

Overpressure loads result from increases in pressure due to expanding combustion process. Description of time dependent overpressure and drag pressure is given in Figure 1 and Figure 2.

Drag is a vector quantity in contrast to the overpressure which is scalar, i.e. drag has three independent components. Drag, which is proportional to square of flow velocity, is a significant load for long and slender objects if flow speed in the plane normal to object's length is high. For this reason drag is always measured in a plane, not in a direction, which will be referred to as plane drag. For example drag load in plane XY is significant for objects (pipes) spanning in Z direction etc. (Figure 2).

3.2.2 Overpressure

Module walls, blast and fire walls and decks should be designed to resist explosion overpressures.

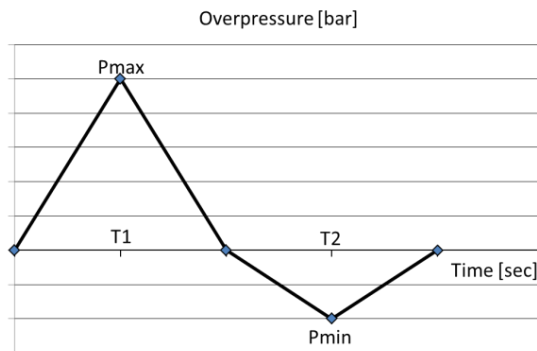


Figure 1: Parameters defining design overpressure and drag pressure

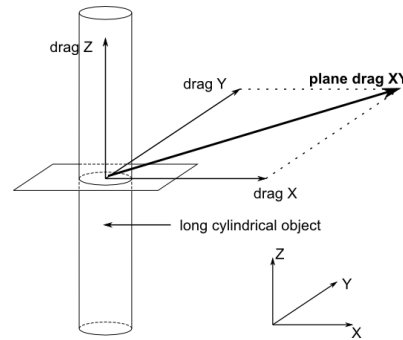


Figure 2: Directions of design drag pressure

For static analysis of walls and decks structure proper Dynamic Amplification Factors (DAF) should be accounted for and applied to increase values of overpressures. When using non-linear dynamic FE method, overpressures can directly be used in the analysis process.

3.2.3 Drag Loads

Drag load is a directional loading due to the passing air/gas flow. Gas explosions can generate both high overpressure and high-speed gas flows as a result of gas combustion process.

According to UKOOA (2003) drag loads dominate for obstacles with dimensions smaller than 0.3 m or on cylindrical obstacles smaller than 0.3 m in diameter, in particular in regions of high gas velocity near vents. Both drag and pressure difference loads are significant on objects between 0.3 m and 2 m in the flow direction. Drag loads are particularly important in open areas such as on the deck structures of an FPSO. The gas clouds associated with explosions on FPSOs may be very large and gas velocities up to 500 m/s could be experienced. The direction of gas flow may also be very variable, for example in the case of the pipe rack of an FPSO acted on by an explosion ignited at low level. Secondary projectiles may be a problem for FPSOs in view of the higher gas velocities.

Drag forces can be represented as, Czujko (2001):

$$F = F_d + F_p$$

Where:

- F_d = Form drag contribution proportional to the area, density and velocity square, and depending on Reynolds number and function of Mach number (U/c), where U is velocity of expanding gas and c is speed of sound.
- F_p = Contribution from the differential pressure.

For small piping and equipment form drag is a dominant contribution in drag forces. Large equipment, as for example compressors, is mainly subjected to effects of differential pressure. Large items like scrubbers are subjected to both drag components. The principles in Table 2 should be used for the calculation of drag force.

Table 2: Limit for equipment size to calculate drag force contributions.

$D [m]$	F		
	F_d	$F_d + F_p$	F_p
$< 0,6$	×		
$0,6 < D < 2,0$		×	
$> 2,0$			×

3.2.4 Scenario Definition

For an explosion to occur a gas cloud with a concentration between the upper flammability limit (UFL) and lower flammability limit (LFL) must be ignited. The overpressure caused by the explosion will depend, amongst other things, on, API (2006):

1. The gas or gas mixture present
2. The cloud volume and concentration
3. Ignition source type and location
4. The confinement or venting surrounding the gas cloud
5. The congestion or obstacles within the cloud (size, shape, number, location)

Factors affecting the origin of accident events according to Norsok (2010):

- Storage (number and size of inventories)
- Equipment type
- Risers and wells
- Product type
- Ignition sources
- Type of operations
- Production operations
- Deck type
- Structure location

The problems of creating inhomogeneous clouds by dispersion simulations are commonly solved through establishment of equivalent stoichiometric clouds at time of ignition. This may, however, result in a too short duration of the load.

3.2.5 Probabilistic Explosion Risk Model

The explosion risk model considers each leak scenario individually. A leak scenario is described by a transient gas leak rate, gas properties, leak location and direction, and ventilation conditions (wind speed and direction for naturally ventilated areas). For a Total Risk Analysis (TRA) in compliance with NORSOK Z-013 (2010) it is common to apply 9 initial leak rates and 6 leak directions. In addition, 12 wind directions and a range of wind speeds shall be reflected. Normally, the leaks from each process unit are considered individually. This results in a tremendous number of leak scenarios, even if an analysis normally applies a reduced number of wind directions (for which dispersion conditions are similar). The frequencies for the specific leak scenarios are quantified as follows for each process unit:

$$f_{leak\ scenario} = f_{leak} \cdot P_{rate\ and\ fluid\ in\ category} \cdot P_{wind\ direction} \cdot P_{wind\ speed} \cdot P_{jet\ direction}$$

The time dependent leak rate depends primarily on the segment inventory and the blow down system capacity, but also time until the segment is isolated and blow down initiated. A frequency of occurrence is quantified for each leak scenario. The explosion risk is the sum of the explosion risks for each individual scenario.

The HSE Hydrocarbon Release Database (HCRD) is the best quality dataset that exists on offshore releases and has thus become the standard source of leak frequencies for offshore quantitative risk assessment (QRA). Statoil have observed that different solutions by different analysts lead to QRAs having significant inconsistency in leak frequencies despite being based on the same dataset. Statoil therefore initiated a study in 2008 to standardise the leak frequency model to be used for their offshore facilities in the North Sea. Falck (2011) provides a thorough review and presentation of the leak frequency modelling principles established during the study.

3.2.6 Generation of Exceedance Curves

There is currently a lot of industry interest in the generation of curves of the probability of exceeding a specified explosion load at a given location. These curves can relate to overpressure at a point, or averaged over a wall, or other explosion properties such as dynamic pressure or impulse. Exceedance curves are typically plotted on a graph with overpressure plotted on a linear scale on the horizontal axis and annual exceedance frequency plotted on a log scale on the vertical axis. An exceedance curve will always be a monotonically decreasing (discrete) function.

UKOOA Approach

The process provided by United Kingdom Offshore Operators Association (UKOOA) (UKOOA, 2003) is a method of medium complexity for the generation of exceedance curves for the purpose of identification of the design explosion events corresponding to the SLB and DLB. It is advisable to consider space averaged peak overpressures for this purpose as they are more representative of the general severity of the load case. The chosen scenarios will themselves give rise to simulations which have large local variations of peak overpressure.

NORSOK Z-013 Approach

Detailed procedure for generation of exceedance curves is presented in NORSOK (2010).

The procedure described in this document is meant to be used for detailed analyses of platforms in operation or in the project phases where the necessary information on all design elements influencing the risk picture is available. The purpose of the procedure is to standardise the analyses so that the risk of explosions can be compared between different areas, installations and concepts, even if the analyses are performed in different circumstances and by different personnel. Although this procedure is prepared for platforms, many of its guidelines might be useful for generation of exceedance curves for FPSO structures.

Pressure-Impulse Exceedance Surface

Czujko (2001) and NORSOK (2010) recommend generation of Pressure-Impulse exceedance surface instead of pressure exceedance curve, to obtain an improved characteristic of explosion pressure load.

Exceedance Curves for FPSO

A quantitative method for the calculation of explosion risk on FPSO is given in Paik and Czujko (2011c) and Paik *et al.* (2011). The method is a result of a Joint Industry Project on Explosion and Fire Design of FPSO. These procedures can be efficiently applied in offshore development projects, and the application includes the assessment of design explosion and fire loads as well as the quantification of effects of risk control options such as platform layout, location and number of gas detectors, isolation of ignition sources etc.

3.2.7 Design Explosion Loads

Design explosion loads were in the past derived from the worst credible event assuming a gas cloud of maximal extent with stoichiometric composition ignited at the worst time in the worst position. Usually the ultimate peak overpressure derived in this way is far too large to be accommodated by the structure.

NORSOK (2010) defines dimensioning accidental load as the most severe accidental load that the function or system shall be able to withstand during a required period of time, in order to meet the defined risk acceptance criteria. The dimensioning accidental load (DAL) is typically established as the load that occurs with an annual probability of 10^{-4} .

Design accidental load is a chosen load that is to be used as the basis for design. The applied/chosen design accidental load may sometimes be the same as the DAL, but it may also be more conservative based on other input and considerations such as ALARP. Hence, the design accidental load may be more severe than the DAL. The design accidental load should as a minimum be capable of resisting the DAL.

API (2006) and UKOOA (2003) recommend two levels of explosion loading by analogy with earthquake assessment. These are:

- Ductility level blast (DLB) / Design level blast
- Strength level blast (SLB) / Reduced blast load

Low risk installations may be assessed using only the DLB.

The ductility level blast is the design level overpressure used to represent the extreme design event. It is also defined as a low-probability high-consequence event, which must be investigated for at least retaining the integrity of the temporary refuge, safe muster areas and escape routes.

The strength level blast represents a more frequent design event where it is required that the structure does not deform plastically and that the SCEs (safety critical elements) remain operational. It is defined as a higher-probability, lower-consequence event. Performance criteria associated with the SLB may include elastic response of the primary structure, with the safety critical elements remaining functional, and with an expected platform restart within a reasonable period. This load case is recommended for the following reasons:

- The SLB may detect additional weaknesses in the design not identified by the DLB (robustness check).
- An SLB event could give rise to a DLB by escalation – this should ideally not occur as elastic response of SLB and supports should be maintained.
- The prediction of equipment and piping response in the elastic regime is much better understood than the conditions which give rise to rupture. The SLB enables these checks to be made at a lower load level often resulting in good performance at the higher level (strength in depth).
- The SLB is a low consequence event important for the establishment of operability.
- This load case offers a degree of asset protection.

Figure 3 represents an example (simplified) overpressure exceedance diagram. This curve is conventionally plotted with a logarithmic scale for the vertical frequency axis which gives the frequency per year for which the given overpressure will be exceeded. The horizontal axis is a linear scale usually with the peak space averaged overpressure

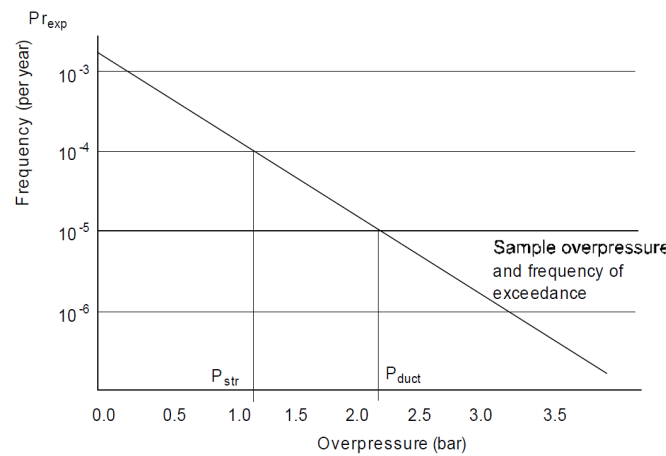


Figure 3: Example overpressure exceedance curve - location of SLB and DLB design load cases (P_{str} and P_{duct})

for the combustion region plotted in bar. This parameter gives a good general measure for the choice of design scenarios. Each of these scenarios may have a large range of local peak overpressures and associated durations within it.

The SLB overpressure, ' P_{str} ' may then be identified as that overpressure corresponding to a frequency one order of magnitude more frequent or with a magnitude of one third of the DLB overpressure, ' P_{duct} ', whichever is greater. The reason for the reduction factor of one third is related to the expected reserves of strength in the structure and the observation that the primary structure will often only experience received loads of this magnitude.

4 HYDROCARBON FIRES

The main purpose of fire analysis of offshore installations is to support risk calculations, particularly verify fire heat to topside structures and smoke exposure to escape and evacuation means.

The fire simulations are typically carried out using the commercial CFD codes as for example Kameleon FireEx, Fluent or Ansys.

The following loads resulting from fire event can be distinguished:

- Radiation from flame to the surroundings
- Convection from the hot combustion products passing over an object surface
- Conduction – not described in this paper, because it is usually small comparing to other methods of heat transfer
- Smoke load (soot and carbon monoxide) formed during an inefficient combustion of hydrocarbons

4.1 Fire Types. Assessment of Fire Action

The most complete classification of fire types, described in UKOOA (2006), comprises:

- gas jet fires
- two-phase jet fire
- pool fires on an installation
- hydrocarbon pool fires on the sea
- gas fires from subsea releases

- BLEVE (boiling liquid expanding vapour cloud explosion)

4.1.1 Gas Jet Fire

An ignited pressurised release of a gaseous material (most typically natural gas) will give rise to a jet fire. A jet fire is a turbulent diffusion flame produced by the combustion of a continuous release of fuel. Except in the case of extreme confinement which might lead to extinguishment, the combustion rate will be directly related to the mass release rate of the fuel. In the absence of impact onto an object, these fires are characteristically long and thin and highly directional. The high velocities within the released gas mean that they are relatively unaffected by the prevailing wind conditions except towards the tail of the fire.

The fire size is predominantly related to the mass release rate which in turn is related to the size of the leak (hole diameter) and the pressure (which may vary with time as a result of blow down). In the case of high pressure releases of natural gas, the mixing and combustion is relatively efficient resulting in little soot (carbon) formation except for extremely large release rates. CO concentrations in the region of 5 to 7 % *v/v* have been measured within a jet fire itself but this is expected to drop to less than 0.1 % *v/v* by the end of the flame.

Typical characteristics of jet fire are given in Table 3.

Effect of Deluge on Gas Jet Fires

Deluge has little effect on the size, shape and thermal characteristics of a high pressure gas jet fire. Therefore, the heat loading to engulfed obstacles is not diminished. There is some evidence that the deluge increases combustion efficiency resulting in lower CO

Table 3: High pressure gas jet fires, UKOOA (2006).

Characteristic	Unit	Size				Effect of confinement
		Small	Medium	Large	Major	
	kg s ⁻¹	0.1	1	10	>30	
Flame length	m	5	15	40	65	Affected by enclosure shape and openings.
Fraction of heat radiated, <i>F</i>		0.05	0.08	0.13	0.13	
CO level	% <i>v/v</i>	< 0.1	< 0.1	< 0.1	< 0.1	Increased up to 5% at a vent prior to external flaming, but after external flaming < 0.5% at the end of flame.
Soot concentration	gm ⁻³	~0.01	~0.01	~0.01	~0.01	Depends on equivalence ratio from 0.1 gm ⁻³ at $\Phi=1.3$ to 2.5 gm ⁻³ at $\Phi=2.0$.
Total heat flux	kW m ⁻²	180	250	300	350	Increased heat loadings up to 400kW m ⁻² (280kW m ⁻² radiative 120kW m ⁻² convective, $T_f=1600K$, $\epsilon_f=0.75$, $h=0.09$).
Radiative flux	kW m ⁻²	80	130	180	230	
Convective flux	kW m ⁻²	100	120	120	120	
Flame temperature, <i>T_f</i>	K	1560	1560	1560	1560	
Flame emissivity, ϵ_f		0.25	0.4	0.55	0.7	
Convective heat transfer coefficient, <i>h</i>	kW m ⁻² K ⁻¹	0.08	0.095	0.095	0.095	
Effect of deluge		No effect on heat loadings to engulfed objects. In far field, take 1 row water sprays, 2 rows and >2 rows at 12 lm ⁻² . May improve combustion efficiency and reduce CO levels within flame.				Risk of extinguishment and explosion hazard if deluge activated when enclosure is already hot and fire is well established.

and increased CO₂ levels within the flame. The major benefit of area deluge with jet fires arises from the suppression of incident thermal radiation to the surroundings, which protects adjacent plants and, in particular, aids personnel escape. Nozzles producing smaller droplet sizes can have an enhanced mitigation effect, but there is an increased risk that the droplets will be blown away by the wind.

4.1.2 Pool Fires on an Installation

A pressurised release of a hydrocarbon liquid which is not sufficiently atomised or volatile to vaporise and form a jet fire will form a pool. Similarly a spillage from non-pressurised liquid storage will result in a liquid pool being formed. Ignition of the vapours evolving from the liquid can lead to a pool fire which is a turbulent diffusion flame. For hydrocarbons such as condensate the vapours will evolve readily from a spillage and be easily ignited. For heavier hydrocarbons, such as diesel or crude oil, little vapour evolution occurs unless the fuel is heated and hence initial ignition of a spillage may be dependent on the presence of other fires in the vicinity providing sufficient energy to initiate vapour evolution.

Combustion of these hydrocarbons inevitably leads to the production of large quantities of soot, particularly in large pool fires where the size of the pool reduces the ability of air to mix with the fuel evolving in the centre of the pool. The soot emissions result in the characteristic yellow flame and large quantities of smoke can be produced to the

Table 4: Pool fires on the installation.

Characteristic	Unit	Methanol pool	Hydrocarbon pool diameter		Effect of confinement
			Small	Large	
	m	5	<5	>5	
Flame length	m	equal to pool diameter	twice pool diameter	up to twice pool diameter	Take values as per large hydrocarbon pool fire for worst case. If confinement is severe then mass burning rate will decrease to match available air flow and large external fire at vent is expected.
Mass burning rate	kgm ⁻² s ⁻¹	0.03	crude 0.045-0.06	crude 0.045-0.06	
			diesel 0.055	diesel 0.055	
			kerosene 0.06	kerosene 0.06	
			condensate 0.09	condensate 0.1	
			C3/C4s 0.09	C3/C4s 0.12	
Fraction of heat radiated, <i>F</i>		0.15	0.25	0.15	
CO level	% v/v	negligible	< 0.5	< 0.5	Increased CO up to about 5% v/v at a vent prior to external flaming, but after external flaming about 0.5 % v/v at the end of the flame.
Soot concentration	gm ⁻³	negligible	0.5 – 2.5	0.5 – 2.5	Soot levels up to 3 gm ⁻³
Total heat flux	kWm ⁻²	35	125	250	Take values as per large hydrocarbon pool fire.
Radiative flux	kWm ⁻²	35	125	230	
Convective flux	kWm ⁻²	0	0	20	
Flame temperature, <i>T_f</i>	K	1250	1250	1460	
Flame emissivity, <i>ε_f</i>		0.25	0.9	0.9	
Convective heat transfer coefficient, <i>h</i>	kWm ⁻² K ⁻¹	–	–	0.095	
Effect of deluge		Extinguishable using AFFF. Water soluble but effect of water deluge unknown.	Considerable fire control and potential extinguishment can be achieved. Expect a reduction in flame coverage of up to 90% within 10 minutes. Rapid extinguishment with AFFF. Up to 50% reduction in radiative heat flux to engulfed objects. In far field take <i>F</i> =0.8 <i>F</i> for 1 row of water sprays, <i>F</i> '=0.7 <i>F</i> for 2 rows and <i>F</i> '=0.4 <i>F</i> for >2 rows at 12 l/minm ² .		Expect reduced flame temperatures and reduced or no external flaming. Mass burning rate reduces to match available air flow.

extent that the smoke can lead to reduced thermal radiation to the surroundings by screening the radiant flame. Hence, the fraction of heat radiated, F , tends to decrease with increasing fire size, although the smoke hazard may increase.

Except in very large fires where buoyancy driven turbulence may become significant, the low velocities within the fire result in the flame being affected by the wind and this factor determines the trajectory of the flame. These low velocities also result in low convective heat fluxes to objects engulfed by the fire; the predominant mode of heat transfer being radiation.

Typical characteristics of pool fires are given in Table 4, UKOOA (2003b).

4.1.3 BLEVE

Fire impingement on a vessel containing a pressure liquefied gas causes the pressure to rise within the vessel and the vessel wall to weaken. Even within a short time, this may lead to catastrophic failure and the total loss of inventory. The liquefied gas which is released flashes producing a vapour cloud which is usually ignited. These events are known as Boiling Liquid Expanding Vapour Cloud Explosions, BLEVEs. This highly transient event generates a pressure wave and fragments of the vessel may produce a missile hazard leading to failure of other items in the vicinity and hence the potential for escalation. In addition, there is a flame engulfment and thermal radiation hazard produced by the fireball.

4.1.4 Simplified and Early Phase Design

In general assessment of fire loads is conducted by analysis of a number of probable fire scenarios. However NORSOK (2008) and DNV (2008) require that the structure is designed for the fire loads shown in Table 5 (unless otherwise documented):

4.1.5 Fire Scenarios in Design

The fire scenario establishes the fire type, location, geometry and intensity. NORSOK (2007) list the following fire scenarios that should be considered:

1. Burning blowouts in wellhead area
2. Fire related to releases from leaks in risers, manifolds, loading/unloading process equipment, storage tanks
3. Burning oil on sea
4. Fire in equipment on electrical installations
5. Fire on helicopter deck
6. Fire in living quarters
7. Pool fires in deck or sea

According to UKOOA (2003), the following specific considerations should be taken into account when defining fire scenario for an FPSO:

Table 5: Heat flux values, NORSOK Z-013.

Storage in an area	Design loads
Both gas containing equipment and oil containing equipment	<ul style="list-style-type: none"> • jet fire 250 kW m^{-2} for 30 minutes • pool fire 150 kW m^{-2} for the following 30 minutes
Only oil or condensate containing equipment	<ul style="list-style-type: none"> • pool fire 150 kW m^{-2} for the following 60 minutes
Only gas containing equipment	<ul style="list-style-type: none"> • jet fire 250 kW m^{-2} for 30 minutes

1. Oil storage tanks – may present hazard in the form of either large scale storage of stabilised crude or with empty storage tanks containing potentially explosive mixtures.
2. Non-process hydrocarbon inventories – The FPSO is a power-hungry installation and requires substantial stores of diesel to maintain station, process utilities power demands plus other life-supporting systems. The vessels are often located in difficult or remote places and will generally be designed to be “self-sufficient” for extended periods in the event that supply vessels cannot reach them.
3. Jet fires on main deck – The process decks on FPSO are often lifted clear of the cargo storage tank, a 5 m gap is not uncommon. The space provided also allows jet fires from the underside of the process to reach other process or utility modules without any impingement to reduce the effect of the flame.
4. Offloading and pool fires on the sea – Offloading to shuttle tankers is a regular event and poses a significant risk both on the FPSO and the shuttle tanker. The risks comprise the breakage or leakage of the transfer hoses and the potentially flammable mixing of hydrocarbon and air in the storage holds of FPSO and shuttle tanker. During the offloading operation, the shuttle tanker and FPSO are in relative proximity and the risks on either vessel are compounded by increased potential for escalation to another vessel.

4.2 Structural Response to Fire Load

Kim *et al.* (2010) have presented a study evaluating the load characteristics of steel and concrete tubular members under jet fire, with the motivation to investigate the jet fire load characteristics in FPSO topsides. ANSYS CFX, and KFX codes were used to obtain similar fire action in the numerical and experimental methods. The results of this study provide a useful database to determine design values related to jet fire.

4.3 Application of Deluge

Many international standards specify firewater deluge rates intended to protect personnel from thermal radiation from the fires during escape, and to cool equipment and structures affected by thermal radiation or direct flame impingement. Application and limitations of existing standards ISO 13702 and NFPA15 are discussed by Madonos and Ramm (2009). The assessment reveals that current standards are generic and in specific cases the application of these standards may lead to an unsafe design of deluge systems.

5 UNDERWATER EXPLOSIONS

After seeing the effects of UNDEX and surface explosions on various ships, such as the USS Cole, Superferry 14 and Limburg it would be very likely to predict similar attacks on offshore structures and platforms such as semi-submersibles, FPSOs, TLP, Spar, or offshore wind turbines. In order to determine the influence of such events this section examines structural response and loading characteristics for underwater explosions.

5.1 UNDEX Load Assessment

Underwater explosions result in loading mechanisms which exhibit significantly different time scales and loading. Initially a high pressure shockwave radiates from the point of detonation after which the explosive products form a superheated, highly compressed gas bubble. The gas bubble expands until the internal pressure becomes smaller than the ambient hydrostatic pressure at the depth of the detonation at which point it will collapse. These events are shown in Figure 4.

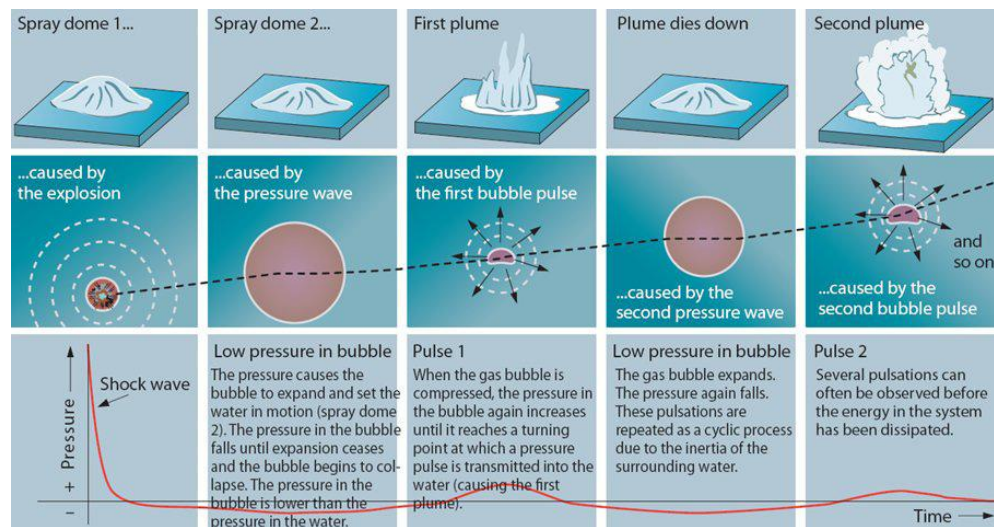


Figure 4: Stages of shock wave and the pressure effect on the sea surface

5.1.1 Experimental Methods for Determining Loading

Lee *et al.* (2010) performed experiments on rigid target plates in order to determine the behaviour in the loading of the bubble collapse at varying standoffs. They found that the bubble collapse loading increased with an increasing standoff up to approximately $0.8R$ at which point the load decreased, as shown in Figure 5. This also shows the bubble collapse impulse loading is significantly larger at close standoffs than the shock impulse loading.

5.1.2 Numerical Methods for Determining Loading

The most likely UNDEX event to occur on offshore structures is a close proximity event. The numerical approach would include a meshed model using a CFD or hydrocode model. Riley *et al.* (2010) performed simulations of the rigid target experiments that were conducted by Lee *et al.* (2010). The numerical simulations were conducted with the CFD code Chinook, which is developed and distributed by Martec Ltd. Simulations were performed for all experimental standoffs, 0.2 times the maximum gas bubble radius, R , up to a standoff of $2R$. The Chinook load predictions

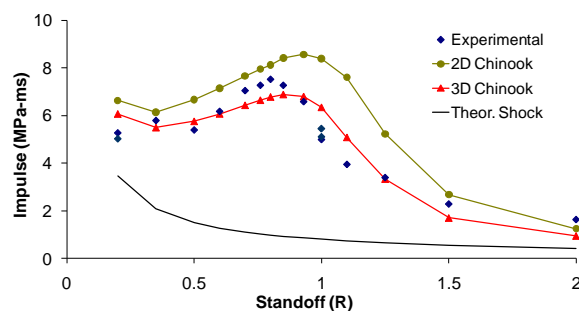


Figure 5: Comparison of experimental and numerical bubble collapse loads along with the theoretical shock loading for rigid plate experiments conducted at DRDC Suffield (Riley *et al.*, 2010)

were found to be qualitatively correct; however quantitative gaps remain, as shown in Figure 5.

5.2 Response Assessment

Determining the structural response for UNDEX can be very time-consuming and costly. Techniques used to determine the structural and operational integrity for offshore structures due to UNDEX events can be performed using numerical and/or experimental methods.

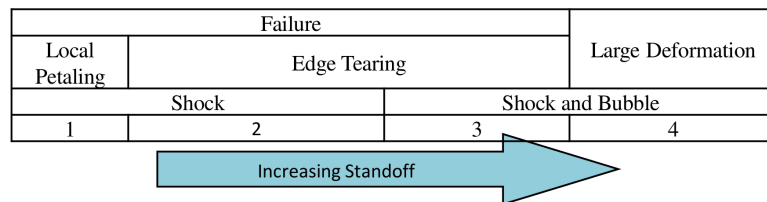
5.2.1 Experimental Response Assessment of Structural Components

Lee *et al.* (2008) showed the result of an extensive experimental close-proximity UNDEX assessment on small scale plate targets. The failure regimes that were determined from the experimental program are outlined in Figure 6.

5.2.2 Numerical Methods – Structural Response

Hung *et al.* (2009) studied the nonlinear dynamic response of cylindrical shell structures subjected to underwater explosion loading through experiments and numerical simulations implementing USA/DYNA software. For far-field UNDEX cases the accelerations and strains from the FE analysis showed good agreement with the experiments. For near-field cases the results were qualitatively correct, however quantitatively there were considerable differences.

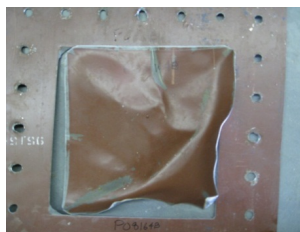
Zhang and Yao (2008) used a coupled BEM and FEM to calculate the coupling between the gas bubble and a structure. The toroidal bubble method developed by Wang *et al.* (1996a, 1996b) was implemented, which was expanded to three-dimensions by Zhang *et al.* (2001). Zhang and Yao calculated the response of a submerged cylinder to the bubble pulsating pressure, retarded flow, and the jet, and compared to experimental results. The error in numerical approach was found to be approximately 10 %. They



(a)



(b)



(c)



(d)

Figure 6: Failure regimes determined from small scale experiments conducted at DRDC Suffield (a) Failure regimes as a function of standoff, (b) local petaling failure (zone 1), (c) edge failure (zones 2 and 3), (d) large deformation (zone 4) (Lee *et al.*, 2008)

Table 6: Critical standoff range where failure occurs in the plate specimens (Lee *et al.*, 2008, Dunbar *et al.*, 2009)

Charge	Plate Thickness	Numerical Range	Experiment Range
20 g C4	1.21 mm	$0.69R - 0.75R$	$0.75R - 0.85R$
20 g C4	0.76 mm	$1.10R - 1.15R$	$1.15R - 1.25R$
50 g C4	1.21 mm	$1.06R - 1.15R$	$1.06R - 1.50R$

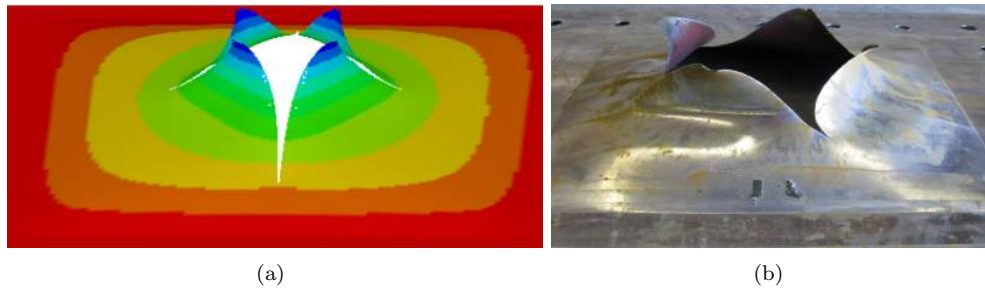


Figure 7: Plate failure from contact charge, (a) numerical prediction (b) experimental results

also showed that ship motion is linked to the phase of the bubble, so that there is a suction force as the bubble starts to collapse, causing ship to sag and putting it in a very vulnerable position for bubble collapse.

Dunbar *et al.* (2009) performed simulations of the small scale target experiments that were conducted at DRDC Suffield by Lee *et al.* (2008). Their results were compared to the ranges observed in the experiments, as shown in Table 6.

Riley (2008) simulated contact/near contact charges to determine the failure limits for centre plate punch-out failure in small scale targets using LS-DYNA and compared them to the experiments conducted at DRDC Suffield. It was found that through thickness shear stress is the dominate failure initiation mechanism for contact/near contact charges. The predicted failure pattern in the targets was found to agree reasonably well with the experiments, as shown in Figure 7.

Recent studies such as that by Yao *et al.* (2009) showed that the conventional shock factors have deficiencies in their ability to reflect the equivalency of structural explosive environments resulting from underwater explosions. Yao *et al.* proposed a new shock factor based on a spherical wave and the area of the structure normal to the wave propagation, S_E , which is a modification of the traditional shock factor C_2 , Eq. (1). Their results showed that their proposed shock factor significantly reduced the variations in the structure kinetic and potential energies for constant shock factors with varying charge weights and standoffs.

$$C_3 = \sqrt{S_E} C_2 \quad (1)$$

6 WAVE IMPACT

For many offshore structures wave impact loads is a design consideration that can influence both the required strength and, especially for ships and floaters, also the in service behaviour.

For ship-shaped structures and other floaters both steep waves impacting the bow and extreme loads due to wave breaking against platform columns may be important

design considerations. In Voogt (2004) it is thus reported that damage from wave impact loading has been experienced by several FPSOs. It also describes some results from the project SAFE-FLOW, where a design evaluation method to investigate bow impact from steep waves was developed. In Helland (2001) the development of a design tool for prediction of loads and responses due to impact from steep waves on ships and platforms is reported. The work is largely based on existing tools validated by model testing and calibration. The practical outcome of the project is a software package for practical engineering use.

A comparison between impact loads due to breaking waves obtained using an available (DNV) recommended practice with results from model tests is presented in Suyuthi (2009). The comparison was not made on an event by event basis, but rather on a q -probability wave impact load level for impact forces against platform columns. The conclusion was that the recommended practice is in reasonable agreement with model test for $q = 10^{-2}/year$, whereas model test results suggest larger impact loads than the recommended approach for $q = 10^{-4}/year$.

7 WAVE-IN-DECK

For fixed offshore structures wave-in-deck loads are becoming of increasing importance, especially for such steel jackets where subsidence has reduced the clearance between the underside of the deck structure and the sea surface to a critical magnitude.

Although the required clearance between the sea surface and the underside of the deck is a design parameter that is always carefully evaluated in any jacket design, more compaction of the reservoir than anticipated sometimes has taken place, and higher wave crests than originally anticipated have been experienced. This leads to a situation where the effective water depth is increased and in some cases to such an extent that the underside of the deck is impinged by large waves. This leads to an additional large wave load from the wave hitting the deck that was not anticipated during the original design, and thus typically is critical to both the deck structure and the support structure.

Current developments are directed towards a better prediction of both the subsidence and a refined prediction of abnormal wave crests that may impinge on the deck structure. In new designs the designer uses an additional safety margin on the calculated required minimum distance between the sea surface and the underside of the deck structure in terms of an air gap requirement. A state-of-the-art procedure to determine the required deck elevation can be found in ISO 19902 (2007).

For those fixed jacket structures that are in danger of experiencing wave-in-deck loads from waves with a return period of the order of 10,000 years or lower, wave-in-deck load becomes a design consideration. Such structures have been investigated in Van Raaij (2005 and 2007). These investigations focus on estimation of wave-in-deck loads on jacket structures in the North Sea for the rare 10,000 year event, especially on horizontal loads. They conclude that there is no general consensus on which method to use to calculate wave-in-deck loads. Further, a distinction is made between two main approaches: 1) the global or the silhouette approach (e.g. API and ISO) which use an effective deck area exposed to the pressure from the water particles, and 2) the component approach (e.g. Kaplan) in which loads on single members are calculated separately and finally added. Such methods typically determine the maximum wave-in-deck load which may be used directly for static analysis. However, it has been found that dynamics sometimes become important and may help the platform to survive,

particularly if the platform is sufficiently ductile. Therefore, a time history of the wave-in-deck load is needed in addition to the maximum value.

Both (Van Raaij, 2005 and 2007) conclude that vertical wave-in-deck loads are of considerable magnitude and therefore should be considered together with coexisting horizontal wave-in-deck loads. However, the actual detailed investigations are restricted to the horizontal wave-in-deck loads.

For the horizontal wave-in-deck load Van Raaij (2007) recommends to use a generic load time history based on a non-dimensional time found as real time divided by a basis load duration and a reference load. This procedure is intended for analyses where detailed information on the deck load is unavailable, and where a simplified 'rough-but-reasonable' estimate can be accepted. However, it should be noted that since this approach does not determine a co-existing vertical wave-in-deck load it does not constitute a complete design tool.

8 DROPPED OBJECTS

Daily lifting operations of any lifted object entails the risk of the object being dropped on the topside or sub-structure. Therefore, the associated risk is assessed based on a daily basis prior to the site-specific lifting operations. A typical risk assessment of dropped objects includes the analysis of the probability, also using statistics, as well as the structural consequences. Therefore, the object to be lifted by cranes and the operational area need to be identified. Furthermore, it is not the damage of a deck itself (e.g. by permanent deflection) that poses a threat during hydrocarbon fire, but the possibility of equipment damage below the deck. Consequently, the aim of a dropped object assessment is the protection of the equipment rather than the structure. Another threat is a dropped object, such as a container, which may bounce off the deck and roll into an unprotected area. Additionally, structural damage may occur when objects like brackets for jackets or pressurised tanks are lifted over the deck. Furthermore, crane booms may collapse to the deck or objects may strike a pipeline or subsea installations.

DNV and Norsok give some recommendations concerning loads and consequences of dropped objects.

Recommended Practice DNV-RP-F107 (DNV, 2001) presents a risk-based approach for assessing pipeline protection against accidental external loads. The DNV document proposes a classification for typical potential dropped objects as well as a classification for damage and assesses the energy absorbed by the impacting pipe with a simple analytical equation.

Norsok N-004 (2004) summarizes formulae for determination of the impact velocity (in air and in water), as well as formula for the strain energy dissipation and the associated damage (indentation or failure).

8.1 Loads Assessment

Typically, the load assessment is a result of a case-by-case risk analysis considering the frequency of occurrence, because there are no rules for the dropped object type. In other words, the risk assessment provides the scantlings of the dropped object and the operational area (target deck and/or plate) as well as the available kinetic energy. Additionally, a selection of possible loads can be found in OTO 2001 013 (HSE, 2001).

8.2 Consequences Assessment

At first, it can be noted that no guidelines for allowable consequences exist, yet NOR-SOK N-004 presents some allowable deformations on structure in terms of energy limits, but no measures for consequence control are provided.

Consequently, the shape, stiffness, orientation, mass and fall height of the dropped object are important parameters and it is necessary to assess them using a case-by-case risk assessment in line with the aforementioned load assessment.

Therefore, a conservative approximation typically utilizes a rigid indented, dropped objects respectively, to impact the operational area. The applicability of this approximation is however questionable and it may not be sufficient. Furthermore, besides the local impact, the global structural deformations may need to be considered, as well as the support effects. The latter may be addressed with analytical formulae; however, a direct simulation approach would be favourable.

8.3 Theoretical Approaches for Pipeline Impact

Besides the simplified indenter geometry, the support boundary conditions of the pipe resting on the soil need to be accounted for, whether it is simply supported, fixed or realistically somewhere in between these theoretical conditions. Therefore, DNV (2001), Wierzbicki and Suh (1988) and Ong and Suh (1996) provide simplified formulae for different boundary conditions.

Furthermore, Poonaya *et al.* (2007), Thinvongpituk *et al.* (2008) and Alashti *et al.* (2008) are concerned with the bending load of the pipe during the impact and propose formulations for ultimate bending moment rather than for dent depth.

Ong and Lu (1996) and Famiyesin *et al.* (2002) utilize the finite element method to obtain a range of results for different boundary conditions, which they utilize to obtain semi-empirical equations through curve fitting.

8.4 Numerical FE Approaches

FE-methods concentrate mostly on the assessment of the multitude of the possible scenarios of dropped objects and structural configurations to be analysed. An FE-analysis is the most flexible method and can account for the possible effects occurring and to assess the following relevant factors:

- Impact energy (constant drop height + variation of dropped object mass)
 - Dropped containers: post dropped object response: bouncing or rolling requires explicit FE-codes (contractual time limits often prohibit this phase, even though it can influence consequences significantly)
- Boundary conditions
 - Support stiffness, Length between supports, Internal pressure in pipes
- Material
- Discrete indenter shape and stiffness
- Indentation location

Although a drop test is typically of a low-speed impact, the short duration of this event and the nonlinear response of the interacting parts, with flexible stiffness behaviour, require the use of an explicit FE solver. It is also important to carry out the drop test at a series of impact angles as highly localised deformation can take place at certain angles. The material properties in the FE model will need to account for work hardening and implement the appropriate failure model, see also Chapter 17 on material modelling.

9 SHIP IMPACT ON OFFSHORE STRUCTURES

Over the past decades, the structural engineering design community has increasingly applied risk assessment methodologies for ship and offshore collision problems. The ISSC 2006 V.1 committee recommended risk assessment methodology to be more widely and frequently applied in analyses, and that structural crashworthiness be explicitly taken into account.

In the design of ships, risks due to collisions and grounding are in general not explicitly considered, except in specific cases. On the other hand, the offshore industry has a risk management concept that is significantly different from that of the marine industry. The offshore industry use systematic assessment procedures for fixed platforms that address the probability of occurrence, risk ranking, structural analyses, and acceptance criteria, see API (2000). Risk assessment methodologies are discussed in detail in Chapter 15: Design and assessment process.

Given that the collision event takes place, the loads and consequences of the collision event must be determined. A number of analysis tools and procedures for collision analyses have been developed and presented during the last decades. The current chapter gives an overview of common deterministic principles and methods applied in analysis of ship and offshore structures collisions.

The main concern in ship impacts on fixed platforms is the reduction of structural strength and possible progressive structural failure. However, the main effect for buoyant structures is damage that can lead to flooding and, hence, loss of buoyancy. The measure of such damages is the maximum indentation implying loss of water tightness. However, in the case of large damage, reduction of structural strength, as expressed by the indentation, is also a concern for floating structures.

9.1 Loads

In ship impacts on offshore structures, the loads are governed by the kinetic energy of the striking ship. The kinetic energy may be estimated from the mass of the ship, including the hydrodynamic added mass, and the speed of the ship at the instant of impact. If the collision is non-central, a part of the kinetic energy may remain as kinetic energy after the impact. The remainder of the kinetic energy has to be dissipated as strain energy in the installation and in the vessel. Generally this involves large plastic strains and significant structural damage to either the installation, the ship or both. The mass of the offshore structure is usually much larger than the striking ship, and most kinetic energy will be dissipated to strain energy.

The collision event is a complex interaction between vessel motion, offshore structure motion, interaction with the fluid, global hull response in the ship and offshore structure, inelastic deformations in both structures, friction etc. A common, simplified approach is to split the problem into two uncoupled analyses; external mechanics and internal mechanics. The external mechanics analysis uses global inertia forces and hydrodynamic effects to estimate the amount of kinetic energy available to be dissipated to strain energy during collision. The internal mechanics analysis calculates the energy dissipation and distribution of damage in the two structures.

The external dynamic analysis is able to predict the motion of the vessel and offshore structure in the surrounding water during the collision event. The goal of the analysis is to estimate the fraction of the initial kinetic energy which will be released for plastic deformation and rupture in the ship and offshore structure. Several methods are available; full time-domain analysis, simplified analytical methods, simplified formulas

from rules, ex. NORSOK (1998). Zhang (1999) showed that external mechanic methods developed for ship-ship collisions may in general apply to ship impacts on offshore structures as well. However, offshore structures are usually moored, which can give different external mechanics characteristics from those of ship-ship collisions.

Several assumptions are required to split the problem into two individual problems; negligible interaction between the global movement and local plastic deformation, dominant inertia forces, no damping, etc. Hence, the uncoupled analysis methods can only predict the penetration for few conditions correctly. Arbitrary collision angles, sliding and different mass ratios are generally difficult to capture and these methods can only predict the response with high accuracy in symmetric collision events.

Efficient coupled dynamic collision simulation methods are available and are able to take the ship motion and structural deformation and their interaction into account simultaneously. Coupled methods will in general provide a better energy correspondence and increased accuracy. Pill and Tabri (2009) present an efficient and robust method for coupled dynamic analysis in LS-DYNA. The method considers the most important force components accurately; the inertia force and contact force. The ship motions are limited to the horizontal plane which enables neglecting the restoring force, buoyancy and gravity, which are not straightforward to include. Similar methods are available, but the advantage of the proposed method is that special user subroutines are not required, and only conventional tools are used.

9.2 Consequences

The consequence of the collision is dependent upon numerous parameters, but the most important factor is the energy released during the collision event. In split methods, the results from the external mechanics analysis may be compared to the absorbed energy vs. penetration curve found from the internal mechanics analysis. Integrated approaches take both into account simultaneously, and at a higher accuracy.

The analysis methods of internal mechanisms can be categorized into four groups; Simple formulae, Simplified analytical approach, Simplified FEM and Non-linear FEM simulation.

The simple formulae are mostly used to estimate the initial energy absorption. Simple formulae have been developed for a wide range of problems, including head-on collision, grounding and ship-bridge collision.

Simplified analytical approach may be used to calculate the initial energy absorption and loads. This group of methods may estimate the basic characteristics of structural crashworthiness with minimized structural modelling efforts. Applications of this methodology to various collision and grounding situations were summarized extensively by the ISSC 2003 Committee V.3.

Non-linear FEM simulation is the preferred choice for advanced and accurate analysis of collision events. Progress in software development and hardware technology has enabled advanced non-linear analyses including large deformations, sophisticated non-linear material models, complex and robust contact algorithms and more accurate modelling of rupture. Several commercial non-linear FE-solvers are available and commonly used for collision analysis. Non-linear FEM simulations have become standard, and numerous examples have been presented in conference proceedings. The selection of elements, meshing, loads and boundary conditions have become more straightforward because of extensive development in commercial software codes. The material definition and selection is still a major challenge, especially with respect to prediction

of ductile crack initiation and propagation. Several models and methods have been proposed and used with success lately, see Chapter 17 for details.

A simpler approach is to utilize force-deformation curves. NORSOK (1998) Appendix A, includes recommended design curves for supply vessels for various scenarios. The code includes characteristic force-deformation curves for tanker (i.e. FPSO) bow impact as well.

Simplified, analytical methods may also be used to estimate the damage. These can be divided into three classes; empirical methods, analytical plastic methods and analytical buckling methods. The empirical methods relate the energy dissipation to the volume of the damaged material in the offshore structure and striking ship. This may be used to establish a relationship between the intrusion depth of a structure and the amount of absorbed energy for ships at collision. The analytical plastic methods calculate the entire crushing process, and will assess the average collision force. The method assumes that the structure is built from a few fundamental components. The energy dissipation for each component is estimated and the total energy dissipation is found by summarizing for all components.

The analytical buckling method assumes that the maximum strength of a component may be calculated from the plate slenderness factor of basic components. The slenderness factor is found by reducing the cross section to flat flange elements.

9.3 Literature Study

Isshiki *et al.* (2010) presents a model where the struck ship is replaced by a system composed of rigid bars and elasto-plastic hinges. This model not only can express the response of the struck ship more reasonably, but also does not require much time for numerical simulation.

Hogstrom *et al.* (2010) presents Finite Element analysis of a ship-to-ship collision scenario, where the damage opening of a struck ship is simulated for a selection of damage degradation models and realistic material properties. Both the model and material properties include uncertainties. A holistic approach is developed, combining structural integrity and damage stability research with the use of a systematic parameter (sensitivity) and collision-scenario-based analysis.

Hu *et al.* (2010) study a collision scenario in which a moored semi-submersible is struck by a containership through the model test, simplified analytical method and numerical simulation. Two special devices, Ship Launching Device and Energy Absorbing Device are used for the model test. It is shown that the prediction by a NTNU in-house program developed by simplified analytical method is consistent with the results by the model test. And then, it is shown that the collision force dominates the accidental moment, and that the tension forces of the mooring lines are much smaller than the collision force, with an obvious lag behind.

Qiu and Grabe (2010) carry out Finite Element analysis using a Coupled Eulerian Lagrangian approach in order to simulate the collision experiment of waterway embankments of inland waterways with an experimental ship. The stopping distance and the reaction force obtained by the numerical simulation shows good agreement with the field test results. The effects of initial velocities and bow types of the ship on the collision process are also investigated.

Lin *et al.* (2010) show how FEM is used to simulate the collision process of two submersibles. Stress and strain distributions, collision forces, and plastic energy absorption are obtained. The motion lag of the struck submersible in the collision process is

discussed and it is found that it is sensitive to impact velocity which increases with the increase of velocity.

10 EARTHQUAKE

The earthquake induced loading of an offshore structure can cause severe structural damage due to the ground accelerations or as a result of subsidence. Hence, according to NORSOK standard N-003 earthquake actions should be determined based on relevant tectonic conditions and seismological time histories describing further earthquake motions including peak ground accelerations at the site in question. In the absence of such information, the peak ground acceleration at annual exceedance probabilities of 10^{-2} and 10^{-4} given in seismic zonation maps in NFR/NORSAR (1998) can be applied.

However, in severe cases like the 3.11 disaster in Japan, those measures would have failed, because the intensity of the earthquake (M 9.0) surpassed previous measurements, and the occurrence of more than 400 aftershocks with $M > 5.0$, wherefrom five aftershocks with $M > 7.0$, contributed to the damage, see Figure 8. Consequently, the soil-structure interaction as a result of the subsidence of up to 5.3 m horizontally and 1.2 m vertically would not have been predicted sufficiently based on history measurements, see Figure 9.

Typically, earthquake design includes an ultimate strength check of relevant components as well as accidental limit state check of the overall structure to prevent collapse during the earthquake; for details see for example NORSOK N-001.

Furthermore, structural action effects may be approximated using simplified response spectra, also considering different soil conditions for specific seismic zones. Additionally, before any detailed analysis is carried out, an estimate of the global force based on a single dynamic mode of the response spectrum may justify its necessity. However, such simplified analysis may be limited to the underlying soil conditions and this has to be judged on a case-by-case basis due to the large regional variations, see Chapter 17.3 for soil materials and the references there in. For details on structure-soil interaction see Clouteau *et al.* 2012 and Menglin *et al.* 2011. Furthermore, the structural

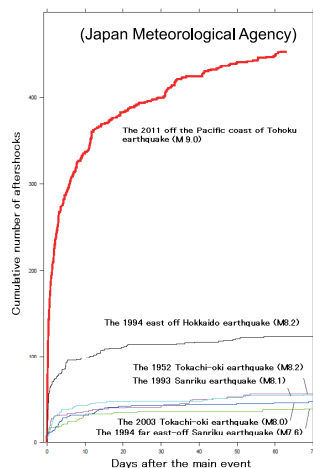


Figure 8: Cumulative number of aftershocks (Fujita, 2011)

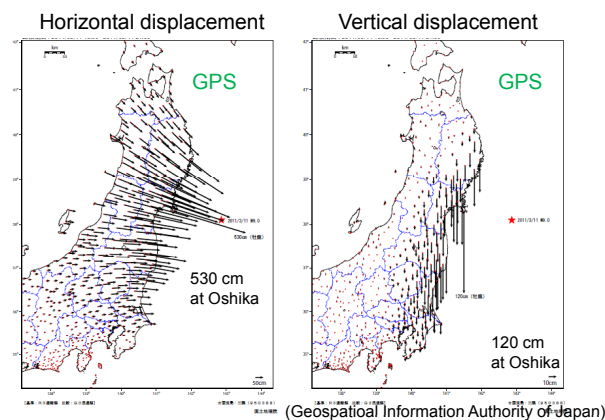


Figure 9: Subsidence due to coseismic slip (Fujita, 2011)

response can be assessed with the nonlinear finite element method with confidence, if however, the loading condition is known accurately.

11 ABNORMAL ENVIRONMENTAL ACTIONS

Abnormal waves have many times been reported in the maritime folklore, but until recently it was believed that these huge waves only existed in legends. These waves have been known under many different names such as: rogue waves, freak waves, killer waves, extreme waves and abnormal waves. In the following the term freak waves will be used.

11.1 Freak Waves

In oceanography freak waves are according to WIKIPEDIA (2010) defined as waves whose height is more than twice the significant wave, which is itself defined as the mean of the largest third of waves in a wave record. Therefore freak waves are not necessarily the biggest waves found at sea. They are rather, surprisingly large waves in a given sea state.

The existence of freak waves was not positively confirmed until New Year's Day 1995 at the Norwegian Draupner jacket platform, where an unusually large wave was recorded and analysed (Haver, 2004a). The wave record is shown in Figure 10.

A close examination showed that the wave crest was large, but not beyond the abnormal (10^{-4}) wave crest specified for the design. However, it was much higher than could be associated with the measured sea states. The crest height was well beyond the 10^{-2} crest height, but the platform loads did not exceed that level. This suggests that the shape of the wave differs from typical design waves. In Haver (2004b) it was found that freak waves should be considered a separate population well beyond the population used for design purposes. Further, it was found that freak waves are not seen as likely to represent a problem for offshore structures with the frequency of occurrence experienced so far. Nevertheless, to achieve robustness against unknown freak wave extremes it is in Haver (2004b) recommended to include an accidental (10^{-4}) wave event in the design process.

11.2 Tsunami Waves

Very little guidance is provided in currently available structural design codes, standards and guidelines on actions from tsunamis. However, in FEMA P646 (2008) important experience in relation to tsunamis and the design actions generated by tsunamis

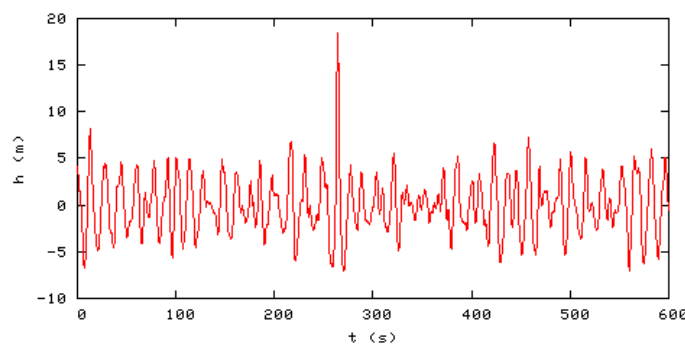


Figure 10: The Draupner wave, a single giant wave measured at New Year's Day 1995, finally confirmed the existence of freak waves, which had previously been considered near-mythical.

is discussed. Although FEMA P646 (2008) addresses the design of structures for vertical evacuation from tsunamis on the shore, the fundamental issues, namely how to predict design actions from tsunamis are shared with offshore structures.

Tsunamis are created in a variety of ways. Perhaps the best known generation mechanism is earthquake-induced displacement of the sea bottom, which causes a related sea-surface elevation that then propagates away from the generation area due to gravity. In case of the 3.11 disaster in Japan, the tsunami had a wave height of up to 25 m as a result of the earthquake. Additionally, submarine slumping of the offshore shelf or the impact of a terrestrial landslide into the sea can also cause devastating tsunami wave.

Similar to other hazards structural design criteria for tsunami effects should be based on the relative tsunami hazard, i.e. given a known or perceived tsunami threat in a region, the first step is to determine the severity of the tsunami hazard. This involves identification of potential tsunami generating sources and accumulation of recorded data on tsunami occurrence and run up. The assessment of tsunami hazard can include a probabilistic assessment considering all possible tsunami sources, or a deterministic assessment considering the maximum tsunami that can reasonably be expected to affect a site. Once potential tsunami sources are identified, and the level of tsunami hazard is known, site-specific information on the extent of inundation, height of run up, and velocity of flow is needed. Given the tsunami hazard and extent of inundation, the potential risk of damage, and loss of life must then be evaluated.

In FEMA P646 (2008), the design tsunami is termed the Maximum Considered Tsunami (MCT). It is anticipated that the hazard level corresponding to the Maximum Considered Tsunami will be consistent with the 2500-year return period associated with the Maximum Considered Earthquake used in seismic design.

For site-specific tsunami hazard assessment, the Maximum Considered Tsunami, should be developed using the Deterministic Maximum Considered Earthquake (Deterministic MCE) as the source (initial condition) of the tsunami model.

It should be noted that the above recommendations do not include modelling for tsunamis induced by landslides, volcanoes, or meteorite impacts.

There is significant uncertainty in the prediction of hydrodynamic characteristics of tsunamis because they are highly influenced by the tsunami waveform and the surrounding topography and bathymetry.

It is essential for the area of refuge to be located well above the maximum tsunami inundation level anticipated at the site. Determination of a suitable elevation for a tsunami refuge must therefore take into account the uncertainty inherent in estimation of the tsunami run up elevation, possible splash-up during impact of tsunami waves, and the anxiety level of evacuees seeking refuge in the structure. To account for this uncertainty, the magnitude of tsunami effects is determined assuming a maximum tsunami run up elevation that is 30 % higher than values predicted by numerical simulation modelling or obtained from tsunami inundation maps. It is further recommended that the refuge elevation include an additional 3 m allowance for freeboard above this elevation. The recommended minimum refuge elevation is therefore the anticipated tsunami run up plus 30 % plus 3 m.

Seismic loads are not considered to act in combination with tsunami loads. While aftershocks are likely to occur, the probability that an aftershock will be equivalent in size to the Maximum Considered Earthquake and will occur at the same time as the maximum tsunami inundation is considered to be low.

12 ICE AND ICEBERGS

The importance of marine transport in the Arctic Regions is further increasing as the ice-extends decrease. The latter, may also contribute to severe ice conditions with large drifting floes, ridges and icebergs as a result of calving or ice field separation combined with northern winds and currents. Therefore, structures need to be designed to withstand the local pressures and the resulting global impact. These structures, usually fixed and rarely floating can collide with drifting ice, ridges, crawlers or icebergs. As a result, their impact velocity is relatively low, however, neglecting a ship colliding with ice at service speed.

Several guidelines and regulations are concerned with brash ice conditions, e.g. a broken channel in level ice; see for example Finish-Swedish ice rules. However, only a limited number of regulations are dealing specifically with extreme loads.

ISO 19902 is concerned with the energy absorption during an iceberg impact arising from the combined effect of local and global deformation. The energy absorbed shall be compared with, and equated to, the impact (kinetic) energy due to a ship collision, and the results shall be documented. Before the publication of ISO 19906 on Arctic structures, all requirements for the design of structures for ice and iceberg loads shall be in accordance with CAN/CSA-S471-04. Furthermore, the design of stiffened plate panel configurations other than uniaxial stiffened plate panels shall be in accordance with other design standard such as DNV-RP-C201 or API Bulletin 2V.

The NORSOK standard N-003 8.3.2 concerns vessel collisions and should be followed according to 6.4.2.3 for iceberg collisions. Furthermore, 6.4.2.3 states the geographical location for iceberg collisions in the Barents Sea together with the probability of exceedance. Additionally, 8.3.2 states that all relevant traffic data needs to be collected for the site in question including icebergs. Hence, the most probable loading may be derived from this collection. Furthermore, a simplified supply ship impact scenario is described, which may be considered for an iceberg collision too. However, a design iceberg and scenario are yet missing a standardized load assessment.

Furthermore, new Polar class rules are about to be released, with eventually more details on such extreme ice impact. The likelihood of, i.e. iceberg impact, needs to be investigated for the site in question. Furthermore, one of the main challenges in iceberg collisions with ships and offshore structures is to obtain the correct magnitude of local pressure acting on the surface of the structure as a result of the ice impact. Recent studies involve full-scale measurements of the local ice pressure during the CCGS Terry Fox bergy bit impact, see Ritch *et al.* (2005) and Johnston *et al.* (2007). Local pressures of up to 10 MPa have been reported, but higher values may be probable too. Furthermore, Eik and Gudmestada (2010) found that the maximum impact load corresponding to a 10,000-year event was 85 MJ and that this value can be reduced to 1.8 MJ if an iceberg management system with iceberg detection, iceberg deflection and disconnection capabilities including emergency disconnect is used.

In this respect, commonly the existing standards fail to give a clear design guideline concerning iceberg collision and need to be improved. Furthermore, new ice material models should be developed to contribute to development of these guidelines; see also Chapter 16.

13 FLOODING

Accidental flooding is one of the main topics related to incidents connected with ships and offshore structures. The obvious major concerns are loss of buoyancy and stability.

Members of the International Towing Tank Conference have carried extensive research on this topic and are still continuing. A recent article by Santos (2009), starts with a nice introduction into the topic. A third issue related to flooding is structural loading, which seems to have attracted much less attention from researchers. Therefore Committee V.1 has decided to dedicate this chapter to this issue.

13.1 State-of-the-art

Very little literature is available on the effect of flooding on the structural integrity of a ship or a floating offshore structure. This may not be surprising because in most reported disasters where flooding played a role, usually (hydrostatic) stability is recognised as the main issue.

When a strength issue comes into play there are in principle two mechanisms;

1. The floodwater changes the ‘deadweight’ distribution along the ship’s hull girder to such an extent that still water bending or torsional moments exceed the capacity of the structure, or reduce the strength margin available for wave loads and loads due to inertia forces.
2. The motions of the floodwater cause pressure loads which exceed the capacity of the bulkheads of the flooded compartments.

Few publications are known to the committee which deal with hull girder loads due to flooding explicitly. Korkut *et al.* (2005) report model test results with a damaged ship ($L_{pp} = 173\text{ m}$) in regular waves. They demonstrate that hull girder loads may increase significantly under damaged condition, to such an extent that they should not be ignored. Figure 11 shows the increase of the torsional moment ‘RAO’ due to engine room flooding.

SSC report 445 does not confirm this finding, it actually states that for the ship investigated, a cruise liner with an $L_{pp} = 242\text{ m}$, bending moments tend to decrease in damaged condition. However, this may be caused by the way in which the analyses were made, where bending moments were calculated through an equivalent design wave, based on predicted wave bending moments with the Ship Motion Program software from David Taylor Model Basin.

It is convenient to refer to the flood water pressure load mechanism as sloshing loads. Gao *et al.* (2011), report on an extensive research effort on the numerical simulation of flooding. The paper includes flood water load predictions on bulkheads which compared favourably with results from model scale tests on a barge. Another interesting

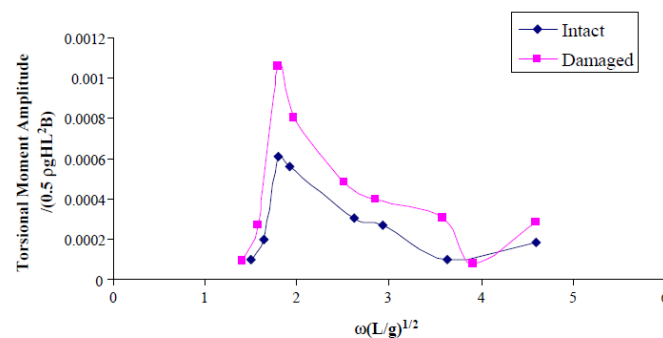


Figure 11: Comparisons of torsional moment R.A.Os at mid-ship in beam seas for large wave height (Korkut *et. al.*, 2005).

article is from Le Touzé *et al.* (2010), who report on the use of Smoothed Particle Hydrodynamics (SPH) for predicting green water and flooding phenomena, which compare favourably with test results. Loads due to flooding are not included in the reported study. However, an article by Dolorme *et al.* (2005), also describes the use of SPH, but now related to sloshing. Satisfactory results are reported with respect to predicted loads on bulkheads.

Ming *et al.* (2010), report on sloshing load prediction methods based on the Volume Of Fluid method. The method is validated against test data recommended by the 23rd ITTC Committee as a benchmark case.

13.2 Suggestions for further Research

Availability of well documented sloshing test data, including the geometry of the inner tank structure, tends to be limited. The committee suggests including data collected by SSC on this topic (SSC report 336) in validations.

It is also suggested to extend efforts on research related to global internal loads of floating structures while flooded, including sloshing resonance while in sea states.

14 ILLEGAL ACTIVITIES LIKE USE OF EXPLOSIVES AND PROJECTILES

It is projected that world consumption of marketed energy is to increase by 49 % from 2007 to 2035 (EIA, 2010). This dependency will grow inexorably as the populace in developing countries replace the use of traditional fuels with marketed ones, such as propane and electricity. Most of the estimated remaining energy reserves are located offshore politically unstable nations, while new explorations take place in areas of long-term assertions (Barents Sea, Aegean Sea, Libyan Sea). During the Iraq-Iran war (1980-1988) several oil fields were attacked and damaged significantly. The Dorra Field is a characteristic example where platforms were attacked indiscriminately during the conflict.

14.1 Terrorist Attack Assessment and Consequences

According to the RAND Corporation's terrorism database only 2 % of all terrorist incidents since 1969 are conducted in the marine environment. Some examples of terrorist attacks on offshore vessels are shown in Figure 12. In these cases most attacks resulted in severe damage to the target structures. The M Star tanker had significant hull deformation, as shown in Figure 12(a), the MV Limburg, Figure 12(b), USS Cole Figure 12(c), had large holes blown in the side of the vessels and Superferry 14, Figure 12(d), sunk as a result of the explosions.

The costs and environmental effects associated with structure damage due to a terrorist attack can be significant, in the case of the MV Limburg, 90,000 barrels of oil leaked into the Gulf of Aden.

Although terrorist attacks have historically been carried out with the use of explosives, this does not preclude future threats from the use of missiles, ramming with large vessels, or use of divers or unmanned underwater vehicles from planting or detonating underwater charges. The structural assessment and load definitions due to a terrorist event could fall into several categories that have been detailed throughout this report including hydrocarbon explosions and fires, underwater explosions, ship impacts and flooding.

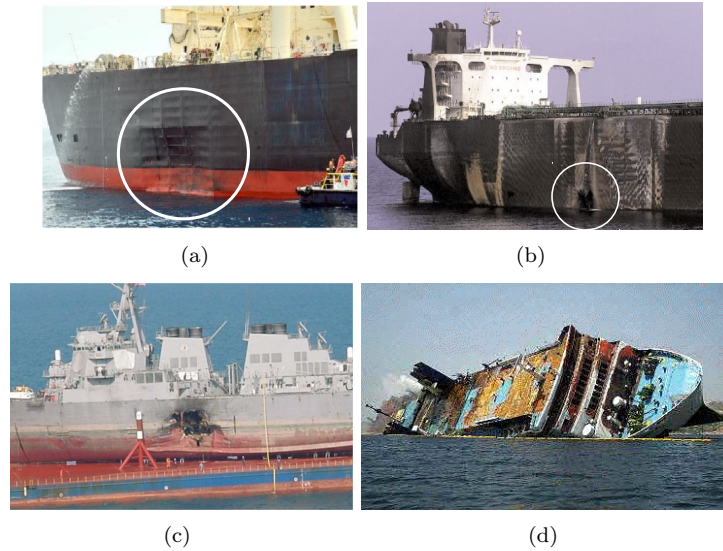


Figure 12: Consequences of historical terrorist attacks on offshore structures (a) M Star tanker, (b) MV Limburg supertanker, (c) USS Cole warship, and (d) Superferry 14

14.2 Definition of Loads

An intentional explosion onboard an offshore platform may result from a relatively small incendiary device after an intended gas leak or an improvised explosive device (IED) planted either above or below the water. Standoff weapons can also be used from a distance outside the facility giving terrorists a safe vantage distance.

Rocket-propelled grenades (RPG) are widely sold in every corner of the world nowadays. It is estimated that as of 2002, at least 9 million RPGs had been produced around the world (O'Sullivan, 2002). RPGs' light weight, low acquisition cost, ruggedness, and reusability are some of the key reasons that make them a weapon of choice with some militia and irregular forces in Southeast Asia and the Middle East (Grau, 1998). RPGs are capable of penetrating up to 500 mm of steel. There is no doubt that their jet can penetrate thin plates used in marine and offshore structures. Figure 13 shows the ballistic response of a cross stiffened panel upon impact with an RPG. The shaped charge jet effortlessly penetrates the panel causing an insignificant out-of-plane displacement in the target.

Figure 14 depicts the out-of-plane deformation of a 10 mm witness plate impacted by a scaffold clip at 150 m/s. Although the plate does not perforate, the kinetic energy of the projectile could inflict lethal injuries on personnel. In a similar manner tools, pipes, fire extinguishers and other loose objects can be turned into projectiles with grave consequences.

Another popular modus operandi is that of explosively-laden skiffs or zodiacs that detonate alongside their target (USS Cole type of attack). The riser as well as tubular members of the facility at the waterline can be damaged in this manner leading to possible fire and or environmental damage or loss of the platform. Figure 15 and Figure 16 depict damage inflicted to a riser and typical offshore joint from a detonation at sea level.

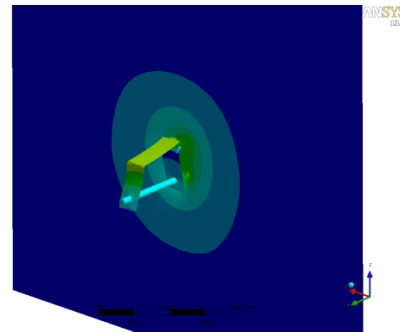


Figure 14: Deformation of witness plate from impact of a scaffold clip

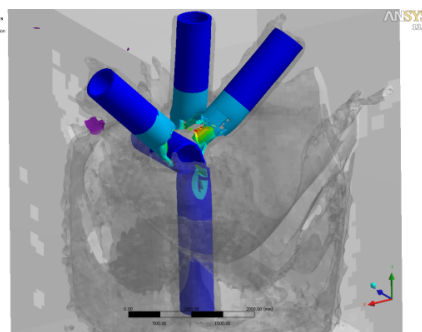


Figure 16: Structural response of a typical joint commonly encountered in offshore structures

The design and assessment process is a part of the total safety management of offshore installations. In Moan (2007) an overview of important developments regarding safety management of offshore structures is given. It is found that the risk can be controlled by the use of adequate design criteria, inspection, repair and maintenance of the structure as well as quality assurance and control of the engineering processes.

By experience, it is often human errors that initiate catastrophic accidents. Damage tolerance is therefore seen as a desirable feature of a structure. Moan (2007) demonstrated how an acceptable risk level may be achieved by introducing Accidental Collapse Limit State (ALS) criteria in the design of offshore structures.

The new ISO standards for offshore structures, see e.g. ISO 19902 (2007), offer a practical implementation of the design approach against accidental or abnormal actions through the identification of relevant hazards and subsequent design using ALS criteria, in principle as proposed in Moan (2007).

15.1 Codes and Standards

15.1.1 General

The focus on Accidental Collapse Limit State (ALS) criteria in the design requirements in different design standards has increased over the last 10-15 years especially for structures of high importance. Irrespective of this development, different approaches both in complexity and completeness are currently used in different design standards. Within the offshore industry the most important sources are Norsok N-004 (NORSOK, 2004), and ISO (ISO 19902, 2007), and the text in this chapter is largely reflecting these criteria.

15.1.2 Robustness

In the new ISO standards for offshore structures it is required that damage from events with reasonable likelihood of occurrence shall not lead to complete loss of integrity of the structure. Further, it is emphasized that the structural integrity in the damaged state shall be sufficient to allow for process system close down and a safe evacuation.

In ISO 19902, (ISO 19902, 2007), it is specifically stated (Clause 7.9) that: 'A structure shall incorporate robustness through consideration of the effects of all hazards and their probabilities of occurrence, to ensure that consequent damage is not disproportionate to the cause'.

The robustness concept is therefore closely related to accidental actions and abnormal actions, consequences of human error and failure of equipment. In ISO terminology such situations are denoted 'hazardous circumstances' or briefly 'hazards'.

Robustness is achieved by considering accidental limit states (ALS) that represent the structural effects of hazards. Ideally all such hazards should be identified and quantified by means of a risk analysis, but in many cases it is possible to identify and quantify the most important hazards based on experience and engineering judgement.

15.1.3 Accidental Limit States

Accidental situations relate to two types of hazards:

1. *Hazards associated with identified accidental events*, often those from ship impact, dropped objects, fires and explosions.
2. *Hazards associated with abnormal environmental actions*, typically environmental actions with a return period of the order of 10,000 years.

The two types of hazards are different by nature. In principle accidental events can in some cases be avoided by taking appropriate measures to eliminate the source of the event or by bypassing and overcoming its structural effects. In contrast to this, the possible occurrence of abnormal actions cannot be influenced by taking such measures.

An accidental design situation is considered in an accidental limit state (ALS), and normally comprises the occurrence of an identified accidental event or abnormal environmental actions, in combination with expected concurrent operating conditions and associated permanent and variable actions.

15.1.4 Designing for Hazards

When the hazard cannot reliably be avoided, the designer has a choice between minimizing the consequences (the consequences of damaging or losing a structural component due to the hazard), or designing for the hazard (making the component strong enough to resist the hazard). In the first case, the structure should be designed in such a way that all structural components that can be exposed to a hazard are non-critical,

i.e. can be lost without causing failure of the whole structure or a significant part of it. In the second case, critical components that can be exposed to hazards (failure of which would cause failure of the whole structure or a significant part of it) shall be made strong enough to resist the hazard considered.

It is specifically noted in ISO 19902 that the robustness requirements do not imply that structures shall be able to survive removal of any structural component. If there is no hazard, then there is no requirement in relation to robustness. Also, only one hazard at a time should be considered.

15.2 Risk Assessment Issues

15.2.1 General

This chapter reviews and discusses the framework for a risk-based design against accidental actions in a broader perspective. Conceptually, the main elements in such a discussion are: the probability of a given accidental action, the conditional probability of damage given the accidental action and finally the conditional probability of a global failure given damage. In the following chapter these aspects will be discussed in some detail.

15.2.2 Accidental and Abnormal Actions

In Figure 17 taken from Moan (2007) accident rates for mobile (drilling) and fixed (production) platforms have been shown according to the initiating event of the accident. Although the curve is rather old, the general trend is still believed to be true. It is most noticeable that none of these accidents should occur, but they still do so because of operational errors and omissions. Despite the efforts made to avoid error induced accidental actions they cannot be completely eliminated. Therefore, Accidental collapse Limit State (ALS) criteria are introduced to prevent progressive failure.

In a rational ALS criterion the accidental action should be defined as a characteristic value preferably defined in probabilistic terms. This has been done both in ISO 19902 (ISO 19902, 2007) and Norsok (NORSOK, 2004) where the characteristic accidental action for offshore structures is specified by an annual exceedance probability of 10^{-4} .

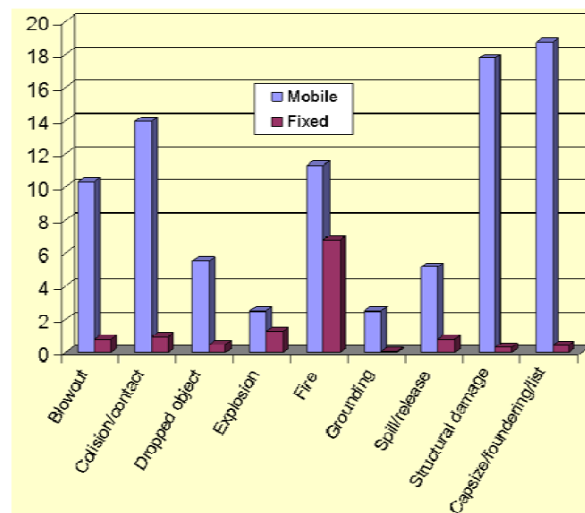


Figure 17: Number of accidents per 1000 platform-year

$$P_{FSYS}(i) = \sum_{j,k} P[FSYS | D] \cdot P[D | A_{jk}^{(i)}] \cdot P[A_{jk}^{(i)}]$$

probability of damaged system failure under relevant F&E actions
 probability of damage, D given $A_{jk}^{(i)}$ (decreased by designing against large $A_{jk}^{(i)}$)
 probability of accidental action at location (j) and intensity (k)

For each type of accidental action
 $P[A_{jk}^{(i)}]$ is determined by risk analysis while the other probabilities are determined by structural reliability analysis.
 $P[FSYS | D]$ is determined by due consideration of relevant action and their correlation with the hazard causing the damage

Figure 18: Probability of system loss due to accidental action (i)

The ALS criterion also applies to abnormal environmental conditions such as hazards associated with abnormal environmental wave actions. In this connection focus should also be given to abnormal waves with high crest or unusual shape – especially in such cases where the 10^{-2} wave might not reach the platform deck, but the 10^{-4} wave crest hits the deck and causes a significant increase in the wave loading.

15.2.3 Framework for the Design Against Accidental Actions

As outlined in Moan (2007) a truly risk based design should account for the various sequences of progressive development of accidents into total losses. However, in a design context simplifications are necessary. One such approach is, as previously discussed, to prevent escalation of damage induced by accidental actions by requiring the structure to resist relevant actions after it was damaged.

The probability of system loss due to accidental action (i) may be written as shown in Figure 18 and to demonstrate compliance with ALS requirements calculation of damage due to the accidental actions is needed. In general nonlinear analysis is required to estimate structural damage, i.e. permanent deformation, rupture etc. of structure components.

According to Moan (2007) the implied conditional annual probability of failure for a damaged structure designed to Norsok criteria will be of the order of 0.1. The probability of total loss implied by the ALS criterion for each category of accidental or abnormal action would then be of the order of 10^{-5} .

As a further consideration in Moan (2007) it is mentioned that hazards associated with normal variability and uncertainty inherent in prescribed payloads and environmental loads and resistance are handled by ULS and FLS design criteria. Such criteria do not reflect human errors and the notional annual failure probability of components implied by current ULS requirements for offshore structures is of the order $10^{-3} - 10^{-5}$. Fatigue and fracture are controlled by a combination of design for adequate fatigue life and robustness (ALS criterion) as well as by inspection and repair. If the fatigue design factor is taken to be 1, the fatigue failure probability in the service life is 0.1, but this value can be reduced significantly by using more restrictive design criteria and/or inspection.

15.3 Assessment of Structural Consequences of Accidents

15.3.1 Numerical and Simulation Tools

Numerical modelling can be carried out with the use of various methods and types of solvers including finite element, computational fluid dynamics, and hydrocodes.

Table 7: Tools for Determining Accidental Actions

Code	Type of Code	Uses for load and consequence determination
ANSYS FLUENT Chinook Cobalt Kameneon FireEx FLACS	CFD	Fluid flow physics models Air and Underwater blast analysis software Detailed flow field diagnostics HC Fire loads HC Explosion loads
MSC/DYTRAN Abaqus/Explicit LS-DYNA	Explicit FE	Extreme short duration events Nonlinear continuum, transient dynamic phenomena Thermal, ALE, fluid-structure interaction, multi-physics coupling
ALE3D	FV and FE	Heat conduction, multi-phase flow, chemical kinetics, species diffusion, detonation, deflagration, convective burn

Various numerical and simulation tools are available and should be selected based on the event being modelled. Commercially available numerical analysis codes which can be used to predict the load and structural response resulting from accidental events are given in Table 7.

In many cases hydrocodes are within government or defence organizations and are kept proprietary.

15.3.2 Experimental Methods

In some cases experimental programs are required to determine the structural integrity due to accidental events, and can be used to validate numerical approaches. Experimental programs are generally carried out to determine the effects of air and underwater blast, dropped objects, and fire as well as determining nonlinear material properties. Due to costs, structure availability, and environmental effects experimental programs may be limited to smaller scale. Even though the costs are high, large scale experiments are very useful for verifying structural behaviour under extreme actions.

16 RESIDUAL STRENGTH/STRUCTURAL INTEGRITY

In most design standards it is a requirement that an after damage situation following an accidental event or abnormal environmental action shall be considered, and that the structure in this condition shall remain intact for a period of time sufficient for all personnel to be safely evacuated and all process equipment to be closed down to avoid pollution.

16.1 Damage Tolerance

For the design of new structures, or assessment of existing structures not triggered by actual damage, damage tolerance considerations must be based on accidental limit states reflecting the relevant hazards. If a linear structural analysis of the damage scenario indicates sufficient capacity of all components it is often assumed that the hazard has not damaged the structure, i.e. the resistance is not degraded in relation to the after damage situation. However, in most cases the structure's resistance is more or less reduced as compared to its undamaged condition, and a reliable prediction of the extent of the damage requires the application of non-linear structural analysis methods.

Different design standards tend to specify slightly different acceptance criteria for the after damage situation. In lieu of more specific requirements ISO 19902 requires the after damage situation analysed using environmental conditions with a return period of the order twice a conservative estimate of the time required to perform suitable repairs by which the structure's strength would be restored to the design strength, the minimum return period shall be one year. The strength of damaged components shall either be estimated using a rational approach (according to ISO 19902) or shall be neglected, and the normal design requirements (using the usual action and resistance factors) for the design of new structures apply.

16.2 Damaged Structures

For existing structures where physical damage has been detected, the nature and extent of the actual structural damage must be established.

The analysis of the damaged structure determines any immediate requirement for shut-in and/or evacuation as well as the need for temporary repairs, while awaiting a decision and plan for the implementation of definite repairs or abandonment. Verifications of the after damage design situation for physically damaged structures are typically carried out in compliance with the design requirements for assessment of existing structures. In some design standards the assessment criteria and the design criteria for new structures are identical, while e.g. ISO 19902 potentially allows the use of relaxed acceptance criteria (Clause 24).

16.3 Mitigation and Repairs

As discussed in Chapter 15.3.2 accidents, human and operational errors are the most important causes to failures of offshore structures. It is therefore primarily important to avoid these errors in order to limit the risk of undesirable events. Secondly, it is crucial to carry out quality assurance and control in all life cycle phases.

In Moan (2007) the causes of failures are categorized and the corresponding measures to control the accident potential are listed. In general the measures include design criteria, quality assurance and control (QA/QC) relating to the engineering process, as well as the hardware and operational procedures.

17 MATERIAL MODELS FOR STRUCTURAL ANALYSIS

Offshore structures exposed to hazards as defined above may undergo highly non-linear structural deformations, including rupture. Therefore, finite element analyses of these events require the input of appropriate material relations including failure representing the local material behaviour. Depending on the hazard to be analysed and the materials found on the offshore structures a selection of recommended material models can be made, see Table 8. The physical origin of these material models will be briefly presented, followed by numerical implementation possibilities as well as comments, hints and shortcomings arising from the use of those models as well as concerns of guidelines and standards. However, hazard simulations utilizing the recommended material models and input parameters can be used for basic physical checks, but they may not be applicable in general.

The material modelling represents a crucial part of all numerical simulations, because it predefines how the material is assumed to behave during the simulations. Hence, the ability of the material model to represent the physical behaviour accurately directly influences the accuracy of the simulation results and their reliability. Furthermore,

Table 8: Recommended material models and associated hazards.

Hazard \ Material	Steel	Aluminium	Foam, Isolator, Rubber	Ice	Air	Water	Explosives	Risers, umbilical or power cables	Composite	Concrete	Seabed
Hydrocarbon explosions	■	■	■					■	■	■	
Hydrocarbon fires	■	■	■					■	■	■	
Underwater explosions	■	■	◆		■	■	■		◆	◆	◆
Wave Impact	■	■	■			■		◆	■		
Water-In-Deck	■	■	■			■			■		
Dropped Objects	■	■	■			◆		◆	■	■	■
Ship Impact	■	■	■	■				◆	■	■	◆
Earthquakes	■	■	■			◆		■	■	■	■
Ice, Iceberg	■	■		■		■		■	■	■	
Flooding	■	■	■		■	■			■		

■ - recommended, ◆ - recommended where applicable

the correct physical behaviour may be represented well by the underlying assumptions of the material model, because it can correspond well to the physical experiment done to obtain the properties of the material in question. However, whether or not this experiment or the correspondence represents the true material behaviour remains often a question, e.g. a classical tensile experiment is a material test by agreement even though a structural test is carried out. Hence, the utilization of such experimentally based material models using small structural tests can lead to inconsistent results when applied to general structures. Furthermore, it remains often questionable whether the obtained material model corresponds to the discrete mathematical model, i.e. the finite element mesh, of the structure to be analysed. Hence, a material model should be unique and usable for any mesh size or conditions and should therefore not affect the results with a change in discretization of the simulation domain. In the past, often the term ‘true’ material model was utilized, which is however misleading as it implies that it is ‘true’ by all means and could be universally applied. In fact, all material measures are ‘true’ with respect to their determination scale, i.e. the engineering measure obtained by a tensile experiment is true with respect to the specimens’ gauge length.

Hence, this chapter seeks to provide appropriate guidance to identify the material model to be used with the associated hazard according to Table 8 in such a way that it is consistent with the discretized, respectively meshed, simulation domain. Furthermore, engineering based best practices are provided as well as the associated shortcomings. The nomenclature of the numerical implementation used in the material input cards can be found in Hallquist (2007). The effects the material models account for, e.g. strain rate, temperature or damage criteria, will be provided alongside a selection of references relevant to the given material. Thereby, this database of material models will clarify common questions and uncertainties associated with the use of material models.

17.1 Guidelines and Standards

ISO 19902 Ed. 1 requires that the expected non-linear effects, including material yielding, buckling of structural components and pile failures, should be adequately modelled and captured. Strain rate effects should be considered as well as temperature dependency. NORSOK standard N-003 and DNV Recommended Practices DNV-RP-C204 suggest the use of the temperature dependent stress-strain relationships given in NS-ENV 1993 1-1, Part 1.2, Section 3.2. To account for the effect of residual stresses

and lateral distortions compressive members should be modelled with an initial, sinusoidal imperfection with given amplitudes for elastic-perfectly plastic material and elasto-plastic material models. General class rules or CSR commonly state that an appropriate material model should be used; possibly in the form of a standard power law based material relation for large deformation analysis of steel structures. Additionally, some specify critical strain values to be used independent of the mesh size, which should, however, be sufficient, may be specified.

Hence, these guidelines and standards fail to provide a clear guidance for the analyst and may easily lead to diverse results simply by choosing different, yet not necessarily physically correct, material parameters.

17.2 Material Model Database

17.2.1 Steel

Commonly, the nonlinear material behaviour is selected in the form of a power law; see, for example, Alsos *et al.* 2009 and Ehlers *et al.* (2008). The power law parameters can be obtained from standard tensile experiments; see Paik (2007). However, with this approach agreement between the numerical simulation and the tensile experiment can only be achieved by an iterative procedure for a selected element size chosen a priori. Hence, the procedure needs to be repeated if the element size is changed.

Furthermore, the determination of the material relation alone does not necessarily suffice, as the failure strain, i.e. the end point of the stress versus strain curve, depends in turn on the material relation. However, a significant amount of research has been conducted to describe criteria to determine the failure strain, for example by Törnqvist (2004), Scharrer *et al.* (2002), Alsos *et al.* (2008), and to present their applicability (e.g. Tabri *et al.* 2007 or Alsos *et al.* 2009). However, all of these papers use a standard or modified power law to describe the material behaviour, and none of these papers identifies a clear relation between the local strain and stress relation and the element length.

Relations to obtain an element length-dependent failure strain value are given by Peschmann (2001), Scharrer *et al.* (2002), Törnqvist (2004), Alsos *et al.* (2008) and Hogström *et al.* (2009). These curve-fitting relations, known as Barba's relations, are obtained on the basis of experimental measurements. However, they define only the end point of the standard or modified power law. Hence, Ehlers *et al.* (2008) conclude that the choice of an element length-dependent failure strain does not suffice in its present form.

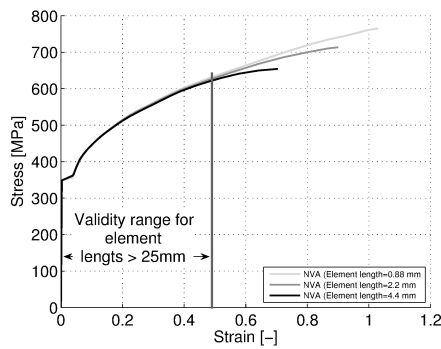
Therefore, Ehlers and Varsta (2009) and Ehlers (2009a) presented a novel procedure to obtain the strain and stress relation of the materials, including failure with respect to the choice of element size using optical measurements. They introduced the strain reference length, which is a function of the discrete pixel recordings from the optical measurements and corresponds to the finite element length. As a result, they present an element length dependent material relation for NVA grade steel including failure, see Figure 19.

Moreover, Ehlers *et al.* (2010) identified that a constant strain failure criterion suffices for crashworthiness simulations of ship structures and that the strain rate sensitivity of the failure strain and ultimate tensile force is less than three per cent, see Figure 20. Hence, for moderate displacement speeds the strain rate influence is negligible.

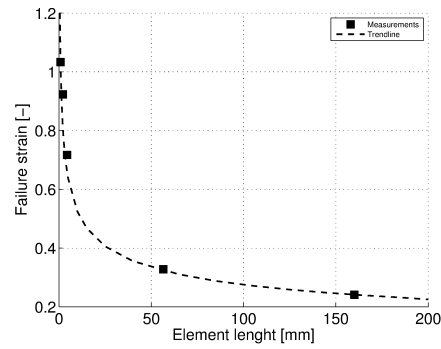
An example input card following the LS-DYNA nomenclature for a piece-wise linear material (mat.24) is given in Table 9.

Table 9: Piece-wise linear steel material model

*MAT_PIECEWISE_LINEAR_PLASTICITY								
\$#	mid	ro	e	pr	sigy	etan	fail	tdel
	1	7850.00	2.06E+11	0.3000	3.423E+8	0.000	0.661000	0.000
\$#	c	p	lcsc	lcsr	vp			
	0.000	0.000						
\$#	eps1	eps2	eps3	eps4	eps5	eps6	eps7	eps8
	0.006	0.02612	0.04019	0.06865	0.15071	0.345	0.64477	0.74
\$#	es1	es2	es3	es4	es5	es6	es7	es8
	3.423E+8	3.530E+8	3.731E+8	4.219E+8	4.901E+8	5.827E+8	6.621E+8	6.737E+8



(a)



(b)

Figure 19: NVA grade steel: measured local strain and stress relation (a) and failure strain (b) (Ehlers 2009b)

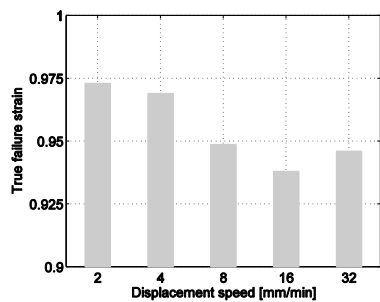


Figure 20: Deformation of witness plate from impact of a scaffold clip

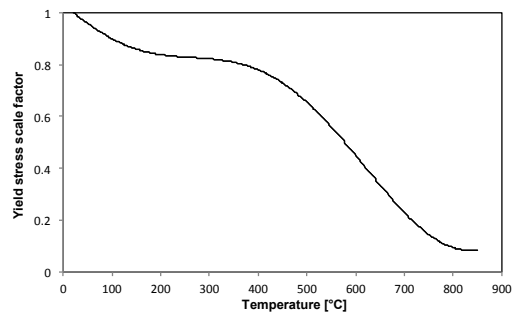


Figure 21: Global yield stress scale factor versus temperature for mild steel

However, the material behaviour, that is the change in the yield stress, at higher strain rates, $\bar{\epsilon}$, can be calculated according to the Cowper-Symonds relation

$$1 + \left(\frac{\bar{\epsilon}}{C} \right)^{1/p}$$

where C , p are the strain rate parameters and may be chosen as 40.4/sec and 5 for mild steel, respectively. Additionally, effects on elevated temperatures may be accounted for by scaling the global yield stress as a function of the temperature, see Figure 21.

The increase in yield- and ultimate strength at cryogenic temperatures, i.e. -100 and -163°C , is presented by Yoo *et al.* (2011) for mild stainless steel.

17.2.2 Aluminium

Various thin-walled aluminium structures under crash behaviour, i.e. large deformations including rupture, have been analysed experimentally and numerically in the past.

Langseth *et al.* (1998) uses an elasto-plastic material model with isotropic plasticity following the von Mises yield criterion and associated flow rule, see Berstad *et al.* (1994). Strain rate effects are often neglected for aluminium alloys, such as AA6060, in the strain rate range of 10^4 to 10^3 s^{-1} , see for example Lindholm *et al.* (1971). As a result, Langseth *et al.* are able to obtain good correspondence in terms of deformed shape, and shape of the force-displacement curve.

However, if high strain rates are to be expected, then the yield stress scaling according to Cowper-Symonds may be used. Négre *et al.* (2004) study the crack extension in aluminium welds using the Gurson-Tvergaard-Needleman (GTN) model and obtain reasonable correspondence in terms of force versus crack mouth opening displacement (CMOD). However, the GTN model requires a vast amount of input parameters whose physical origin cannot be directly provided. Furthermore, Négre *et al.* use 8-node brick elements, which are not suitable for large complex structures at present. Hence, from an engineering viewpoint this model does not suffice.

Lademo *et al.* (2005) utilize a coupled model of elasto-plasticity and ductile damage based on Lemaitre (1992) using the critical damage as an erosion criterion. They are able to simulate aluminium tensile experiments numerically with very good agreement using co-rotational shell elements and an anisotropic yield criterion Yld96 proposed by Barlat *et al.* (1997).

Such advanced material models can be easily implemented into numerical codes, and further increase in yield and ultimate strength at cryogenic temperatures, i.e. -100 and -163°C , can be considered following the results by Yoo *et al.* (2011) for mild aluminium. Furthermore, a strain reference length-based approach using optical measurements as proposed by Ehlers (2009a) for steel may be used to obtain a consistent material relationship. However, for most analyses a consistent determination of the global material behaviour, see Figure 22, together with a von Mises yield criterion will suffice.

An example input card following the LS-DYNA nomenclature for a piece-wise linear material (mat.24) is given in Table 10.

Table 10: Piece-wise linear aluminium material model

*MAT_PIECEWISE_LINEAR_PLASTICITY								
\$#	mid	ro	e	pr	sigy	etan	fail	tdel
	1	2.712E-9	75499	0.3000	200	0.000	0.1063	0.000
\$#	c	p	lcsc	lcsr	vp			
	0.000	0.000						
\$#	eps1	eps2	eps3	eps4	eps5	eps6	eps7	eps8
	4.940E-4	8.928E-4	0.002087	0.01000	0.03630	0.0796		
\$#	es1	es2	es3	es4	es5	es6	es7	es8
	220.7480	230.8739	241.2000	253.2500	270.3999	293.2200		

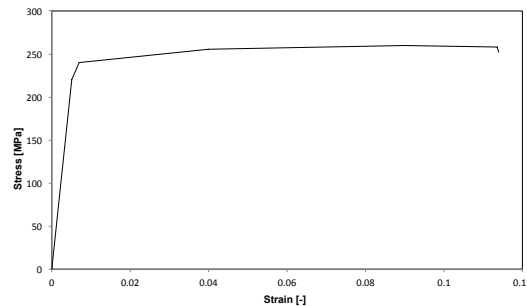


Figure 22: Example of a global strain versus stress curve from experiments

17.2.3 Foam, Isolator, Rubber

Gielen (2008) presents an isotropic polyvinyl chloride (PVC) foam model, which exhibits elasto-damage behaviour under tension and elasto-plastic behaviour under compression. His damage model is consistent with the physical behaviour of the foam, a full-scale application and verification is however missing.

Cui *et al.* (2009) present a model for uniform foam based on Schraad and Harlow (2006) for disordered cellular materials under uni-axial compression. As a result, they obtain various influencing parameters affecting the energy absorption capacity under impact. Hence, functionally graded foams may be used to increase impact resistance.

In the case of rubber, a simplified rubber/foam material model (mat.181) may be used, which is defined by a single uni-axial load curve or by a family of uni-axial curves at discrete strain rates, see Figure 23. An example input card following the LS-DYNA nomenclature for such rubber material is given in Table 11.

Table 11: Simplified rubber/foam material model

*MAT_SIMPLIFIED_RUBBER/FOAM									
\$#	mid	ro	k	mu	g	sigf	ref	prten	
	1	1.75E-9	1000	0	0	0	0	0	
\$#	sgl	sw	st	lc/tbid	tension	eps6	avgopt	pr/beta	
	80	50	15	1	0	0	0	0.495	

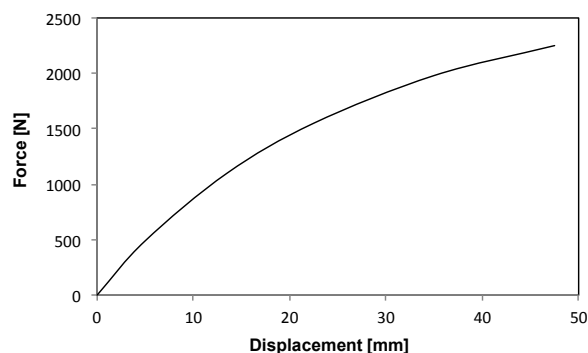


Figure 23: Exemplary force-displacement curve for rubber referenced as LC/TBID in mat.181

Table 12: Simplified ice material model

```
*MAT_ISOTROPIC_ELASTIC_FAILURE
$#      mid      ro      g      sigy      etan      bulk
      1 916.96000 3.0800E+9 7.6000E+5 6.8900E+9 8.0300E+9
$#      epf      prf      rem      trem
      0.010800 -3.080E+5      0.000      0.000
```

17.2.4 Ice

One of the main difficulties when modelling ice is the prediction of ice failure, i.e. fracture, under loading at temperatures around the melting point of the ice. Thus the local ice-structure interaction includes transitions between the different phases. The failure process of ice begins when the edge of the moving ice hits the structure. This contact induces loads to the edge of the ice causing a stress state in the ice. When the stresses exceed the strength of ice, it fails. Ice becomes ductile with visco-elastic deformations during low loading rates and brittle during high loading rates.

Polojärvi and Tuhkuri (2009) developed specialized simulations tools utilizing the boundary element method, whereas Forsberg *et al.* (2010) utilize the cohesive element method (CEM) to model ice failure. The latter is however of highly stochastic, or even random, nature and eventually results in reasonable agreement if experimental validation data becomes available.

However, Liu *et al.* (2010) treat the ice in a coupled dynamic ship – iceberg collision as an isotropic material, see Riska (1987), using the well-known Tsai-Wu strength criterion, see Tsai (1971). As a result, the obtained numerical results give an indication of the structural damage of the ship structure. However, their model erodes the ice at failure in an unphysical fashion resulting in purely numerical pressure fluctuation in the contact surface.

Therefore, the underlying material models and ice properties are in need to be defined consistently to account for the possible scatter and thereby to result in reliable design methods for ships and offshore structures. Hence, unless material model data is not available explicitly for tension and compression including an appropriate failure criterion for brittle ice failure based on micro-crack growth, a simple elastic model may be employed. The latter is however only valid to some extent, if, e.g. the flexural strength of an ice sheet is of interest.

Therefore, as a first attempt, ice may be modelled as a volumetric body following non-iterative plasticity with a simple plastic strain failure model (mat_13). However, therein the yield- and failure stress is note rate or pressure dependent and the temperature is assumed constant. An example input card following the LS-DYNA nomenclature for Baltic Sea ice is given in Table 12.

17.2.5 Air

For numerical simulations of structures subjected to underwater explosions, where the target is air-backed, the air needs to be modelled. The main material parameters are the mass density and the equation of state (EOS). The latter can be expressed

Table 13: Air material model

```
*MAT_NULL (n, kg)
$#      mid      ro      pc      mu      terod      cerod      ym      pr
      1      1.280      0.000      0.000      0.000      0.000      0.000      0.000
```

Table 14: Linear polynomial equation of state for air

```
*EOS_LINEAR_POLYNOMIAL (cm, g)
$# eosid      c0      c1      c2      c3      c4      c5      c6
      1      0.000  1.0e-02  0.000  0.000  0.400  0.400  0.000
$#      e0      v0
      0.000  0.000
*END
```

Table 15: Ideal gas equation of state for air

```
*EOS_IDEAL_GAS
$# eosid      cv0      cp0      c1      cq      t0      v0
      1  718.0000  1005.000  0.000  0.000  270.00  1.000
```

as a linear polynomial defining the pressure in the gas as a linear relationship with the internal energy per initial volume. The ideal gas EOS is an alternative approach to the linear polynomial EOS with a slightly improved energy accounting algorithm. In most cases, the mass density is the only parameter defined for the air. The same material properties were used in Trevino (2000) and Webster (2007).

An example input card for air following the LS-DYNA nomenclature is given in Table 13 according to Webster (2007).

The EOS example input following the LS-DYNA nomenclature is given in Table 14 according to Webster (2007) in the most common form, which defines the parameters such that it is an ideal gas behaviour.

Do (2009) describes the calculation process of e_0 , which can be used to define an initial pressure within the air. Additionally, an example input card for the ideal gas EOS following the LS-DYNA nomenclature is given in Table 15 according to Marc Ltd. (2007).

The ideal gas EOS is the equivalent of the linear polynomial with the C_4 and C_5 constants set to a value of $(\gamma - 1)$.

17.2.6 Water

When conducting simulations of structures subjected to underwater explosions, water models are required.

The primary mechanical property to be defined is the mass density and in some cases the pressure cut-off and dynamic viscosity coefficient is needed. The cut-off pressure is defined to allow the material to numerically cavitate when under tensile loading. This is usually defined as a very small negative number, which allows the material to cavitate once the pressure goes below this value.

Table 16: Material model for water (Trevino, 2000)

```
*MAT_NULL (cm, g)
$# mid      ro      pc      mu      terod      cerod      ym      pr
      1  1.000000  0.000  0.000  0.000  0.000  0.000  0.000
```

Table 17: Material model for water (Webster, 2007)

```
*MAT_NULL (m, kg)
$# mid      ro      pc      mu      terod      cerod      ym      pr
      1  1025.000 -1.0e-20  1.13e-3  0.000  0.000  0.000  0.000
```

Table 18: Equation of state for water

```
*EOS_GRUNEISEN
$# eosid      c      s1      s2      s3      gamma0      a      e0
    1 2417.000 1.410000 0.000 1.000 0.000 0.000 0.000
$#      v0
    1.000
```

Additionally, the equation of state (EOS) needs to be defined, most commonly as a Gruneisen EOS with cubic shock-velocity-particle velocity defining the pressure for compressed materials. The constants in the Gruneisen EOS are found from the shock wave velocity versus particle velocity curve. Two example input cards following the LS-DYNA nomenclature for water (mat_009) are given according to Trevino (2000) and Webster (2007) in Table 16 and Table 17, respectively.

Additionally, Gruneisen EOS is the most commonly used EOS for defining the water behaviour with underwater explosion events. An example input card following the LS-DYNA nomenclature is given in Table 18 according to Webster (2007).

17.2.7 Explosives

An explosive material requires two keywords to define the behaviour of the material. These include the material keyword and the equation of state (EOS). The mechanical properties to be considered are the mass density, the detonation velocity in the explosive and the Chapman-Jouguet pressure. Furthermore, the bulk modulus, shear modulus and yield stress may be required depending on the model.

For the EOS, there are three possibilities to define the pressure for the detonation products. All of these EOS define the pressure as a function of the relative volume and the internal energy per initial volume. The most commonly used EOS for explosive behaviour is the standard Jones-Wilkins-Lee (JWL). This EOS was modified by Baker (1997) and has the added feature of better describing the high-pressure region above the Chapman-Jouguet state.

In addition to the material and EOS definitions in LS-DYNA, the INITIAL_DETONATION keyword is required to define the position and time of the initiation of the detonation process. This is the point at which the detonation initiates and the time for the remaining explosive to detonate is determined by the distance to the centre of the element divided by the detonation velocity. In the material definition for MAT_HIGH_EXPLOSIVE_BURN (mat_008) the value of BETA determines the type of detonation. If beta burn is used, any compression of the explosive material will cause detonation. For programmed burn, the explosive material can act as an elastic perfectly plastic material through the definition of the bulk modulus; shear

Table 19: Explosive material model

```
*MAT_HIGH_EXPLOSIVE_BURN
$# mid      ro      d      pcj      beta      k      g      sigy
    1 1630.000 6930.00 2.1e10 2.000 0.500 0.000 0.000
```

Table 20: Equation of state for the explosive material model

```
*EOS_JWL
$# eosid      a      b      r1      r2      omeg      e0      vo
    1 3.71e11 3.23e9 4.15 0.950 0.300 7.0e9 1.000
```

modulus, and the yield stress. In this case, the explosive must be detonated with the INITIAL_DETONATION keyword.

An example input card following the LS-DYNA nomenclature for TNT (mat_008) is given in Table 19 according to Webster (2007).

Furthermore, the most commonly used Jones-Wilkens-Lee EOS is given in Table 20 according to the LS-DYNA nomenclature (Webster, 2007).

17.2.8 Risers, Umbilical or Power Cable

What all these structures have in common is the fact that they are typically very long, therefore slender. Their global mechanical properties to be defined are the bending-, torsional- and axial stiffness. Furthermore, the main aspect to be covered when modelling such structures is their stiffness dependency with respect to tension, torsion and curvature, i.e. stick-slip effects.

Therefore, experimental measurements of the global and local behaviour as well as a local analysis of the cross-section are needed. Typical numerical implementations would utilize elasto-plastic and visco-elastic material models considering friction, contact formulation (lift-off) as well as torsion/rolling effects on pipes.

Sævik (2011) studied the local behaviour of stresses in flexible pipes with a detailed model considering the cross-section build-up. However, for global analysis of an offshore structure, where the support effect of the slender structure is of interest, a simpler discretisation using beam elements with local stiffness properties can be used, see Rustad *et al.* (2008).

For a typical 8" flexible riser the following global parameters can be found: $EI = 200 \text{ kNm}^2$, $EA = 7.7 \cdot 10^8 \text{ N}$, $GI_t = 5.9 \cdot 10^6 \text{ Nm}^2$.

An example input card following the LS-DYNA nomenclature for a visco-elastic material (mat_117) is given in Table 21.

Table 21: Visco-elastic riser material model

```
*MAT_VISCOELASTIC
$#      mid      ro      bulk      g0      gi      beta
      1 8650.000  2.06e11  0.8e11  0.1e11  0.200
*END
```

17.2.9 Composites

Composite materials can be of various types, such as classical fibre-reinforced plastics or various stacks of materials, i.e. sandwich like structures. Therefore, their material parameters are very specific to the exact type of composite found in the offshore structure.

Menna *et al.* (2011) simulate impact tests of GFRP composite laminates using shells and provide the material parameters for a Mat Composite Failure Option Model (mat_059) of LS-DYNA. Feraboli *et al.* (2011) present an enhanced composite material with damage (mat_054) for orthotropic composite tape laminates together with a series of material parameters.

Most orthotropic elastic materials can be described until failure according to:

$$[C] \{\sigma\} = \{\epsilon\}$$

Table 22: Composite material model

```
*MAT_COMPOSITE_MATRIX
$#      mid      ro
      2 7850.0000
$#      c11      c12      c22      c13      c23      c33      c14      c24
      2.8409E+9 3.3956E+8 1.1319E+9      0.000      0.000 3.9615E+8 7.4958E+7 2.3769E+7
$#      c34      c44      c15      c25      c35      c45      c55      c16
      0.000 8.3506E+6 2.3769E+7 7.9231E+7      0.000 1.6645E+6 5.5485E+6      0.000
$#      c26      c36      c46      c56      c66      aopt
      0.000 2.7731E+7      0.000      0.000 1.9420E+6      0.000
$#      xp      yp      zp      a1      a2      a3
      0.000      0.000      0.000      0.000      0.000      0.000
$#      v1      v2      v3      d1      d2      d3      beta
      0.000      0.000      0.000      0.000      0.000      0.000      0.000
```

where C is the compliance matrix besides the six stress and strain components. Hence, the compliance matrix can be composed of the extensional stiffness coefficients, the extensional-bending stiffness coefficients and the bending stiffness coefficients.

An example input card following the LS-DYNA nomenclature for a composite matrix material (mat.117) using such compliance matrix formulation is given in Table 22 for an equivalent stiffened plate.

17.2.10 Concrete

Concrete material requires two keywords to define the behaviour of the material. These include the material keyword and the equation of state (EOS). The mechanical properties to be considered are the mass density, the shear modulus and an appropriate measure of the damage, respectively softening. The EOS describes the relation between the hydrostatic pressure and volume in the loading and unloading process of the concrete uncoupled from the deviatoric response. These parameters are typically obtained by experimental testing of the concrete under different loading directions and rates. Thus, the damage includes strain-rate effects.

Markovich *et al.* (2011) present a calibration model for a concrete damage model using EOS for tabulated compaction and a concrete damage, release 3, model (mat.72r3) and provide the required input parameters. Tai and Tang (2006) studied the dynamic behaviour of reinforced plates under normal impact using the Johnson-Holmquist Concrete equivalent strength model with damage and an EOS, which requires less input parameters and allows for easier implementation with good accuracy.

An example input card following the LS-DYNA nomenclature for concrete material (mat.111) is given in Table 23 according to Tai and Tang (2006).

Table 23: Concrete material model

```
*MAT_JOHNSON_HOLMQUIST_CONCRETE
$#      mid      ro      g      a      b      c      n      fc
      1 2240.000 13.467e11      0.750      1.650      0.007      0.760      48.00
$#      t      eps0      efmin      sfmax      pc      uc      pl      ul
      0.000      1.000      0.010      11.700      13.60      0.00050      1.050      0.100
$#      d1      d2      k1      k2      k3      fs
      0.030      1.000      17.40      38.80      29.80      0.000
```


Table 24: Soil material model

*MAT_MOHR_COULOMB								
\$#	mid	ro	gmod	rnu	phi	cval	psi	
\$#	1	1834.862	5.0e06	0.300	0	0.523	5.0e03	0.000
\$#	nplanes			lccpdr	lccpt	lccjdr	lccjt	lcsfac
\$#	0	0	0	0	0	0	0	0
\$#	gmoddp	gmodgr	lcmep	lcphiep	lcpsiep	lcmst	cvalgr	aniso
\$#	0.000	0.000	0.000	0.000	0.000	0.000	0.000	1.000000
\$#	dip	dipang	cplane	frplane	tplane	shrmax	local	
\$#	0.000	0.000	0.000	0.000	0.000	1.00E+20	0.000	

17.2.11 Soil

For some simulations of hazard the seabed has to be included. However, the material parameters for seabed, respectively soil, are fairly location dependent and may vary significantly within close proximities. Therefore, it is of utmost importance to obtain experimental data for the site in question.

Typically those experiments should identify the soil stiffness in different directions, the friction, the break out resistance and a cycling behaviour (trenching). Henke (2011) presents numerical and experimental results for Niederfelder sand and uses a hypoplastic constitutive model, assuming cohesionless linear elastic behaviour, to achieve good correspondence. Vermeer and Jassmin (2011) use a SPH approach with an elastic-plastic Mohr-Coulomb model to simulate drop anchors and present the utilized material parameters. Furthermore, solid elements can be used to represent sandy soils or granular materials following the Mohr-Coulomb behaviour.

An example input card following the LS-DYNA nomenclature for a Mohr-Coulomb material (mat_173) is given in Table 24 according to the material parameters from Vermeer and Jassmin (2011).

18 BENCHMARK STUDY: RESPONSE OF STIFFENED PANEL SUBJECTED TO HYDROCARBON EXPLOSION LOADS

18.1 Scope of Work

The objective of the Benchmark Study is to compare procedures and the strength assessment results of stiffened steel panels subjected to hydrocarbon explosion loads performed by the members of Committee V.1. The capabilities of modern software to simulate such complex loads and responses are also to be evaluated. Structural response of stiffened steel panels subjected to explosion loads is analysed and compared in particular with respect to:

1. Time-displacement profile at the centre of each panel.
2. Residual deflections at 25 locations over the panel surface.

The benchmark is based on a full scale test experiments carried out at the Spadeadam test site, UK, Figure 24.

Input data regarding geometry of tested panels and results of the tests are obtained by courtesy of The Steel Construction Institute, UK, (SCI, 1998).



Figure 24: Spadeadam test site, offshore module and location of the panel during the test.

The following committee members contributed to the benchmark:

Participation	Affiliation	Analysis software	Reference on Figures
J. Czujko	Nowatec, Norway	LS-Dyna	Nowatec
Wen-Yong Tang	Shanghai Jiao Tong University, China	Abaqus, Dytran	SJTU
M. Riley	Defence R&D (DRDC) Canada	LS-Dyna	DRDC
S. Ehlers	Aalto University, Finland ¹	LS-Dyna	AU

¹Currently, NTNU, Trondheim, Norway

18.2 Benchmark Model, Geometry

For this benchmark a stiffened panel (Panel 1 from the test) is selected. Geometry of the panel is presented in Figure 25.

18.3 Material data

Material properties derived from coupon tests for panel no.1 are presented in Table 25.

18.4 Loads

A panel loading is provided in the form of idealized representations of the pressure time profiles. For each pressure transducer, the idealized load pulse rise time and duration was calculated. Figure 26 presents representative rise time (T1) and duration (T2) of load pulse.

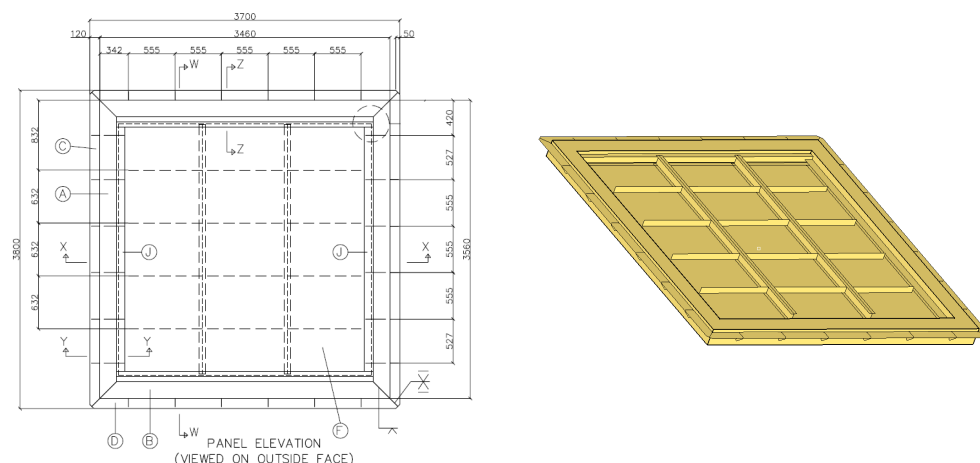


Figure 25: Stiffened panel. Geometry used in the benchmark study.

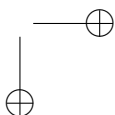


Table 25: Material properties of the panel.

		Flat Stiffener (75 × 6)	RSJ Stiffener	Plates
Young modulus	<i>MPa</i>	210000	210000	210000
Poisson ratio	-	0.3	0.3	0.3
Yield stress	<i>MPa</i>	270	300	305
Ultimate tensile stress	<i>MPa</i>	477	460	490
Elongation	%	29.9	27.5	28.8
Density	<i>t/mm³</i>	7.85E-009	7.85E-009	7.85E-009

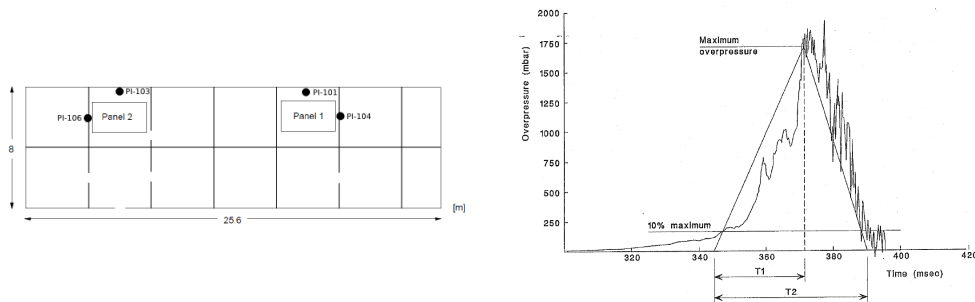


Figure 26: Location of the panel in the test rig and interpretation of the blast overpressure.

The pressure and duration information for each pressure transducer is summarised in Table 26.

18.5 Monitoring of Results

The results of the Benchmark represent transient dynamic response of the test panel and damage of the panel in 25 predefined points, Figure 27.

18.6 Benchmark Procedure

Benchmark study has been carried out in two phases:

1. *Phase 1* where all model development including geometry, boundary conditions, materials and loads was based on individual participants' interpretation of input data from the test.
2. *Phase 2* where assumptions regarding explosion loads were agreed upon between participants of the benchmark study.

In addition, parameter studies involving modelling assumptions regarding representation of geometry, FE mesh density, strain rate effects and application of explosion overpressure have been carried out.

Table 26: Pressure and duration data.

Pressure Transducer ID	Coordinate [m]			Maximum Overpressure [mbar]	Maximum Overpressure > 1 ms duration [mbar]	Time of arrival [ms]	Idealised Profile Representation	
	X	Y	Z				Rise Time [ms]	Duration [ms]
PI-101	18	7.5	7.9	1320	1034	544.3	77.2	110.0
PI-104	20	7.9	6.1	910	792	523.7	51.9	100.7

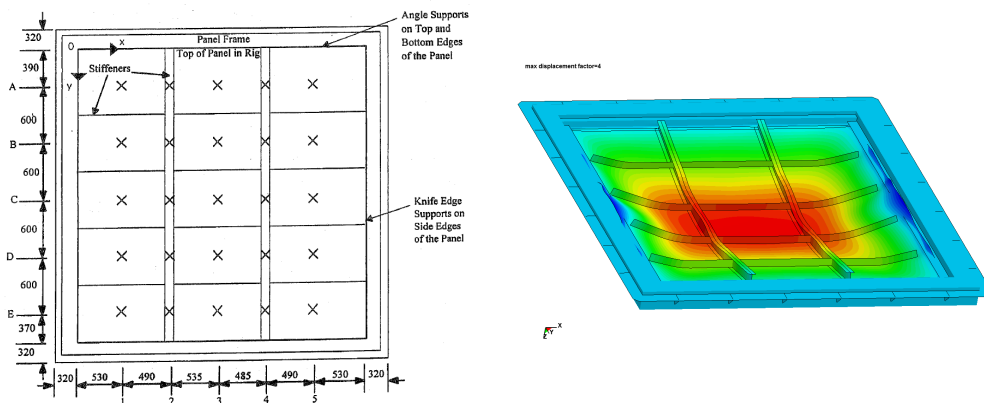


Figure 27: Location of the monitoring points for damage control and final deformation of the panel.

18.7 Phase 1 – Modelling Assumptions and Results

18.7.1 Modelling Assumptions

The following modelling assumptions have been considered:

- Geometry and boundary conditions
- Material properties
- Overpressure magnitude and profile based on input data supplied

Table 27 summarizes the modelling approach of all participants in the benchmark study.

18.7.2 Summary of Results

Transient response

Time response data prepared by modellers is presented in Table 28. Results from AU and SU (ABAQUS) represent upper bound of results. In turn results from Nowatec,

Table 27: Phase 1. Modelling approach for benchmark study.

Panel no.1	Modeller	SU	SU	DRDC	AU
	Nowatec (LS-DYNA)	(DYTRAN)	(ABAQUS)	(LS-DYNA)	(LS-DYNA)
Geometry	full panel	full panel	full panel	full panel	quarter of panel
	with outer frame	without outer frame	without outer frame	with outer frame	with outer frame
BC (knife edge)	with separation	no separation	no separation	with separation	with separation
Material (strain effects)	evaluated	evaluated	evaluated	evaluated	evaluated
Loads	average of P-101 (1 ms) and P-104 (1 ms)	average of P-101 (1 ms) and P-104 (1 ms)	average of P-101 (1 ms) and P-104 (1 ms)	3 zones of pressure	average of P-101 (max) and P-104 (max)
	$P = 913 \text{ mbar}$	$P = 913 \text{ mbar}$	$P = 913 \text{ mbar}$	$B = 910 \text{ mbar}$, 51.8 ms, 100.7 ms	$P = 1115 \text{ mbar}$
	$T1 = 64.5 \text{ ms}$	$T1 = 64.5 \text{ ms}$	$T1 = 64.5 \text{ ms}$	$E = 1183 \text{ mbar}$, 68.8 ms, 107.0 ms	$T1 = 64.5 \text{ ms}$
	$T2 = 105.35 \text{ ms}$	$T2 = 105.35 \text{ ms}$	$T2 = 105.35 \text{ ms}$	$F = 636 \text{ mbar}$, 34.9 ms, 94.5 ms	$T2 = 105.35 \text{ ms}$

Table 28: Summary of max transient deflection predictions as a ratio of observed maximum deflection from test.

	Nowatec	SU (DYTRAN)	SU (ABAQUS)	DRDC	AU
Panel no.1	0.83	0.85	1.05	0.66	1.09

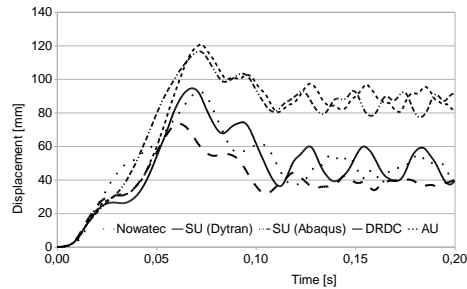


Figure 28: Summary of max transient deflection.

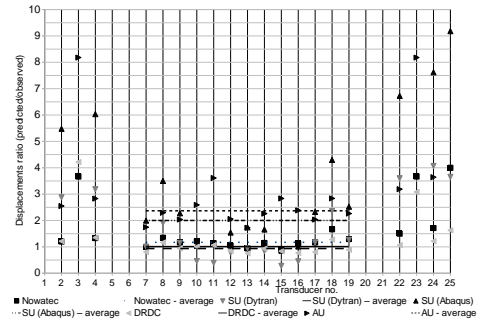


Figure 29: Comparison of residual deflections.

Table 29: Summary of residual deflection predictions as an average ratio of predicted residual deflection vs. measurements.

	Nowatec LS-DYNA	SJTU (DYTRAN)	SJTU (ABAQUS)	DRDC (average) LS-DYNA	AU (average) LS-DYNA
Panel no.1	1.13	1.00	1.99	0.93	2.36

DRDC and SU (DYTRAN) represent lower bound results. Transient response of the panel for Phase 1 is given in Figure 28.

Residual deflections

Residual deflections are presented in Figure 29. All modellers obtained deflections comparable to experiment in measuring points from 7 to 19 that lie in the centre of the panel. All modellers, excluding DRDC, failed to predict deflections in the panel's corners that are close to experimental results.

An average of displacements ratio was calculated to compare predictions between modellers. Results are presented in Table 29.

The closest predictions were obtained by Nowatec, SJTU (DYTRAN) and DRDC. Models analysed by Aalto University and Shanghai Jiao Tong University in ABAQUS over-predicted the residual deflections.

18.8 Phase 2 – Modelling assumptions and results

18.8.1 Unified explosion overpressure

In order to unify modelling of explosion overpressure it has been agreed to repeat benchmark study with overpressure obtained from transducer PI-04 with maximum overpressure 792 mbar.

18.8.2 Summary of results

Transient response

Time response data prepared by modellers are illustrated in Figure 30 and summarised in Table 30. Results compared represent a case where dumping and friction are not

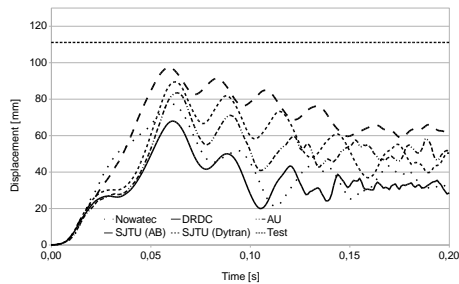


Figure 30: Summary of max transient deflections. Strain rate included.

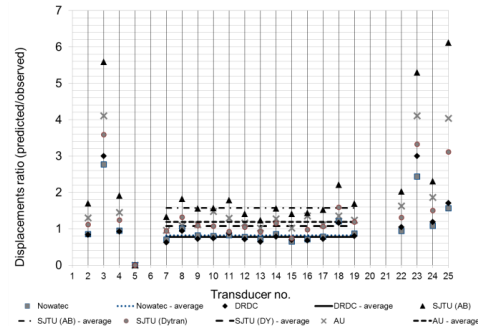


Figure 31: Comparison of residual deflections. Strain rate included.

Table 30: Summary of max transient deflection predictions as a ratio of observed maximum deflection from test.

	Nowatec	SJTU (DYTRAN)	SJTU (ABAQUS)	DRDC	AU
Panel no.1 No strain rate	0.91	1.13	0.93	0.92	0.79
Panel no.1 Strain rate	0.70	0.81	0.88	0.61	0.75

Table 31: Summary of residual deflection predictions as an average ratio of predicted residual deflection vs. measurements.

	Nowatec	SJTU (DYTRAN)	SJTU (ABAQUS)	DRDC	AU
Panel no.1 No strain rate	1.70	2.68	1.64	1.88	1.29
Panel no.1 Strain rate	0.82	1.07	1.57	0.77	1.19

accounted for. Results from SJTU (both DYTRAN and ABAQUS) represent upper bound of results. In turn results from Nowatec and DRDC represent lower bound results. Results from AU give a slightly unusual conservative prediction.

Residual deflections

Residual deflections are presented in Figure 31. All modellers obtained deflections comparable to experiment in measuring points from 7 to 19 that lies in the centre of the panel. All modellers, excluding DRDC, failed to predict deflections in the panel's corners that are close to experimental results.

An average of displacements ratio was calculated to compare predictions between modellers. Results are presented in Table 31.

The best predictions were obtained by AU, SJTU (ABAQUS) and Nowatec. Models analysed by Shanghai Jiao Tong University in DYTRAN over-predict the residual deflections. Further the strain rate dependency does not seem to be considered to the same extent by ABAQUS when compared to DYTRAN and LS-DYNA.

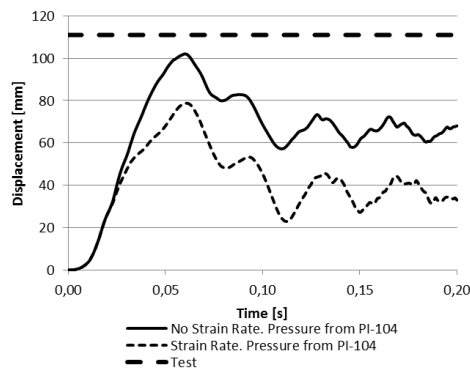


Figure 32: Effects of strain rate obtained for different models.

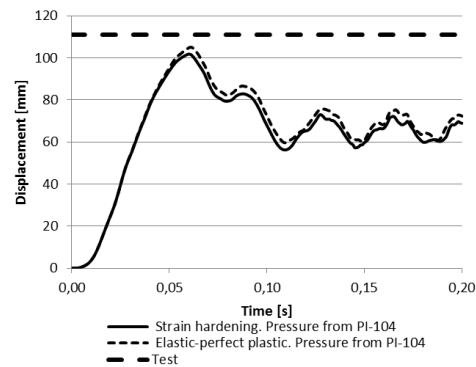


Figure 33: Effects of different material models on panel response.

18.9 Parameter Study

18.9.1 Effects of Strain Rate and Material Models Applied

Different material models: elastic-perfectly plastic and elastic-plastic with hardening, influence the maximum and residual deflections. Strain rate effects have been implemented using Cowper-Symonds equation with $D = 40$ and $P = 5$. Effects of strain rate and different material models applied are illustrated in Figure 32 and Figure 33.

18.10 Conclusion from the Benchmark study

The presented benchmark study consists of a relatively simple structural arrangement, i.e. a stiffened panel supported by a frame, subjected to a hydrocarbon explosion load. However, the study proved to be sufficiently complex to cause significant scatter in results when analysed by a group of experts. This scatter is attributed to the underlying simulation assumptions made by the analysts. These results provide invaluable insight into the variability in predictions when different values are used for influential parameters, one of which is the analysts themselves.

In the first phase the analysts were provided some model details and left to make assumptions which they saw fit. This phase unveiled the influence of the individual approximations including the assumed pressure loading, geometric discretization, and boundary conditions. It was found that ABAQUS and DYNA were able to predict the transient deflection with good accuracy, both for the full panel with and without the outer frame and knife-edge support modelled. However, for the residual deflection neglecting the frame and support or simplifying the applied pressure as the average of the measured pressures caused significant deviation from the full-scale measurements. On the contrary, the lower pressure assumption causes an under-prediction of the transient deflection, but could lead to accurate residual deflections. Furthermore, it is worthwhile to note that only by modelling the asymmetry in the pressure load, the panels' corner deflections can be captured accurately. In experiments the steel material shows a large reduction in deflection from the peak transient value to the residual deflection, which is not accurately described in the numerical material models. Hence, it was found that it is possible to predict either the transient or the residual deflection accurately, but not both with a single simplified model. A detailed strain rate dependent material test and modelling series would bring more light into this phenomenon in the future.

Phase two of this study which defined the load, material properties, and system characteristics (i.e. damping and friction) significantly reduced the variation in the different analysts' results, except for the quarter model, which was overly stiff. The exclusion of strain rate dependency provided poor results compared to experimental measurements and confirms the significant rate dependency of the panel materials. This would have to be known in order to provide more accurate predictions compared to experiments. Additionally, a global geometric model considering the actual supports as well as a more accurate load distribution compared to the experiments would be favourable.

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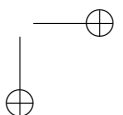
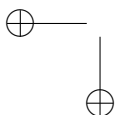
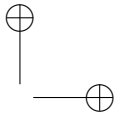
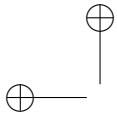
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18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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18th INTERNATIONAL SHIP AND
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09-13 SEPTEMBER 2012
ROSTOCK, GERMANY
VOLUME 2



COMMITTEE V.2
**NATURAL GAS STORAGE AND
TRANSPORTATION**

COMMITTEE MANDATE

Concern for the safety and design of containment systems for the storage and transportation of natural gas in connection with floating platforms and terminals, and onboard ships. This is to include assessing the performance of various containment systems for gas under compression (CNG), liquefaction under cooling (LNG), and combinations of the two methods. Particular attention shall be given to the integrity and safety aspects of containment systems under pressure and thermal loads, and the interaction between fluid and structure under static and dynamic conditions. Needs for revision of current codes and regulations shall be addressed.

COMMITTEE MEMBERS

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Hannis Bogaert
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Wha S. Kim
Magnus Lindgren
Eric Martin
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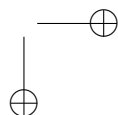
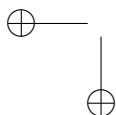
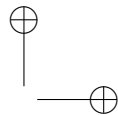
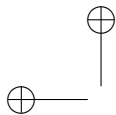
KEYWORDS

Cargo Containment Systems, Liquefied Natural Gas Carrier, Floating Liquefied Natural Gas, Floating Storage and Regasification Unit, membrane tank, spherical tank, prismatic tank, Compressed Natural Gas, sloshing, offshore terminal, arctic, structural integrity, collision, flooding, fatigue, vibration, fire safety, corrosion, Boil off Gas, cryogenic spillage, fuel Liquefied Natural Gas.

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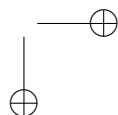
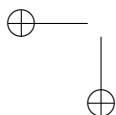
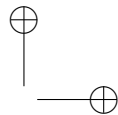
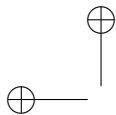
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1 INTRODUCTION

The Committee V.2 is a new specialist committee, the mission of which is to outline the safety and design aspects of containment systems used for natural gas storage and transportation on the ocean. With the increase in the worldwide demand for natural gas as a relatively clean energy source compared to other fossil fuels, new concepts and technologies related to the storage and transportation of natural gas have been emerging recently. Based on the committee's mandate and the specialities of its members, the Committee has reviewed the performance of existing and new containment systems and has discussed their safety.

The initial section of the report describes the safety records, transportation and market trends as the background of the Committee's work. Next, the safety aspects of LNG are discussed and an overview of Cargo Containment Systems (CCSs) and operational features related to the safety and design of natural gas storage and transportation systems are described.

The following chapter deals with the measures that must be taken to assure the safety of the Cargo Containment System per mode of failure, including brief summaries of the phenomena. At the beginning of the chapter, structural integrity management is outlined. Possible failure modes caused by several incidents such as sloshing, collision, fatigue, and the like, together with the measures to mitigate them, are discussed.

The necessity of establishing new rules and regulations is emphasized with regard to the new concepts of natural gas storage and transportation, for example in applications such as Floating Liquefied Natural Gas (FLNG), Arctic and for applications of LNG as fuel.

2 BACKGROUND

2.1 LNG Transportation Safety Records

Over the LNG industry's 60-year history of 40,000 voyages, there has not been recorded a spill from a ship into the water from either a collision or grounding (Ostvik *et al.*, 2005 and Foss, 2006). During the period from 1964 to 2008 (44 years) with over 30,000 shiploads of LNG delivered and more than 100 million miles travelled in the loaded condition, the overall safety record of LNG carriers (LNGCs) has been remarkably good with no fatalities, no record of fire occurring in the deck or in the cargo area or cargo tanks of any LNG ship (CH-IV International, 2009). Typical incidents include failure of containment tanks, tank cover and deck fractures due to LNG released, valve leakage, rollover incidents, tank overfilled, broken moorings, hull fatigue cracks, collision, and other incidents.

Statistical data showing the safety of LNG transportation today in comparison with general ship transportation is shown in Figure 1 (Data developed based on Lloyds World Fleet Statistics). The statistics show that the safety records of LNG transportation is better than the safety records for general ship transportation. It can be assumed that the good records are due to several reasons: LNG ships have traditionally been built and maintained to high standards. Well trained personnel onboard with a relatively low turnover. Operation of LNG ships has been based on trading on fixed routes on long term contracts. The LNG fleet has also been growing slowly and steadily with about 4 ships per year from 1970 to 2000.

From about year 2000 the orders of LNG shipping increased drastically to about 25 ships per year until 2010. The increase in both size and number (Figure 2 and

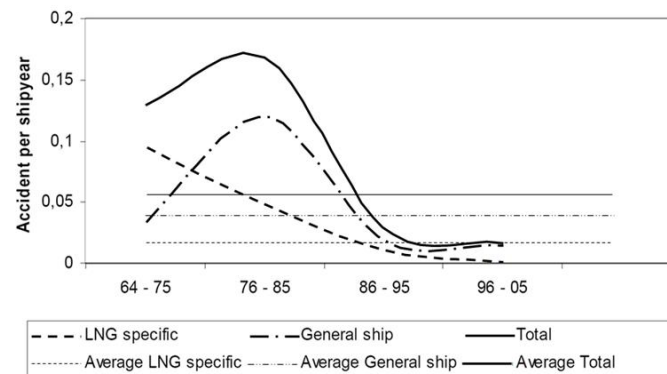


Figure 1: Safety records for LNG ships compared to general ship transportation

Figure 3) may have an impact on the safety records. It will be of great interest to study what impact the large orders of ships and the corresponding shortage of qualified and experienced personnel may have on the statistics of safety. Larger ships (Figure 2) and with some new operation profiles together with loading and offloading at more exposed locations with more harsh environmental conditions may also have an impact on the safety. LNG handled in offshore applications give new challenges to the safety aspects of LNG transportation. Another new application is the use of LNG as fuel for all type of ships. Here LNG will be handled by a wide range of operators. New safety aspects and ways to handle LNG from a safety perspective are required.

2.2 LNG Market and Trends

Figure 4 shows the current and future worldwide LNG trades. At the moment, main consumers of LNG are East Asian countries (Japan, Korea, Taiwan, etc.), and some European countries (Spain, France, etc.). China and India are expected to increase LNG import in the near future.

Figure 3 shows the historically accumulated number of LNG carriers with different cargo containment systems. As illustrated in this figure, LNG transportation continues to expand, and in particular the number of completed LNG carriers has dramatically increased since the year 2000. This increase in LNG ship deliveries was driven largely by the massive expansion of LNG production in Qatar. This trend is unlikely to continue at the pace of the past decade and many factors such as the exploitations of

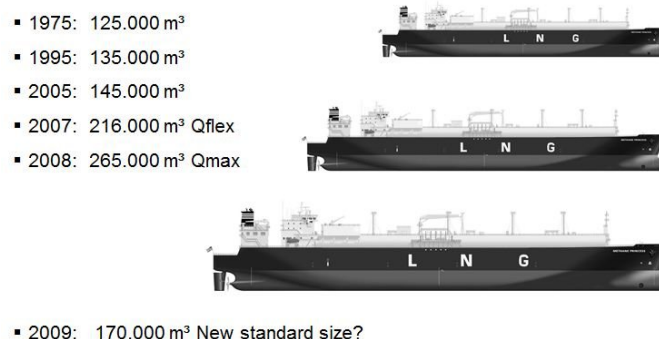


Figure 2: The development of standard LNG ship size

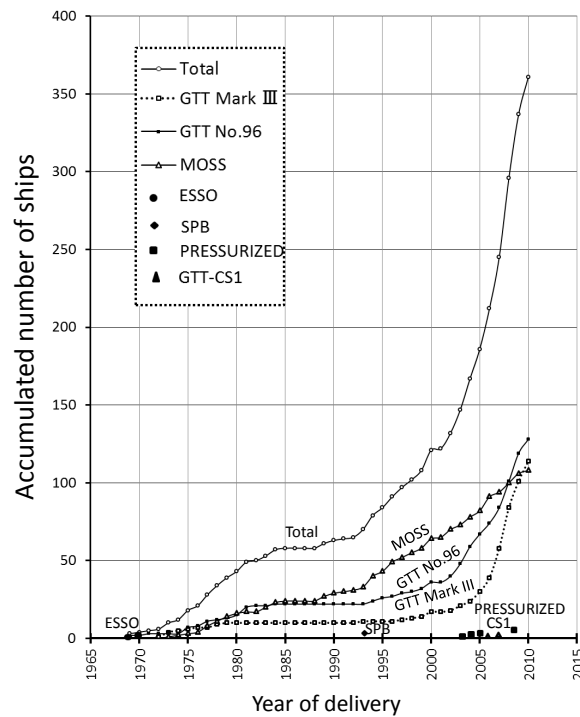


Figure 3: Accumulated historical number of LNG carriers

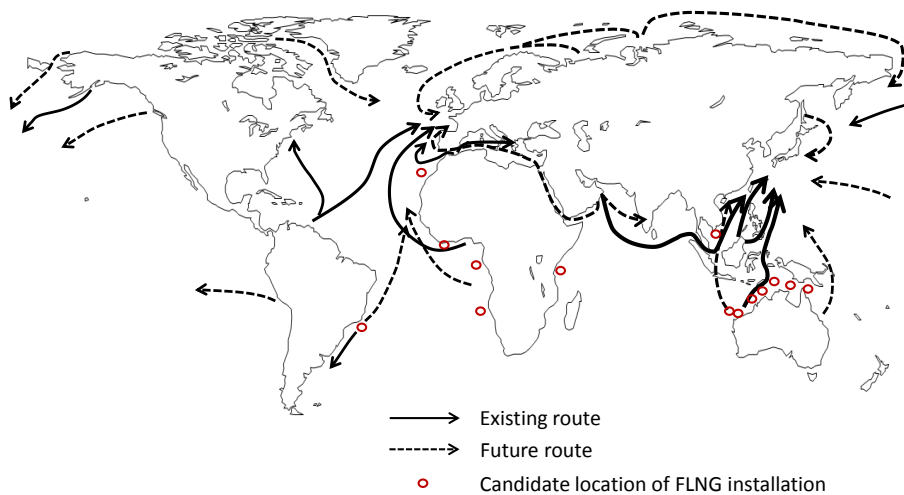


Figure 4: Current and potential future trades

natural gas with new technologies (shale gas) at the fields closer to consumer markets, and the development of new power sources such as wind, solar, etc., will determine the future requirements for LNG ships. Other factors might be the usage of LNG as ships' fuel, secession of some countries from using nuclear power after the accident of Fukushima nuclear power plant, etc. As an indication of such uncertainties, it can be

noticed that membrane LNG tankers ordered in 2010 was less than 10, but more than 45 in mid-2011.

The cargo containment systems used today for LNGCs are mainly of the membrane type (GTT's Mark III & NO. 96), and spherical type (MOSS) and in a few cases the structural prismatic design (SPB). Membrane systems and Moss type spherical tanks have different advantages. Suez Canal fees, which are dependant on the internal volume of a ship, penalize the spherical type design compared to the membrane type design, due to void space around the tanks being counted in computing canal charges. This has in large part contributed to there being more LNG ships built with Membrane CCS than Spherical type designs in recent years. The Membrane type has, in addition, a relatively higher utilization of the hull volume for the cargo capacity. That is, for the same cargo capacity, the ship dimensions of the membrane carriers are somewhat smaller than those of the spherical carriers.

Moreover, new aspects and issues of LNG appears with the offshore exploration, dedicated to gas fields or to monetization of gas associated with oil production. Similar to the FPSOs in the last decade, FLNGs need specific rules and regulations considering their specific design and operation. For example, large FLNGs may require longitudinal bulkheads considering both the sloshing in cargo tanks with intermediate filling levels and the strength of deck structure required to support the weight of the onboard liquefaction plant.

Songhursts (2009) reported that eight Floating Storage and Regasification Units (FSRUs) were in operation, three under development and seven projects were in the planning phase. On the other hand, more than fourteen FLNG projects were in different stages from feasibility studies to Engineering, Procurement and Construction (EPC) contracts. According to different sources the first FLNG system could be delivered in 2015.

Historically, when involving LNG transportation by ships:

- gas is transported from production area to onshore plant by pipes
- gas is liquefied and stored onshore at an export terminal
- liquefied gas is transferred on LNGCs for transport to market
- liquefied gas is delivered to a receiving terminal and offloaded to onshore storage tanks
- liquefied gas is then regasified as required and distributed to local consumers, by pipeline.

With new producing locations as offshore gas fields, for example SHELL PRELUDE or associated gas monetization, such as Petrobras PNBV Gas floating liquefaction, storage and offloading unit (FLNG of Gas FPSO), some operators are now choosing not to liquefy onshore, but directly at sea. In recent years, on the receiving terminal side, the FSRU concept has appeared, consisting of ships moored offshore supplying natural gas to shore after regasification in their onboard vaporization plants. FSRUs can be permanently moored, or temporary moored in case of ships which are both transport and regasification units.

In addition, the use of LNG as fuel for ship propulsion has started to be adopted, not only for LNGC but also other ship types. This will require LNG tanks to be installed on many different ship types, where in the past fuel oil tanks only have been installed. Distribution network of LNG will extend accordingly to more places than now, and LNG handling facilities use will become more widespread. This may also

drive the development of marine distribution systems where coastal navigation and inland waterways may be used to deliver LNG to smaller more distributed markets than at present. These new market developments will require dedicated rules and guidelines, together with training of all these crews and distributors to LNG handling.

3 SAFETY AND DESIGN

The Cargo Containment Systems for LNG for ship transportation are regulated by the International Maritime Organisation (IMO) through the International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (IGC code). The Safety aspects to LNG transportation are linked to handling Liquefied Natural Gas (LNG). The precautions and safety measures related to handling of LNG is mainly due to the following safety aspects:

- To carry LNG as a liquid at atmospheric pressure the LNG temperature has to be -163°C . Any spillage of Liquid Natural Gas on the ship steel would be hazardous and can cause immediate damage to the ship hull.
- Natural Gas in a mixture of between 5 to 15 % with air is explosive and shall be avoided.
- Liquid Natural Gas contains about 600 times the volume of Natural Gas in the gas phase. The boiling temperature is -163°C and heating Liquid Natural Gas may cause rapid increase of gas volume and if it is enclosed the gas will cause significant pressure build up.

The below discussed areas under this chapter can be directly addressed to these core safety issues.

3.1 Cargo Containment Systems

3.1.1 Non-self Supporting Tanks - Membrane Tanks

Membrane tanks are non-self-supporting tanks which consist of a thin layer (membrane) which is supported through insulation by adjacent hull structure (Figure 5). The membrane is designed in such a way that thermal effects are compensated for without significant stressing of the membrane. To control the effects on ship structure from the potential leakage of cryogenic liquids, a secondary barrier is required, by the IGC Code (Deybach, 2003). A secondary barrier is a liquid-resistant outer element of the cargo containment system designed to provide temporary containment of potential LNG cargo leakage and to prevent lowering the temperature of the hull structure to unsafe levels. The full secondary barrier of membrane tanks is fitted within the insulation system (Figure 5). This space is kept purged by inert gas which is circulated and has hydrocarbon detectors present so that any increase in the presence of methane can be readily detected and appropriate action should be taken. More details about the leakage control of membrane tanks are described in Chapter 3.11.

3.1.2 Independent Tanks

The IGC code categorises independent type of tanks in following tank types: Tank Type-A, Tank Type-B and Tank Type-C. In Figure 6 parts of the safety philosophy between the independent tank types is illustrated. All tank types are designed to comply with a comparable level of safety. The Type-A tank has a full secondary barrier with the function of providing a redundancy to any possible leakage regardless of the leakage is caused by fatigue cracks or due to over load of the tank causing a rupture of the tank primary barrier. The tank is designed with strength utilisation similar to a deep tank in a ship structure. The Type-B tank on the other hand is

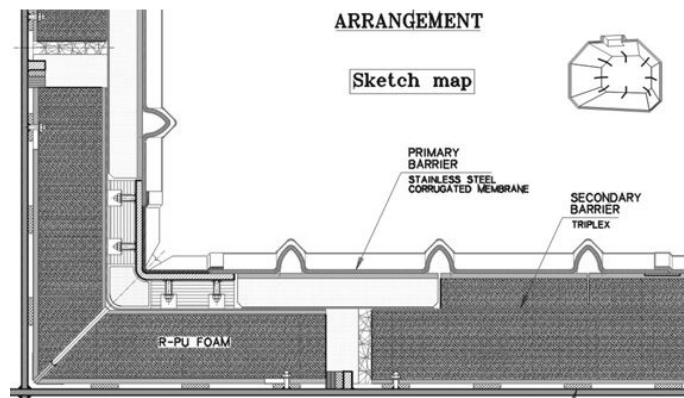


Figure 5: Build-up of a Mark III membrane tank (by courtesy of GTT)

designed with a partial secondary barrier that provides redundancy to fatigue cracking only. The size of the secondary barrier is dimensioned to the worst possible leakage that may occur. The tank design requires detailed control of the possible fatigue strength and the corresponding crack propagation properties. It is required by the IGC code to document that, if a crack occurs and grows through the thickness, the crack will remain stable for a sufficiently long time (normally 15 days). This is to allow the crack to be detected and the tank closed down to empty the cargo and to make necessary repairs. The Type-B tank is accordingly designed for redundancy to fatigue damage but has no redundancy for a damage caused by extreme loads. The material utilisation for extreme loading is therefore stricter as compared to a Type-A tank. This is to provide a larger safety factor against over loading. A Type-C tank on the other hand has no redundancy to either fatigue damage or damage caused by extreme loading. The material utilisation for a Type-C tank is therefore as strict as a B-type tank for extreme loading but more strict with respect to fatigue loading. For a Type-C tank the fatigue safety is incorporated in the formulation of a minimum design pressure, i.e., designed for large static loads compared to the dynamic loads resulting in small dynamic stress amplitudes.

The secondary barrier has the primary functions to provide temporary containment

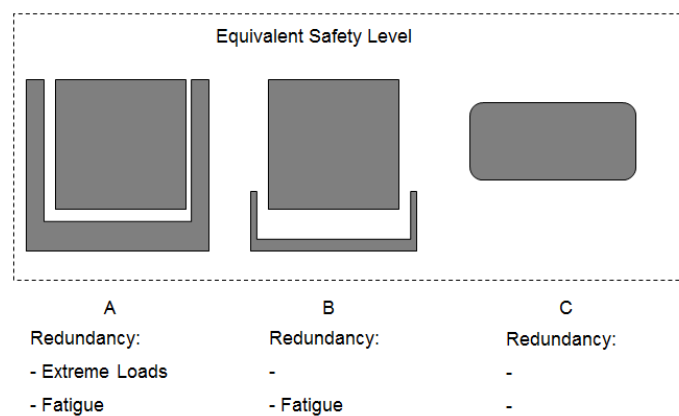


Figure 6: Different application of the safety philosophy between independent tank types

of cargo and to prevent the hull structure from being cooled to an unsafe level. The partial secondary barriers shall be designed to safely contain any envisaged leakage of cargo for a certain period of time (15 days). Continuous monitoring of the secondary barrier space is required to detect leakage of the primary barrier. The partial secondary barrier in the case of Type-B tanks (spherical or independent prismatic tanks) is usually designed as a drip tray capable of containing the estimated quantity of leaking cargo for sufficient time for corrective action to be taken.

For A-type tanks the hull structures are normally permitted as secondary barrier for cargoes with boiling temperature not lower than -55°C . The ship hull is accordingly not relevant as a secondary barrier in connection with LNG transportation. Type-A tanks are therefore normally not considered as a realistic alternative for LNG transportation. Type-A tanks are applied for cargoes such as Propane, Butane and Ammonia. These liquid gases are transported at temperatures above -50°C and therefore the ship hull itself can function as a full secondary barrier in case of a leakage from the cargo tank.

Independent tanks Type-B

Type-B tanks are divided into two main categories: Spherical tanks (MOSS type tanks) and Prismatic tanks primarily constructed by plane stiffened panels (IHI prismatic tank). The design pressure Maximum Allowable Relief Valve Setting (MARV) is normally 0.25 bar but shall not exceed 0.7 bar . These tank types are applied in LNG carriers with capacities up to $135,000\text{ m}^3$. However, larger ships have been designed. The tanks are built by Aluminium grade 5083-0 or Stainless steel L304/316 grade (Figure 7). Typical analyses required to document a Type-B tank are:

- Detailed FE based stress analysis
- Fatigue analysis
- Crack propagation calculation
- Calculation of leak rates
- Leak before failure analysis
- Tank support loads including interaction with ship hull deflections

Type-C tanks

The Type-C tanks are designed to a minimum design pressure. The Type-C tanks are usually not used for LNG transportation except for LNG carried as fuel where the advantage with the Type-C tanks is the possibility to handle boil off gas (BOG) by increased tank pressure. The tanks have the disadvantage compared to the other tank alternatives by a higher weight of the containment and a lower effective utilisation of space.



Figure 7: Cross sections of independent tank types A and B (with courtesy to DNV, Moss Maritime and IHI respectively)

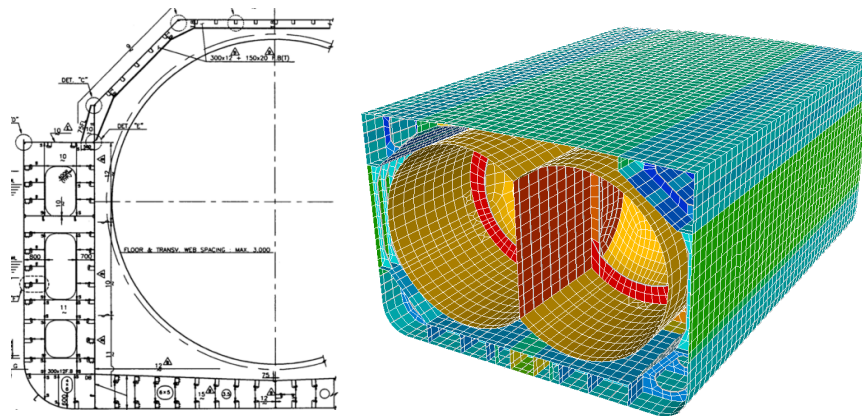


Figure 8: A cross section of a FEM model of a Bi-lobe Type-C tank (with courtesy to DNV)

The Type-C tanks can also be designed as bi-lobe tanks, see Figure 8. Bi-lobe tanks are normally designed for large tank sizes. Detailed finite element stress analyses are often required as documentation for these types of tanks.

3.1.3 New Tank Systems

Containment systems for LNG carriers have been well established and the regulation regime through Class Societies and the IGC code have been maintained without major changes for decades. However, the recent development in the offshore business, the offshore loading and offloading terminals have challenged the established designs and required the designs to be suitable for any filling height. These issues have forced changes to the established designs and new designs have been developed. Also the stricter emission requirements have made LNG an interesting alternative as fuel. Containment systems suitable for LNG fuel have resulted in new design proposals that do not directly fit with the existing regulations. In this section some of the new design features are discussed and safety issues are addressed.

Pressurized Prismatic Tanks

When LNG is used as fuel there is a need for boil off gas (BOG) handling when the ship is not in operation. This is commonly solved by increasing the tank pressure. A Type-C tank is therefore ideal for this application. The Type-C tank on the other hand is not ideal when space usage is limited. A new type of LNG tank design has been developed by Aker, the ADBT tank, where both pressure is handled and a prismatic shape is applied to utilize the required volume more efficiently in a ship installation (Lund, 2011). The IGC code is developed for particular tank designs but does not give guidelines for new designs not directly fitting into the existing tank definitions. New regulations need to be developed that give design guidelines that consistently maintain the safety level.

Type-A tank Designs

To develop a Type-A tank applicable for LNG transportation is a challenge as the hull is functioning as a full secondary barrier and need to be insulated from the low LNG temperature. Marine Gas Insulation AS (MGI) has developed such insulation systems.

Double Barrier Designs Based on Extruded Aluminium Profiles

Several new LNG tank concepts have been developed based on welded Aluminium profiles where the flanges are welded together and forming a double barrier system (GASTECH 2009 ADBT tank by Aker yards). The interpretation of the double barrier system may be a challenging issue as the barriers may be considered a secondary barrier system for fatigue damages but not for Ultimate strength loads. How strength criteria shall be applied need to be developed and consistently applied to maintain the safety on equal level with other LNG containment systems.

Compressed Natural Gas (CNG)

Natural gas can be brought to the consumer by ships with compressed natural gas (CNG) technology. These ships may serve for both storage and transportation. The cargo can be discharged directly into a land based gas grid via an on/offshore discharge terminal, an offshore platform or offshore buoys. The CNG technology does not require a liquefaction process and a regasification unit on each end of the transportation chain, but may require pressurized storage if ships are to minimize loading and discharging time. The natural gas is transferred in a gas to gas phase at a high pressure (but may have to be let down in pressure to match pipeline pressure specifications). A CNG system may also be an alternative to pipelines between the gas field and the consumer, although no such alternative had proved to be sufficiently competitive to be implemented at this time. The weight of the pressure containment system for CNG ships is significant. Several different CNG concepts have been developed the last years. The CNG Coselle system with coiled pipe in stacks, the Knutsen design with vertical steel pipes, the FRP wrapped steel pipes by Trans Canada, the horizontal Composite CNG tanks by CETech and Vertical steel pipes by EnerSea (Marine CNG Transport and Development Forum, London 22–23 Sept 2010).

There are no international common rules or regulations for CNG carriers such as the IGC code for LNG carriers. The Classification Societies apply different basis for their CNG rules. Some are designed based on pressure vessels codes such as ASME VIII, Division 3 code combined with additional specific requirements (ABS Guide for Vessels Intended to Carry Compressed Natural Gases in Bulk) and others based on modified offshore pipeline codes where improved production quality and stricter tolerance requirements open possibilities to optimize weight without reducing the safety levels (DNV Rules for Classification of Ships. Compressed Natural Gas Carriers), alternatively CNG designs may be designed based on Goal based standards. Commonly the CNG rule standards are benchmarked with the safety levels of LNG carriers. It may be a need for common international regulations covering CNG designs.

3.2 Unrestricted Filling

The LNG containment systems for transportation are large in size and a free surface of LNG may cause violent sloshing and high impact pressures due to tank oscillations generated by ship motions. Sloshing impact loading on the containment boundaries is especially an issue for the large membrane containment systems and not as critical for the spherical tank design where the spherical shape reduces the effect of sloshing impact on the tank boundaries. In all tank designs pump tower arrangements may be exposed to large loading due to sloshing. Traditionally the sloshing loading in membrane tanks has been limited through filling level restrictions. Normally the operation of ships are with almost full tanks (70 % – 98 % filling height) or alternatively at ballast with less than 10 % of the filling level in the tanks. However, LNG carriers and tanks filled and

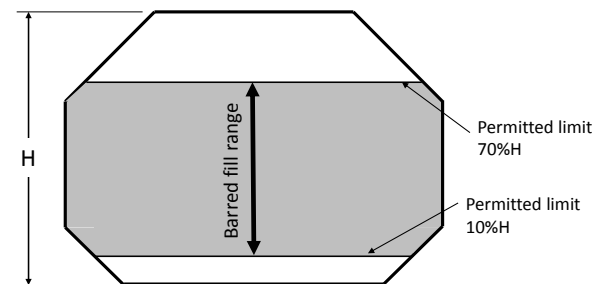


Figure 9: Membrane tank typical fill range limitation (GTT)

emptied at offshore installations require tanks to be able to operate at any filling level. The importance of controlling the sloshing loading for these applications is therefore extremely important to maintain the integrity of the containment system.

Ships with partial filling of liquid cargo may suffer the problem of liquid sloshing inside their tanks. The long safety record history of LNG carriers mentioned in the section 2.1 demonstrates the effectiveness of the filling level limitation of the cargo that has been applied in the traditional operation of the membrane carriers (see Figure 9). Basic idea of the filling level limitation is as follows. Since sloshing occurs mainly by a resonant motion of a liquid free surface inside a storage tank with a frequency close to the lowest natural frequencies of the tank-liquid system, the practical measure to mitigate the sloshing is to avoid the resonance by a proper design of the tank dimensions and the selection of a suitable fill level, as well as operating the ship in a manner such that sea-state conditions and ship speed do not encourage resonance.

The recent development with offshore terminals and FLNG applications has raised a new requirement of being able to handle unrestricted filling. This has required new investigations of the consequences due to possible sloshing effects on the tank containment system and pump tower arrangements. See further section 3.5 about sloshing.

3.3 Operation and Human Error

LNG storage and handling need to be considered in connection with the evolution of LNG usage and LNG industry.

Concerning LNGC, the hazards are well identified and controlled at present time. The good safety records (Section 2.1) of these ships are evidence of this assessment. Meanwhile, however, the increased fleets may be exposed to the risk of less skilled crews, less conscious ship owners regarding training and maintenance, and less demanding flag states. Mainly during transfer phases, the ship's crew is involved in cargo handling and is the focus point for human errors. With emergence of LNG as propulsion fuel for any kind of ships, these operations will be more widespread, while always critical. Authorities need to establish the correct procedure and safety knowledge accordingly.

Regarding FLNG, and to a lesser extent FSRU, their increased complexity may lead to increased exposure to human error. These new dual concepts (a ship and a plant in the same unit) shall be operated with good interface between process and marine crews, even more than for an FPSO considering the highest potential hazards (processing, handling and storage of very cold and volatile hydrocarbon products compared to crude oil) occurring in LNG treatment and storage. This duality should not be a culture clash, but rather a perfect integration of each ones' requirements and limitations.

The high complexity of LNG plant increases the number of possible failure modes. This requires a high level of operators' knowledge and competency of the integrated control systems. This pinpoints the need for the operator to be strongly involved in the design of the control system to gain the complete knowledge of this complex tool.

Less complex, but not to be neglected, is the fact that FLNG is a continuous flow process. These flows are to be managed and stored. Some FLNG plants produce several products (LNG, LPG, condensates) that cannot be mixed after separation. Particular attention in design, procedures and operators' training shall be paid to avoid mixing of highly incompatible products (LNG/LPG) during the following processes: a) during production, b) during offloading, using common loading hoses (non simultaneously), c) after offloading during purging process and d) redirecting in storage or any dedicated tanks.

As interlock systems may not solve this problem, the human factor is of major importance in that case. Additionally, inerting systems that can also be designed with common headers for cost/simplicity reasons may be a source of mixing products in case of overflow with more severe consequences than for FPSOs.

3.4 Structural Integrity Management

The modes of failure listed in Sections 3.5 to 3.12 along with associated hazards for the main components of the structural system should be considered in the integrity management plan, which is an integrated and focused effort that includes inspection and monitoring, with the goal of maintaining the integrity of the asset over its service life. Figure 10 presents a flowchart of the integrity management process, Quinn *et al.* (2007). As part of the integrity management plan, hazards need to be identified along with probabilities and consequences of failure to perform risk analysis. Based on risk assessment results the structural components can be prioritized to optimize inspection, maintenance and monitoring resources. Risk mitigating measures can be implemented in components with high risk. Findings could trigger local or global integrity assessment.

According to Lee *et al.* (2008) risk assessment is required at the early stages of design for mooring in survival conditions, off-loading operation, vessel collision, fire, hazardous operation of topside process, etc.

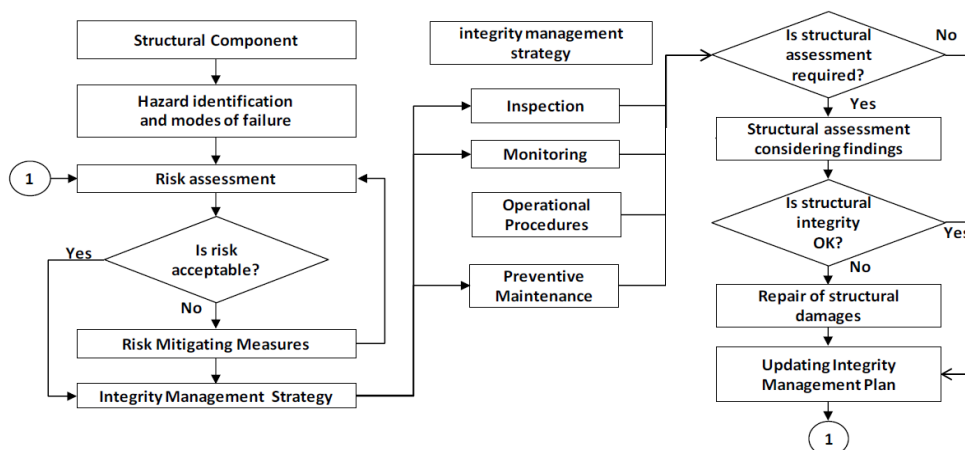


Figure 10: Integrity management process flowchart

Typically the integrity programs from the shipping industry are based on the detailed inspection performed at frequent intervals (5-year) in dry docks, along with thousands of ship years of service resulting in effective empirical inspection practices. In the past decade Reliability Based Inspection practices have supplemented the historic empirical approaches. In recent years integrity management programs have incorporated risk based inspections (RBI) for hulls, marine systems and specific components such as mooring components and risers, Wisch *et al.* (2009). Similar practices should be considered for the future FLNG systems.

FLNG systems will be designed and constructed/converted for continuous operation during their service life in a fixed offshore position. FLNG systems cannot easily be taken to dry dock for inspection, maintenance, and repair. These activities will normally be performed in situ. Therefore, permanent means of access have to be provided to facilitate these activities. The reliability of containment systems including second barrier construction will be more important in FSRU and FLNG applications where long term, continuous operation at a fixed site is required (Lee *et al.*, 2008). Consequently, the activities previously mentioned have to be risk assessed and performed with the required safety measures.

Verghese (2011) reported that the major hazards resulting from the release of flammable material can be controlled by suitable design. Effective measures can reduce the consequences of incidents that could compromise the integrity of the FLNG unit to an acceptable level.

3.5 Sloshing

One of the design issues for the membrane-type LNG carriers is the sloshing phenomenon, because the containment systems have almost no internal structures and they are prone to violent liquid motion. Many studies of sloshing in the membrane-type CCSs have been carried out and reported in conference proceedings of ISOPE, OMAE, PRADS, and related organizations, and also in some related technical journals (e.g., Kaminski *et al.*, 2010; Iwanowski, 2010; Kim *et al.*, 2010). When violent sloshing occurs, complex phenomena are generated, such as a mutual interaction between ship motions and sloshing, high impact load on the tank ceilings and walls, dynamic response of the tank structures, etc.

As a most practical measure to mitigate the sloshing, tank dimensions and selection of a suitable filling level has been considered in the design and operational phase of membrane LNG carriers (Section 3.2). However recent development of FLNG concepts requires a new investigation of sloshing since FLNGs should be operated without any filling limitation.

In view of new concepts and other changes the LNG market undergoes, the new designs or operational conditions should provide equal or higher level of safety as for the vessels currently operated. From a sloshing point of view the position of the tank is important – generally forward tanks have been more susceptible to sloshing than aft tanks. Also there is a possibility to operate one or more slack tanks. A practical solution might be to design the cargo storage and transfer system such that LNG is produced into smaller tanks (i.e., slack tanks) as it comes from the liquefaction train and then is transferred into empty tanks to quickly fill them to capacity. Based on a similar idea, Rokstad *et al.* (2010) proposed an optimization model for the redistribution of cargo to reduce sloshing loads in LNG cargo tanks during regasification of LNG from an FSRU.

To minimize sloshing events, rather than trying to design structures to withstand the sloshing loads, can be more reasonable measure against sloshing. Noble *et al.*, (2005) proposed tank geometry to minimize sloshing loads. Anti-slosh devices proposed by Anai *et al.* (2010) and Chun *et al.* (2011) might be another potential measure.

3.5.1 Rules and Standards

Sloshing loading may be determined either by direct calculation or model tests. Classification societies have rules of minimum sloshing loads to apply to the tank boundaries and pump towers, for example DNV Rules for ships Pt.3 Ch.1 Sec.4 C300. DNV Classification note 30.9 describes how sloshing loads for membrane cargo containment systems shall be determined. A guideline for sloshing loading on pump towers is defined by ABS "Sloshing and Structural Analysis of LNG Pump Tower". Similar rules, procedures and guidelines are provided also by other classification societies.

Differences in the rules may however be observed. Most of the classification societies adopt the comparative approach (by comparing the loads or responses in the new designs to those of existing and operating ones), while others suggest applying more direct approaches.

The comparative procedure is relatively simple and straightforward, but Zheng *et al.* (2010) pointed out that it is only applicable to a target design (including the ship, CCS and filling range) that is similar to a service-proven reference design and uses CCS's from the same designer.

Due to all the uncertainties involved in the sloshing phenomenon for a cryogenic liquid operating at its boiling point, an absolute approach may not be considered as fully reliable and must be applied with care.

3.5.2 Long Term Assessment (Including Screening Techniques)

The estimation of long-term sloshing response in LNG tanks is a challenging task. This is due to uncertainties related to determining the local fluid motion and loads acting on the tank structure, structural load effects, and their comparison with appropriate resistance criteria.

Performing experiments to provide sloshing pressures in the tank can be a time consuming step in this procedure (nevertheless experimental methods are the most usual technique for this purpose). The common practice of determining the critical conditions has been a screening procedure of the extreme conditions with a given return period. However, it is observed that more benign sea states with higher probability of occurrence may considerably contribute to the long term estimate; see for instance Graczyk *et al.* (2007), Rognebakke *et al.* (2009) and Ryu *et al.* (2009).

In order to limit the number of screening cases, methods for identifying conditions with large sloshing response are desirable. A simplified measure for sloshing response, developed on the basis of the idea of RAO for linear systems, may be utilized for this purpose. This measure is formulated so that it expresses sloshing severity in an approximate but effective way.

A semi-analytical method is applied by Graczyk *et al.* (2007) with the quasi-RAO calculated from tank acceleration spectra, with emphasized frequencies close to the tank natural frequency, considering nonlinearities in the response. This method shows reasonable accuracy while maintaining the computational efficiency of semi-analytical approaches.

A more refined approach is presented by Kim *et al.* (2010). They solve the internal sloshing problem by use of potential solver and formulate a quasi-RAO by analysing

the total sloshing wave energy in the tank. They assume a linear, damped free surface flow. The results indicate that the method may be an efficient tool for determining the critical cases.

Similarly, Cao *et al.* (2011) illustrate that potential flow solvers may be utilized to assess the sloshing severity.

3.5.3 Global Fluid Motion

The current practices to determine the global motions include semi analytical approaches such as multimodal method (Faltinsen *et al.*, 2000), numerical methods based on CFD and experiments. Each approach has its advantages and weaknesses. Semi-analytical methods are very time effective but their application is limited to basic tank geometries and excludes very low filling levels. Such restrictions do not apply for numerical methods or experiments which are time consuming and costly. For the experimental approaches the number of acquired time series is physically limited by the number and location of sensors applied. As to numerical simulations the combination of the large domain with high resolution, both temporally and spatially, may constitute a computational challenge. In practice, experiments have to large extent been used in commercial applications.

Experimental methods

Experimental techniques are mainly used for determining the global motions. The standard testing procedures are based on 6 degree-of-freedom experiments with 3-dimensional rigid-walled tank scaled as 1:70 – 1:35 and filled with water and a heavy gas in ambient conditions. Up to a few hundred small-sized sensors, are typically arranged in rectangular matrices mounted over the most exposed areas, specifically for each considered filling level. A large amount of literature documents the experimental set-up and procedures, including actuator rigs, models and instrumentation, see for instance work by Kuo *et al.* (2009) and references therein.

Full scale in-service measurement campaigns have also been completed, Lund-Joannsen *et al.* (2011). They complemented the model test investigations and numerical simulations by providing valuable benchmark data.

Investigations of the global flow sloshing effects were in the previous decade typically related to transport with different tank systems. In the most recent years they were further developed towards new applications such as FLNG and FSRU, see for instance Ryu *et al.* (2009) and Diebold (2010).

Another trend is that more attention is devoted to investigating local phenomena, dedicated for instance to investigate local hydrodynamic mechanisms, scaling laws or fluid-structure interaction; see Sections 3.5.4 and 3.5.6.

Numerical methods

CFD based numerical simulations are often used to determine the global fluid motion. Previous ISSC Committee V.2 “Impulsive Pressure Loading and Response Assessment” (2009) carried out an intensive survey of the numerical methods and pointed out some numerical problems such as insufficient accuracy of localized pressure, stability problem of numerical schemes and high CPU load. Efforts to overcome those issues are still continuing.

Since sloshing is a violent liquid motion, accurate tracking of the deformation of the free surface is important in numerical analysis. With the advancement of the recent

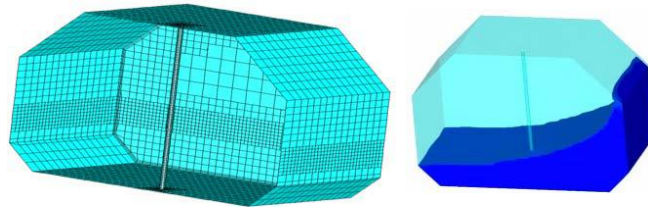


Figure 11: Fluid force analysis of pump tower (Reddy *et al.*, 2006)

Computational Fluid Dynamics (CFD) methodologies, many free surface tracking algorithms have been proposed. In view of their sloshing application, we may classify those algorithms into two groups, i.e. (1) using the Eulerian coordinate system with structured or unstructured grid systems (VOF, CIP, etc.; e.g., Moirod *et al.*, 2010, Liao *et al.*, 2011), and (2) application of a particle-based method without a grid system (SPH, MPS, etc.; e.g., Guicher *et al.*, 2010, Lee *et al.*, 2010). In the latter case, particle tracking is carried out based on the Lagrangian-type formulation.

In general, however, numerical techniques have significant problems when considering highly nonlinear waves and/or overturning waves, the effect of gas cushions and fluid-structure interactions. A comparison study of experimental and numerical sloshing loads in partially filled tanks is reported by Brizzolara, *et al.* (2011). A set of two-dimensional cases, for which experimental results are available, is considered to assess the merits and shortcomings of different numerical methods for sloshing evaluation, namely two commercial RANS solvers (FLOW-3D and LS-DYNA), and two academic software (Smoothed Particle Hydrodynamics and RANS).

An important application of numerical techniques is for calculating the interaction between ship motions and sloshing. Here Wang and Arai (2011a, 2011b) and Moirod *et al.* (2010) reported the results of coupling simulation using ship motion codes based on linear potential theory and CFD-based sloshing codes for wall force evaluation. A significant effect of sloshing on a ship's transverse motion was reported by Wang and Arai (2011a).

Most recently, nonlinear vibrations of an elastic structure with two partially filled liquid tanks subjected to horizontal harmonic excitation are investigated by Ikeda (2011). The equations of motion for the structure and the modal equations of motion for the first, second, and third sloshing modes are derived by using Galerkin's method, taking into account the nonlinearity of the sloshing. Then, van der Pol's method is employed to determine the frequency response curves. Bifurcation sets are also calculated to show the influence of the system parameters on the frequency response.

Yet another application of numerical techniques is for calculating the pump tower responses. The study by Reddy and Radosavljevic (2006) identified different uncertainties associated with present numerical and experimental techniques to obtain fluid forces on pump towers in an indirect way using Morison's equation. As an alternative to the use of the Morison equation, investigations into the feasibility of direct numerical estimation of fluid forces on a pump tower within an enclosed tank resulting from the sloshing of liquids was carried out (Figure 11). Experimental data obtained from 1:50 model for a typical 210k LNG ship for regular and irregular motions by Berget *et al.* (2006) were used as a reference, enabling both the enhancement of the design appraisal procedures and the validation of CFD techniques for assessing sloshing loads and their capability.

3.5.4 Local Effects

It is observed that local phenomena such as jet creation, gas entrapment and escape, compression of the gas and fluid, the change of momentum, and hydro-structural interaction to a small degree influence global flow pattern in the tank. However, these phenomena certainly affect local pressures and structural response of the CCS. Even for repeatable global flow large spreading of the local pressures is observed. The spreading may to some extent be explained by the inherent instability of the fluid and chaotic nature of sloshing, but even more important is the sensitivity of the pressure to local physical phenomena, such as jets and sprays accompanying fluid impact, gas pocket entrapment, ventilation and gas escape or gas fraction in the fluid.

Moreover, scaling of attained pressure time series plays a vital role in post-processing of sloshing model tests. In order to determine a valid and consistent scaling law, the local flow mechanisms need to be well understood. It needs to be certain that the experiments represent all the hydrodynamic phenomena governing the full scale system. Among such local phenomena many are related to the raised elements of the membrane surface and their effect on the fluid flow. It is not an obvious question whether and how the surface protrusions should be modelled.

Investigations of the local effects are therefore intended either to determine the local flow and the following pressure distribution for different wave fronts impacting on a tank wall (with or without membrane surface modelling), or – more often – to study the underlying physics.

Raised element

Local flow investigations most commonly focus on single hydrodynamic impacts on the tank wall, or its part. This is in purpose of ensuring the best possible repeatability of the impact and – in case of the experiments with the simplified set-up – due to practical issues that the correct global sloshing flow may not be represented. Such simplified experiments are drop tests and breaking waves in a flume.

Both these forms for simplifying fluid-structure impacts are found to be an important research tool. Although the flow pattern differs from the global flow in a tank, these tests allow studying physical phenomena related to the local flow.

This is due to the fact that the impacting surfaces may easier be controlled than in the sloshing tank and much larger scales may be applied (for instance breaking waves in a flume up to the full scale have been studied in the Sloshel project, Bogaert *et al.*, 2010c).

Another simplified set-up offering possibilities to investigate local effects is 2-dimensional tanks undergoing 1-degree-of-freedom oscillations of small amplitudes. Here the local flow may be investigated under more realistic sloshing-specific impacts. An example here is studies by Kuo *et al.* (2009) and Graczyk *et al.* (2012), where the effect of membrane corrugation and raised Invar edges on the local pressures is demonstrated. The protrusions may both increase and decrease the local pressures depending on the impact type (wave steepness varies for different conditions) and location in relation to the protrusion. Similar effects are observed in the large/full scale experiments of the Sloshel projects.

Scaling

Studies on scaling laws have mainly been based on investigating physics governing the phenomena involved in sloshing. Focus has been placed on hydrodynamics and thermodynamics of the gas and fluid with local effects such as fluid impact on the wall with

accompanying jets and sprays, gas pocket entrapment, possibility for condensation at the liquid/vapor phase boundary, ventilation and gas escape, gas fraction in the fluid and hydro-structural interaction.

Developing an experimental set-up representing the complete physical system of the violent fluid motion in the tank, including all mechanisms and phenomena involved in sloshing, would be a very challenging task. Usually, formulation that considers separately chosen elements of the system is developed and investigated. This is then validated through studying repeated experiments in different scales. This may also be challenging due to parameters that cannot easily be scaled together with the rest of the set-up, for instance sensors' size.

Yung *et al.* (2010) discuss importance of the ambient vapor during impact event. They introduce a dimensionless *interaction index* and by this illustrate how the ambient vapor properties, liquid properties and interaction between them influence the resulting pressures.

3.5.5 Structural Response

The strength assessment of structures exposed to transient dynamic loads such as those generated by sloshing impacts generally requires the assessment of the dynamic response of the structure. The structural response does not only depend on the pressure peak magnitude, but also temporal and spatial variation of the loading.

Dynamic structural response to sloshing excitation has been investigated independently by the scientific community, classification societies, and industry. To assess the structural response to sloshing loads, two methods are suggested by LR (2009): one is Direct Dynamic Finite Element Analysis (FEA) another is Indirect Dynamic FEA. The Direct Method applies the design sloshing loads scenarios directly to the containment system using a dynamic FEA, which is a more straightforward process. The Indirect Method applies representative sloshing loads, defined by a nominal sloshing pressure and a range of rise times, to derive a Dynamic Amplification Factor (DAF) Envelope curve. The results from a static FEA analysis are then factored by the maximum DAF value to obtain the dynamic structural response. The advantage using the indirect method is that the analysis results are effectively independent on the actual design sloshing load scenarios. However, it is more likely to provide a conservative estimate of the response.

The local response of the membrane system is actually coupled with the response of the steel plate supporting it. A common simplification is to assume that the steel structure does not respond to the sloshing loads and hence, the insulation system is rigidly supported. This is based on the assumption that the pressure duration is much shorter than the structural natural period of the steel plate. In practice, the steel panel that supports the insulation may be flexible under the relevant load conditions and in addition, the loading may cause the steel plate to deform. This effect may significantly affect the stress distribution in the structure (Graczyk and Moan, 2011).

To be consistent with the principles of the comparative assessment, the structural response analysis methodology needs to be capable of accurately predicting the structural response in the entire range up to a level where damages are likely to occur in the structure. Depending on the response characteristics of the considered structure, it may be required to consider non-linear structural response (DNV, 2006).

3.5.6 Fluid Structure Interaction

An important issue related to the sloshing responses is also fluid-structure interaction. Due to the complexity, many investigations on sloshing problems decouple structural

response from fluid, which implies that the structure is assumed rigid. However, the interactive dynamic behaviour of liquid and elastic tank due to their interaction under various loading conditions can have vital impact on the integrity and safe operation of the system.

A wide review of fluid-structure coupling algorithms as well as fluid and structure models is presented by Kamakoti and Shyy (2004) with a specific attention to aero-elasticity. The degree of coupling between the fluid flow and structural response may be classified as loosely-, closely- and fully coupled. Problems associated with fluid impact on the marine structures are presented by Faltinsen (2000). Theoretical and experimental studies are described with a special attention to slamming on ship hulls.

Different approaches may be utilized to account for the hydro-elasticity. A simple method is to account for the hydrodynamic mass forces by modifying the structural mass by the constant, prescribed value of added mass. However, for sloshing impact in a tank a complex fluid flow in the impact region is observed. A number of parameters such as thickness of the fluid layer, its spatial extent and density (due to aeration) may influence the added mass.

Therefore approaches capable of determining an instantaneous value of added mass may be required for calculations of the coupled response. A number of approaches based on the Wagner theory have been developed. This theory describes the initial stage of a water entry problem and is valid when the penetration depth is much smaller than the body width. An important feature of these approaches is that the hydrodynamic coefficients can be calculated analytically, which makes the calculations very time efficient. Korobkin *et al.* (2006) present a two-dimensional method for the fluid-structure interaction with a coupling of the finite element (FE) method for the structural response with a Wagner theory-based approach for hydrodynamics. Malenica *et al.* (2006) show a possible application of this method to an analysis of two-dimensional coupled response of the membrane structure to a simplified impact.

An experimental and numerical investigation of the elastic response of the tank wall under sloshing impact is presented by Lee and Choi (1999). The authors apply the normal mode decomposition method with the thin plate theory and combine it with the boundary element method for the fluid solver. Similar approach is presented by Rognebakke and Faltinsen (2006) who analyse the coupled response to impact on the tank roof. Calculated strains are compared to experiments with an elastic upper part of a wall.

Investigations on dynamic responses of LNG ship most commonly focus on inner liquid ~ tank systems. However, coupling effect between liquid cargo sloshing and LNG ship motion can be significant at certain frequency range of partially filled tanks. This is of great concern to the LNG FPSO/FSRU operation. The coupling effects are expected to become more important as the size of LNG carriers significantly increases with rapidly growing demand. Therefore, natural characteristics of an integrated system consisting of inner LNG liquid ~ elastic membrane type LNG ~ external sea water is investigated by Xiong and Xing (2007a) and dynamic responses excited by regular waves and random motions are analysed by Xiong, *et al.*, (2007b). Figure 12 shows the two selected modes of the coupling system, in which mode 7 has an anti-symmetric pattern of both sloshing motions of internal liquid and external sea water, whereas mode 9 describes a tank rolling motion coupling with the sloshing effect of the internal liquid.

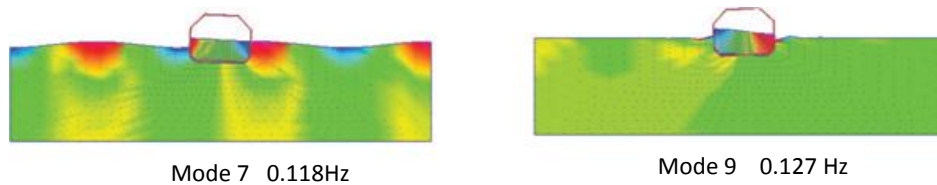


Figure 12: Natural vibration modes of the integrated fluid-structure interaction system (Xiong *et al.*, 2007a)

Studies on fluid-structure interactions are mostly on two phases of liquid-tank interactions. The air-water two-phase fluid flow systems are investigated by Price and Chen (2006) using a curvilinear level set method to simulate free surface waves generated by moving bodies or the sloshing of incompressible fluid in a 2D right tank. The air-water sloshing problem is also recently studied by Thiagarajan *et al.* (2011) in which fundamental analysis and parametric studies on excitation and fill levels are presented. Three-phase interactions, involving air, liquid and elastic tank are investigated by Xiong *et al.* (2006). The dynamic behaviour of an air-liquid-elastic tank interaction system is investigated numerically and experimentally. Based on these simulations, the guidelines to be considered in the dynamic design of LNG containers are provided. Dynamic response analysis of on-shore LNG storage tank with fluid-structure interaction effects has also been investigated (Xing *et al.*, 2009). It has been demonstrated that the developed computer code provides a useful numerical tool for free and forced vibration analysis in linear domain where fluid-structure interaction effect needs to be addressed in the design stage.

For violent sloshing impacts, nonlinear approaches have been developed in recent years. For example, the Meshless local Petrov-Galerkin method based on Rankine Source Solution (MLPG_R) (Ma, 2005,2008) has been employed to simulate the interaction between breaking waves and 3D fixed cylinders and dam breaking on a block (Zhou and Ma 2010). This method is found much faster than other methods for solving fully nonlinear water waves (Yan and Ma, 2010). In addition, the MLPG_R method works with a longer time step than the conventional Smoothed Particle Hydrodynamics (Ma and Zhou 2009, Zhou and Ma, 2010). The hydro-elastic behaviour of a 2D structure subjected to violent waves is studied using an improved MLPG_R method (Sriram and Ma, 2010). The comparison has shown reasonable agreement between the numerical results and the experimental data available in literature. However, there are still many uncertainties associated with wave breaking and splashing, formation of air pocket and air bubbles, and dynamic interaction between wave impact and structural response during violent sloshing.

Another interesting attempt to solve a fluid-structure interaction problem relevant for sloshing in LNG membrane tanks is presented by Nam *et al.* (2005). The finite volume method for the fluid is combined with the FE method for a structure. The analysis is stepwise. First, an uncoupled fluid simulation is performed in order to find time instants for sloshing impacts on the structure. Subsequently, the coupled analysis is carried out for limited temporal and spatial extent.

3.6 Collision/Grounding/Flooding

Issues related to hull-ice interaction and iceberg collision dominate the literature related to collisions and flooding. This is due to the growing interest in LNG trans-

portation through the arctic seas leading to increased concern of CCS integrity due to hull-ice interaction. Recent literature on this topic is described in Section 4.3.

The structural response of CCS in both membrane and spherical types of LNG ships for selected ship-ice interaction scenarios have been investigated for possible operation routes in Arctic areas through the Joint Development Project (Wang *et al.*, 2008). In these studies, ice loads and loading areas in the hull structure were determined based on the energy theory. A local FE model including the partial hull structure with the skirt structure has also been developed for structural analysis. The critical loading location with respect to the deflection of inner hull is determined and deformation of CCS is analysed. Based on the linear buckling analysis and nonlinear static FE analyses it is found that the strength of the both CCS of membrane-type LNG carrier and the skirt structure of spherical-type LNG carrier is sufficient to resist the design ice loads.

As the structure of the containment system is based on materials that are non-standard in marine technology, understanding of the dynamic structural response is still limited. Sensitivity studies addressing the response and strength of the containment system such as by Paik (2006) or Lee *et al.* (2006) are essential.

Another issue is related to the considerable development of offshore LNG terminals. Here, attention is devoted to the risk for ship-to-ship or ship-to-terminal collision. On this topic, Deetjen *et al.* (2008) presented an analysis and consequences of collision between two ships for different event scenarios. Also Montewka *et al.* (2010) performed a risk analysis for an LNG carrier colliding with a tug, as a part of a mooring operation.

Existing codes play an important role for the safety under accidental conditions. IGC code defines the extent of damage that should be assumed and the location of cargo tanks in terms of permitted inboard distances.

Flooding condition is regulated by the IGC code in terms of permitted permeability of different compartments and guidance is provided on internal arrangement and effectiveness of the watertight bulkheads. There are additional requirements regarding withstanding unsymmetrical flooding, permitted position with respect to waterline as well as maximum heel angle and residual stability.

A new IGC code, which is planned to be issued in 2014, may impose more strict requirements to the distance requirement between the tank and the side shell and become a variable depending on the volume of the cargo tank. However, the stricter requirements will have limited, if any, effect on the existing LNG ship designs.

3.7 Fatigue

3.7.1 Introduction

Fatigue analysis is basically required to be carried out for independent tanks Type-B and may, in special cases, be required for independent tanks Type-C and semi-membrane tanks. The objective is to determine the fatigue life of all welds and plates that may lead to leakage of the tank. A fatigue analysis shall be carried out for parent material and welded connections at areas where high dynamic stresses or large stress concentrations may be expected. The fatigue properties shall be well documented for the parent material and welded connections being used in the design. For less investigated and documented materials, the data on fatigue properties shall be determined experimentally. Due attention shall be paid to effects such as: specimen size and orientation, stress concentration and notch sensitivity, type of stress, mean stress, type of weld, welding condition and working temperature.

The fatigue strength of the structure considered is normally defined by Wöhler curves (S-N curves).

3.7.2 Crack Propagation Analysis

- The purpose of a fracture mechanics crack propagation analysis is to show that probability of an extensive cargo leakage due to a fatigue failure is small. This is verified by showing that a potentially growing fatigue crack fulfils one of the following criteria: If the crack grows through the tank thickness, it shall result in a cargo leakage of a rate not exceeding the capacity of the partial secondary barrier (the drip tray) draining system.
- If it can be shown that the resulting through-thickness crack will not reach a critical length during 15 days in the most probable largest load spectrum the ship will experience during 108 wave encounters North Atlantic, Leak-Before-Failure (LBF) has to be proven.

The fracture toughness properties of the tank material and its welded joints in the thicknesses used in the design are normally required to be documented. Crack propagation analysis is in general required to be documented for all welds that can cause a leakage. However, the amount of analysis can be reduced by a careful screening to define selected areas of stress concentration taking into account maximum fabrication tolerances. The size of the cracks assumed in the calculations shall be of minimum the size as those found by applicable NDT methods.

Dynamic stresses are driving fatigue crack growth, whereas the rupture of a fatigue crack of a given size is governed by a maximum ULS load situation. The primary parameter governing final rupture of a fatigue crack of a certain size is the most probable largest one time stress amplitude, static plus dynamic amplitude, during the design life in relevant environmental conditions (usually North Atlantic).

The size of initial defects used in the analysis shall be decided considering the production quality of the builder. As guidance the following initial crack sizes in way of Heat Affected Zone (HAZ) through thickness may be used for the builders who control high production quality standard. (BS7910:1999. "Guide on methods for assessing the acceptability of flaws in metallic structures.")

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

The design crack propagation data are normally to be based on the mean-plus-two-standard-deviation of test data.

For the cargo tank, crack propagation data (C and m in Paris' equation) need to be determined for welded and base material together with the associated crack tip opening displacement (CTOD) values. Documented test data for both room temperature and cryogenic temperature should normally be available.

In order to evaluate the residual fracture of fatigue cracks over the lifetime of the vessel, fracture mechanics analysis has to be referred to the ULS stress range to be compatible with the total ULS stress amplitude that governs potential fatigue crack rupture. (DNV Rules for Ships Pt.5 Ch.5 Sec.5) The fatigue stress range can preferably be determined at a $Q = 10^{-4}$ probability level as for the SN – fatigue approach and extrapolated to the ULS stress range level using the long term Weibull stress distribution. As the IGC code specifies the Weibull shape parameter $h = 1$ this means multiplying the 10^{-4} stress range with 2 to arrive at the 10^{-8} stress range.

For use in the fracture mechanics analysis the principal stresses determined for SN-curve fatigue analysis is often to be further processed as given below:

- a) In order to correctly evaluate crack propagation, the static value plus the dynamic design life ULS amplitude of the principal surface stresses shall be calculated in addition to the dynamic stress ranges.
- b) Based on the inside and outside values of the principal surface stresses, the stresses are to be split into membrane and bending parts separately for dynamic stress ranges and for static plus ULS amplitude values. This is essential for the fracture mechanics analyses but is not necessary for the Miner-Palmgren fatigue analyses.
- c) Select the largest membrane stress for the analysis. This will give the fastest crack growth through the thickness and hence the shortest fatigue life. However, in some cases it might be necessary also to check the maximum bending combination in which the crack will grow faster in length than in depth.

3.8 Vibration

3.8.1 LNG Pump Tower System

The pump tower with its associated pumps and piping, as shown in Figure 13, is the main equipment for discharge and loading of LNG. For membrane type, it is located close to the aft bulkhead, hanging down from the liquid dome and connected to lateral support base at the tank bottom. The structure of pump tower is very slender and flexible, so its fundamental natural frequency is relatively low, which may cause resonance problem with the propeller excitations around normal operation range and hence cause a fatigue failure due to vibration to the pump tower structure. The vibration analysis on pump tower structure is important at the initial design stage for safe operation of the LNG carrier.

3.8.2 Vibrations of LNG Pump Tower

There are potential vibration risks in LNG ships that are magnified in the new larger LNG ships. These vibrations may have unavoidable sources such as propellers, main engines or cargo machinery. If these vibrations are in tune with the natural frequency of the tank system they may have disastrous consequences.

For LNG carriers driven by gas or steam turbines, the propeller is the main source of excitation and, therefore, the main engine may be ignored in the vibration analysis of the pump tower. However, for large LNG carriers driven by low-speed diesel engines,

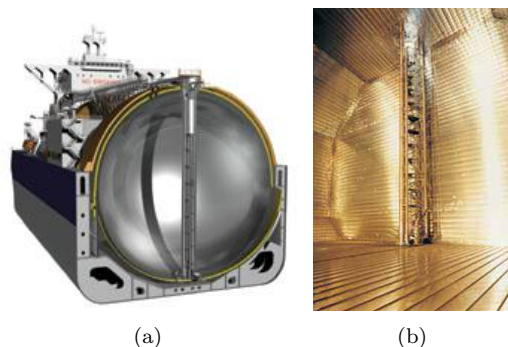


Figure 13: Pump tower within (a) Moss and (b) membrane types of LNG tanks
(http://explo.com/lng_carrier and Lee, 2006, respectively)

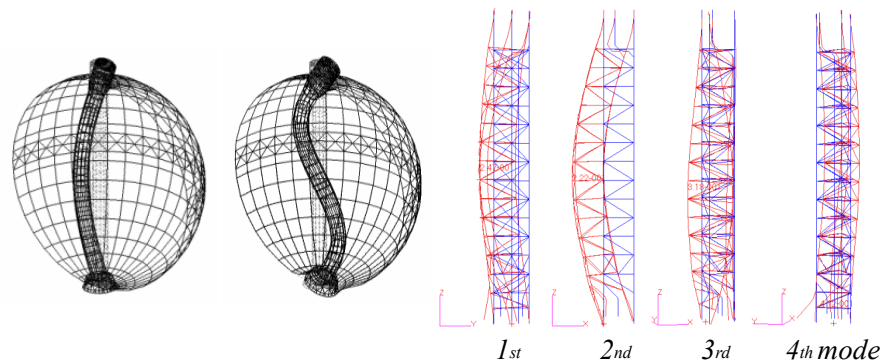


Figure 14: Mode shapes of pump tower in Moss type and membranes tanks (Lee *et al.*, 2006)

excitations from the main engine as well as propeller are to be considered in the vibration analysis.

Vibration assessment of the pump tower is described (American Bureau of Shipping, 2006; Lloyd's Register, 2008). The free and forced vibration of pump tower due to main engine and propeller are considered. The analysis procedure provides guidance on the selection of loading conditions, tank location and filling levels, boundary conditions and critical areas to investigate. Excessive vibration is to be avoided in order to reduce the risk of structural damage such as cracking on the liquid dome, base plate, or tubular joints of pump tower structure. The acceptance criteria for pump tower vibration are provided in terms of the vibration limits for local structures.

A vibration analysis for pump tower of both Moss and membrane type of LNG carriers was carried out by Lee *et al.* (2006) for empty and full loading conditions to reveal vibration characteristics (Figure 14). Added mass effect due to LNG is considered by the virtual mass method of MSC/NASTRAN fluid capability. Also the vibration measurements at sea trial were carried out to confirm the analysis results.

The Campbell diagram as shown in Figure 15 is very useful to identify potential vibration risks in LNG ships, particularly for the new generation of large gas ships (Lloyd's Register, 2006; Lee *et al.*, 2006) by checking the coincidence of vibration

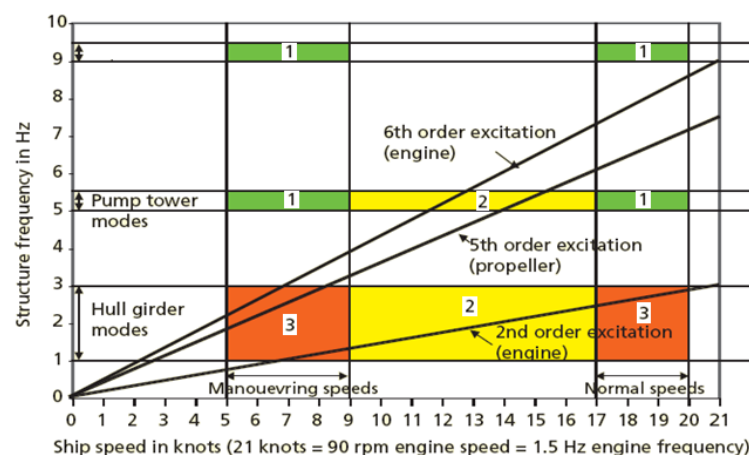


Figure 15: Campbell diagram

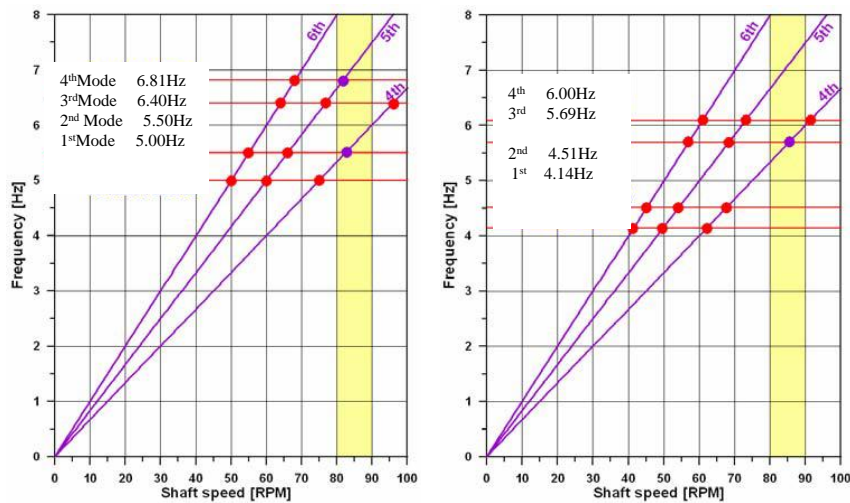


Figure 16: Campbell diagram for pump tower for empty (left) and fully filling (right) in membrane type (Lee *et al.*, 2006)

sources with natural frequencies, in which zone 1 indicates no resonance occurs; zone 2 the transitory resonances may occur while the zone 3 the permanent resonance occurs.

For LNG carriers, propeller blade number is generally selected among 4, 5 or 6 blades in view of ship propulsion performance. Propeller shaft speed is running mainly at the normal operation range, 80 – 90 *rpm*. Surface force and bearing force induced by the propeller are major excitation sources for longitudinal and transverse vibration, and their frequency components are propeller blade order component and higher harmonics. As shown in Figure 16, the resonance of longitudinal vibration will be predicted at 82.5 *rpm* when 4-blade propeller is adopted. Therefore an actual forced vibration response should be checked to ensure the structural safety.

3.8.3 Load on Pump Tower

The load on the pump tower is the combination of the following load components:

1. Hydrodynamic load due to sloshing on the pump tower structural members
2. Inertial and gravity load due to global ship motion on the pump tower
3. Thermal load due to low temperature of LNG cargo
4. Pump torque

The loads on the pump tower are calculated at each time step when the sloshing simulation was carried out. Instantaneous load distribution when the following dominant load parameters (i.e., transverse and longitudinal forces on pump tower as well as these forces on pump tower base support) reach maximum value is used for the structural analysis of the pump tower (American Bureau of Shipping, 2006). Fluid force measurements on LNG pump tower were also reported by MARINTEK (2006).

3.9 Fire Safety

For LNG carriers the fire safety requirements as measured in SOLAS related to tankers in general apply depending on flag state authorisation. In addition special requirements for LNG carriers apply. Special considerations are made to isolate gas safe

spaces from gas dangerous zones. The cargo hold space is basically segregated from other areas on the ship. The superstructure and areas is to be insulated with A60 insulation toward the cargo area and special requirements to the fire main system and a fixed water spray system in the cargo area apply. For rules and regulations of LNG fuelled ships (DNV Rules Pt.6 Ch.13) equivalent safety philosophy is applied for the LNG containment systems as for LNG carriers where A60 fire insulation is applied between areas where LNG is stored and toward other areas. Fire main system and water spray systems are designed equivalently as for LNG carriers. In addition pressure relief valves shall be dimensioned for the maximum vapour generated for a defined fire heat exposure (IGC code Ch.8.5). For LNG handled on offshore installations similar fire safety measures apply (Offshore Service Specification DNV-OSS-103, "Floating Production and Storage Units or Installations"). Fire safety of LNG on board offshore units may require special considerations through Risk Assessments depending on designs and process equipment on board. Applicable methods are described in for example DNV-OSS-121 "Classification Based on Performance Criteria Determined from Risk Assessment Methodology".

3.10 Temperature Control of Hull Structures

The several safety-related criteria in the relationship between LNG cargo tank and temperature control should be established (IGC Code). The design condition of air temperature and seawater temperature for the Boil-Off Rate (BOR) evaluation must first be defined and temperature conditions that the structure during the lifetime can meet the extreme environmental conditions of poles or equator should be considered. In addition, it is important to how the boundary of temperature distribution will be established with respect to the structure to evaluate the structural safety of the inner hull which surrounds the cargo tank. This assumption has a major impact on the material selection of the structure surrounding cargo tank and the design safety. So the temperature condition is an important basis for analysing the structural safety and thermal characteristics (BOR). The interpretation of this condition is an important factor in the design of cargo tank, so accurate information or reasonable hypothesis is necessary. To determine the grade of plate and sections used in the hull structure, a temperature calculation must be performed for all tank types with following assumptions (IGC Code).

- The primary barrier of all tanks must be assumed to be at the cargo temperature.
- In addition, where a complete or partial secondary barrier is required it shall be assumed to be at the cargo temperature at atmospheric pressure for any one tank only.
- For worldwide service, ambient temperatures should be taken as $+5^{\circ}\text{C}$ for air and 0°C for seawater. Higher values may be accepted for ship operating in restricted area and conversely, lower values may be fixed by the Administration for ships trading to area where lower temperatures are expected during the winter months.
- Still air and sea water conditions shall be assumed, i.e. no adjustment for forced convection.
- The cooling effect of the rising boil-off vapour from the leaked cargo should be taken into account where applicable.

If the calculated temperature of the material in the design condition is below -5°C due to the influence of the cargo temperature, the grade for material shall be selected in accordance with IGC code.

The temperature of cofferdams between two cargo tanks is also calculated in the design conditions. This range of temperature would preclude the use of conventional steel grade for the bulkhead since such a design temperature is far under the grade E limit (-30°C); it requires special cryogenic steel, which are extremely onerous (materials procurement, special handling and welding procedures, special QA/QC procedures, connection to conventional steel fabricated blocks,...). A more economical solution is to provide a heating system with following requirement in accordance with IGC Code.

- The heating system shall be arranged so that, in the event of failure in any part of the system, standby heating can be maintained equal to not less than 100 % of the theoretical heat requirement.
- The heating system shall be considered as an essential auxiliary. All electrical components of at least one of the systems provided in accordance with item 1 shall be supplied from the emergency source of electrical power.

The new concept of FLNG with two-row tank arrangement unlike the normal LNGC which has the single-row tank is now emerging (SHELL FLNG in PRELUDE offshore gas field). This arrangement is driven by the very high deck loads imposed by the onboard liquefaction plant and the need to have at least one longitudinal bulkhead to support the deck structure. Regarding to the material selection of structural members in centre longitudinal cofferdam, it is needed to clarify the interpretation of the below clause in IGC code.

In all cases referred to in 4.8.1 and 4.8.2 (IGC Code) and for ambient temperature conditions of 5°C for air and 0°C for seawater, approved means of heating transverse hull material may be used to ensure that the temperature of this material do not fall below the minimum allowable values. If lower temperatures are specified, approved means of heating may also be used for longitudinal hull structural material, provided this material remains suitable for the temperature conditions of 5°C for air and 0°C for seawater without heating.

This interpretation is highly critical since it may request very special material for this normal structure regardless of heating coil system application. The two-row FLNG is very new concept totally different from single-row LNG carriers. Then the Code should be updated reasonably considering this new novel concept.

3.11 Leakage Control

3.11.1 Soundness Control and Cargo Containment System Monitoring

The overall layout of a gas carrier is similar to that of the conventional oil tanker from which it evolved. The CCS and its incorporation into the hull is, however very different due to the need to carry extremely low temperature cargo under pressurized, or refrigerated, or under a combination of both conditions. So perhaps more than any other single ship type, the LNG tanks encompasses many different design philosophies. As the LNG CCS has its own unique characteristics such as a cryogenic cargo storage and material, extreme thermal and fluid stress, in-tank pressure deviation from Boil off Gas (BOG), double barrier concept under IGC requirement, and specially designed insulation structure for Boil off Rate (BOR) control, it is strongly required to cover all these parameters and extensive stresses. These kinds of CCS should resist against externally harsh condition such as wave and temperature for more than 20 – 30 years

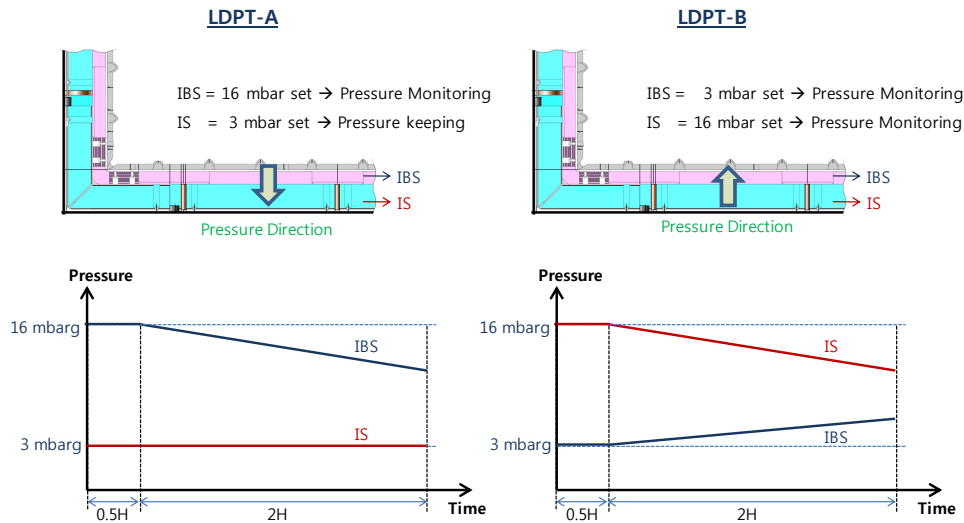


Figure 17: Low differential pressure test monitoring to detect leakage of secondary barrier on membrane type tank

and could be monitored its soundness throughout their lifetime including the construction stage. During construction, ammonia, NH_3 , and helium, He , leak tests are well known method for CCS soundness inspection. Before delivery, differential pressure testing (DPT) is performed after the first thermal cool down test to check the soundness of primary and secondary barriers. During voyages pressure differences change between 1st and 2nd insulation spaces (IS: Insulation Space, IBS: Interbarrier Space) and inert/methane gas detection systems are used to monitor the system's soundness. In addition, periodic dry dock inspections, carried out every 2.5 years and 5 years, low differential pressure testing (LDPT, Figure 17) or differential pressure testing (DPT) is used to check the tightness of the CCS.

3.11.2 Primary Barrier Failure Detection

The IGC Code requires permanently installed instrumentation to detect when the primary barrier fails to be liquid tight at any location where a secondary barrier is required. However the Code does not require the instrumentation to be able to locate the area where liquid cargo leaks through the primary barrier or where liquid cargo is in contact with the secondary barrier. Temperature indicating devices and methane gas detectors are widely used to monitor the primary barrier condition. During construction, independent Type-B tank should be tested hydrostatically or hydro-pneumatically. For membrane type tanks helium or ammonia leak testing is used to control and confirm the primary barrier tightness at the construction stage. Periodically vacuum test for secondary barrier soundness control is used to confirm the primary barrier tightness in a similar procedure. More specifically, membrane type tanks such as the GTT MK III or NO96-2 adopt primary barrier leak inspection methods and membrane deformation inspections with guideline criteria. SPB tanks can be inspected by differential pressure test between internal and external area of the tank wall. Methane gas detection systems are used to monitor the insulation space during voyage condition. MK III membrane corrugation deformation inspection is performed using GTT's guideline.

3.11.3 Secondary Barrier Soundness Control

According to the IGC Code, it is required to have a full secondary barrier for Membrane and a partial secondary barrier for Type-B tank LNG CCS. The purpose of the secondary barrier is to ensure that cryogenic liquid cargo cannot reach to the inner hull in the event of a breach of the primary barrier. The IGC Code also states that this containment of leaked liquid cargo should be for at least 15 days. The implication is that the secondary barrier is required to be liquid tight (not necessarily gas tight). Therefore a small leak in the secondary barrier can be tolerated without the risk of jeopardizing the integrity and safety of the vessel. For independent Type-B tanks which are fitted with a partial secondary barrier – i.e. a drip tray, the condition of the external spray shield that covers the insulation panel can be ascertained through visual inspection from the hold space. This spray shield is not required to be liquid tight but it must be capable of containing any leakage, directing it to the drip tray where detection can take place. Design of drip tray is confirmed by the so-called LBB (Leak before break or leak before failure) concept. It is also a requirement of the IGC Code that the secondary barrier of the membrane type tank is to be “capable of being periodically tested by means of a pressure/vacuum test or another suitable method approved by the administration” (IGC Code 4.7.7). Nowadays a vacuum test is the most widely adopted method to check the secondary barrier soundness of membrane type tanks by applying a vacuum condition to make a pressure difference at secondary barrier and monitoring the pressure decay rate. In case of MK III system, secondary barrier soundness is confirmed by differential pressure testing between IS and IBS.

3.12 Spillage Control

Specifically to offshore LNG plants, spillage refers to very large leaks due to failures in cryogenic systems, either LNG, or cold refrigerants used in the liquefaction process, leading to large amount of product exposing structures to major thermal shocks.

Industry is beginning to address this very new aspect for ship/offshore LNG industry and damage scenario simulations are currently based on more on engineering principles than on codes and regulations.

For design, depending on the client's requirement, consideration may be made up to a pipe full bore rupture (e.g. may be in the range of more than 20") which will result in a very large volume of cryogenic liquid spill until the deficient part is isolated from the process.

Designers may consider three main parts implicated by these risks: topsides structure, hull deck and hull sides, being treated differently.

The area in topsides or on deck subject to potential spillage are identified and localized. Where a full bore rupture is considered, the parts subject to contact with fluid are to be protected against thermal embrittlement (special coating, wood, stainless steel...). In the same time, the spill is to be enclosed by coamings and redirected by channels and scuppers, either to the sea or to drainage tanks. To ensure their function, materials of these barriers are to be carefully chosen, and any eventualities in drains are to be foreseen and planned for (e.g. ice plug in scuppers, overflow, vaporization return pressure...)

Where cryogenic liquid pools on structures may happen, and if not rapidly sprayed (e.g. main deck), coatings and protections are probably required. For vertical side shell, the approach may be different.

Effects of large LNG spill in seawater are not really known. Between the two extreme suppositions 1) a smooth vaporization and 2) rapid phase transfer (RPT) equivalent to an explosion, there is a room for a temporary exposure of side shell to cryogenic fluids. Quantification of RPT is not easy, and shall mainly lead to consideration of overpressure in structural design more than thermal load design for hull.

When dealing with vaporization LNG pool above water, main unknowns are the time exposure and heat quantity taken to side shell steel. Even if this value is determined, next question is: shall we protect the side shell against this thermal ingress. Effectively, such approach may lead to unrealistic designs, such as special coatings or wooden plating of major parts of the units (400 m long, draft variation and wave elevation up to 25 m!). Moreover, offshore units design life is usually 20 to 40 years on site without dry-docking, which is far longer than any coatings design life, especially subject to continuous wave ingress. To avoid “over design”, and considering that all these units are double sided, an alternative approach could be considered: The side shell gets damaged due to thermal embrittlement (shell plating, including or not the stiffeners), and assessment of the consequences of ingress and to evaluate the integrity of the hull with a locally damaged side shell – in the same manner as for a collision. Additionally to insurance of the unit design safety after such event, designers shall develop in-situ means of repair (in design repair philosophy), avoiding disconnection and dry-docking which will cause a major revenue loss for such installations.

Although these points remain in the field of research, decisions will be taken in the very next future, together with classification societies and international players, to allow the safest realistic approaches.

4 SAFETY AND DESIGN FOR SPECIFIC LNG APPLICATIONS

4.1 Offshore LNG Chain

As shown in Figure 18, the LNG chain that was commonly based on onshore plants is currently extended with offshore units such as Floating LNG (FLNG) and Floating Storage and Regasification Unit (FSRU). All these concepts differ from common LNGCs by following points:

- Whole or part of the chain is located in weather exposed offshore locations
- Intermediate fillings at sea (continuous production process, loading/offloading of products, roll-over)
- LNG ship to ship regular transfers at sea (FLNG offloading to LNGC or LNGC offloading to FSRU), demanding for some case up to two operations per week
- Several type of products onboard units (LNG, LPGs and condensates)
- Spillage risks in topsides, in cargo handling pipes used at sea, in offloading systems
- Continuous loading/offloading at sea, stressing hull concomitantly to wave loads
- Inspection, maintenance and repair at sea, without dry-dock, and as far as possible without production stops

At present time, codes are well developed for LNG carriers, following shipyard and cargo containment designers, with continuous progress of classification society and IMO in rules and approval of cargo containment and cargo handling systems. New offshore units, for which industry is in advanced phase compared to classification society rules and international regulation, introduce new aspects not already covered.

Regulations and codes/standards for the design, construction and operation of LNG facilities are summarized in Foss (2006), i.e. IMO-IGC code, which are applicable to

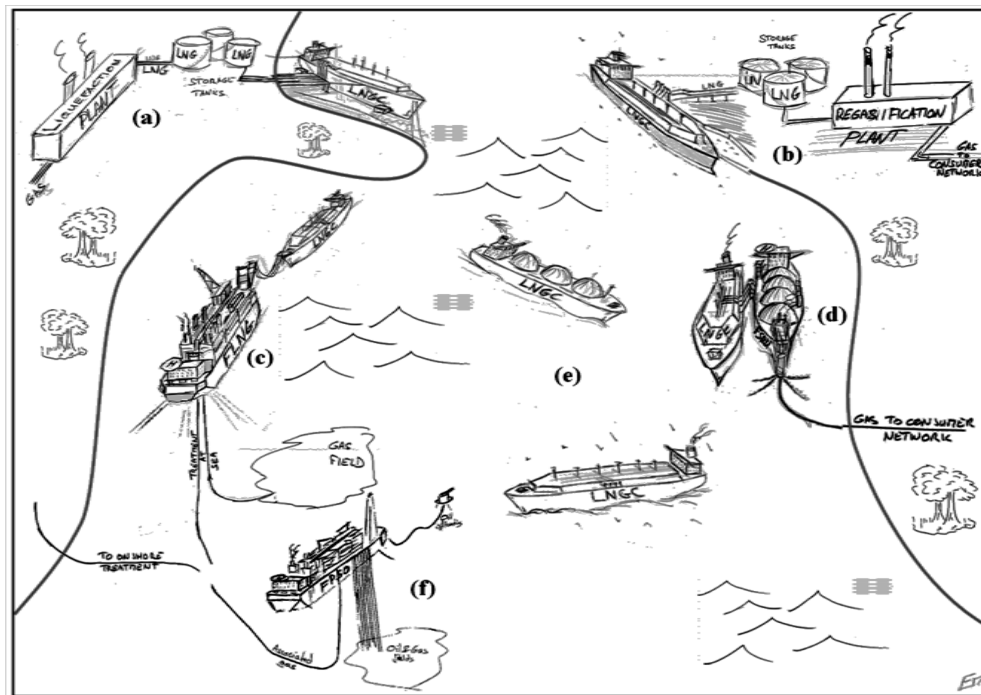


Figure 18: LNG chain involving LNG storage and handling at sea

- (a) **Onshore liquefaction plant**, with LNG storage and export facilities to LNGC (sheltered area)
- (b) **Onshore regasification unit**, with import facilities from LNGC (sheltered area), LNG storage and distribution in consumer network
- (c) **FLNG or Gas FPSO**: Offshore unit, with gas import (from gas field or FPSO associated gas line) treatment and liquefaction, LNG storage and export facilities to LNGC (open sea)
- (d) **FSRU**: Offshore unit, with import facilities from LNGC (open sea), LNG storage, regasification distribution in consumer network
- (e) **LNG ship trading**
- (f) **FPSO**: Floating production, storage and offloading, for crude treatment, and eventual associated gas export to shore or FLNG.

LNG/LPG transportation vessels and loading/unloading terminals. These codes can be used for FLNG systems with modifications to account for the new application and associated hazards.

Classification societies have developed and recently issued guidance and rules for FLNG systems, such as DNV's OTG-02, which addresses critical aspects to the integrity management such as: risk assessments, inspection and maintenance philosophies, RBI, inspection of containment systems, sloshing assessment, fatigue assessment, and corrosion issues as reported by Fagan (2011). The OTG-02 (2011) guide presents the rules/standards applicable to the hull, topsides, and cargo tanks of the FLNG systems.

4.2 *Floating LNG, FLNG and Floating Storage and Regasification, FSR, Units*

As exploration for hydrocarbon resources has continued to expand in offshore waters, interest in the production of LNG at offshore locations has grown. The oil and gas industry has decades of experience with both onshore LNG liquefaction and offshore floating production storage and offtake, FPSO, units.

4.2.1 *Floating LNG Production, Storage and Offloading, FLNG, Units*

In recent years there have been moves to develop and deploy floating offshore LNG production, storage and offtake units, FLNG. While a few such units have been designed for marginal fields with production in the 1 million tonnes per annum range, most of these units have been conceived as very large, 2.5 – 5 million tonnes per annum or even larger to take into account the economies of scale from deploying large scale LNG plant.

Although a number of these projects are underway, no such unit is yet in service (as of December 2011).

The use of FLNG units has several unique challenges as compared to an LNG carrier. These units have to support a very large deck load; can be 50,000 *t* or greater and so require a hull structure that will support this. This has usually resulted in at least one longitudinal bulkhead being incorporated into the design, where LNG ships usually have no longitudinal bulkheads other than the side tank bulkheads. This leads to a reduction in the ship's athwart size of the cargo storage tanks which can be beneficial in reducing sloshing, but since these tanks are cold it may be necessary to treat any longitudinal bulkheads in the same way as the transverse bulkheads in an LNG carrier, i.e. cofferdam bulkheads with heating to maintain structure temperature in an acceptable range.

One of the biggest issues with FLNG units is the potential for loss of cryogenic containment within the deck mounted liquefaction plant. Leaks from valves, vessels, piping etc. in the liquefaction system that can result in direct contact of LNG with the structure of the hull can have severe consequences, although we have already dealt with this risk to some extent in LNG carriers during loading and discharging. The exposure to this risk on a production FLNG is higher due to the continuous nature of the process. A number of ways to deal with this are currently being applied including minimizing flange connections where leaks might occur, fitting of stainless steel drip trays to contain potential spills, and fitting of insulation material on deck in way of potential spill areas (wood or concrete have been examined alternatives).

Another area of concern for FLNG units is the transfer of cargo from the unit to offtake tankers. In most cases FLNG project will not be producing a single product and so must store and offtake LNG, LPG and condensate liquids. This will result on a complex set of offtake equipment and considerable planning and operating issue that need to be addressed with multiple offtake vessels servicing a single FLNG unit. To date most proposals are looking at LNG offtake in a side-by-side mode using the standard ship loading manifold on existing LNG carriers, but work has been undertaken to look for tandem loading of LNG where the offtake LNG carrier would approach the stern of the FLNG unit and take cargo over the bow with a special loading system, in the manner that FPSOs now routinely do.

4.2.2 *Floating Storage and Regasification, FSR, Units*

As import of LNG has expanded into regions of the world which have not previously seen much of that trade, a solution to provide LNG storage and regasification quickly

has been developed. The Floating Storage and Regasification Units is usually a ship shaped unit (often a converted ship) which in addition to its LNG cargo containment system is fitted with regasification equipment which can pressurize the LNG up to pipeline discharge pressures and then vaporize the liquid back to gas for onward transmission to customers on shore. A few such units are in service today with some acting as combined transport and FSR units, loading LNG at the liquefaction port, transporting it to the destination and then sitting in port or offshore while regasifying the LNG for pipeline transmission to customers. Other units are permanently stationed at the receiving ports and simply act as storage and regasification installations.

Regasification equipment may derive the required heat from use of warm seawater in some locations or by heating using natural gas boil off as fuel for the process. It is also necessary to boost the pressure of the gas to be delivered to match pipeline requirements. This has usually been done by compressing the liquid LNG prior to vaporizing.

While the size and weight of regasification plants are small as compared to full-scale liquefaction plants some of the same issue as mentioned above do pertain, i.e. deck strengthening to take additional load, design features to mitigate effects of cryogenic spills and relating to ship to ship transfers.

4.3 Arctic

LNG shipping routes move towards more severe sea regions such as North Atlantic or ice-covered waters in the Arctic. These may bring about new technical and operational challenges as well as increased risk for human errors. The latter may for instance result from the demanding environmental conditions (cold, humidity, icing, darkness) or from the continuous icebreaker proximity that may lead to collisions. It is a “general perception” that the safety for operation in arctic areas will increase if the operation can be designed with less dependency of human assistance.

Enhanced environmental concern in the arctic may require special safety considerations for such transport. This is due to the limited access to any help or assistance (long distances and harsh environment) as well as challenging defeat of potential pollution. On the other hand, activities related to LNG transport and storage constitutes a lesser environmental concern as for instance oil related operations.

Problems associated with ship and offshore structures in the arctic are widely elaborated by the Committee V.6 Arctic Technology. In the present section issues specifically related to LNG transport and storage are only described.

The main technical and operational challenges related to shipping LNG in arctic waters may be found in e.g. Tustin (2005). Among them, the CCS integrity with hull ice interaction is stressed dominant. Here the risk may be related to the hull deformation that may result either in threatening its integrity or reducing the tank volume associated with rapid increase in tank pressure.

The CCS integrity is investigated by Han *et al.* (2008). They perform a risk analysis of the membrane CCS (No.96) investigating thoroughly the capacity of the double hull deflection and potential accidental ice loads. The authors considered various ice features for capacity calculations, including collision with level ice, ice ridge, ice floe, iceberg and ship stuck in level ice. It is found that the invar membrane can afford very large inner hull deflection before the chosen survival criteria are reached; also the integrity of the inner hull structure is to be checked before the invar limit condition due to a risk of ballast water leakage into the containment layer. The results are compared

to the grounding accident case. The potential accidental ice loads are calculated for Baltic Sea and East Canadian coast operation.

Similar study is presented by Suh *et al.* (2008) for Mark III CCS. In a case study presented an ice class LNG carrier is considered under various design, accidental and fatigue scenarios in both ice and non-ice operation regimes. The results of the direct strength calculations or evaluations according to the class rules are analysed in view of the risk for the leak of the cargo. Similar study and supported by an experiment on the CCS specimen is also presented by Oh *et al.* (2010a).

Another interesting study on this topic is presented by Wang *et al.* (2008) for both membrane and spherical type LNG carrier. The authors investigate structural response of the CCS's under six different loading scenarios that may be caused by the ship-ice interaction. More detailed description of this work may be found in Section 3.6.

These studies analyse different, both accidental and design ship-ice interactions. Works focused specifically on the iceberg impact include Oh *et al.* (2009) who performed a study on the membrane type CCS response to the iceberg collision and Lee *et al.* (2010) presenting similar analysis with somehow refined parameters. On the other hand, iceberg-ship collision for the spherical tank type is investigated by Kim *et al.* (2008).

Reference to associated issues may also be found in Section 3.6 Collision, Grounding and Flooding.

Another challenge related to shipping LNG in arctic waters is vibrations due to hull-ice interaction and their effect on the integrity of CCS and pump tower. Problems related to vibration are described in more detail in Section 3.8 Vibration.

Another aspect pointed out by Tustin (2005) is ship operation in severe but ice-free waters. For such extreme wave environment a careful attention may need both fatigue strength - for example with respect to discontinuous decks commonly constructed of higher yield steels at Moss-type carriers, and large sloshing loads for membrane-type carriers. See section 3.7 and 3.5 for more details on fatigue and sloshing, respectively.

LNG carriers for arctic operations may require certain winterization adjustments both in terms of design, equipment and operation techniques; see e.g. Tustin (2005).

Berg and Bakke (2008) investigated ship-to-ship transfer of LNG in arctic environment. They analysed the risk related to different phases of the operation. It is also observed that climatic changes question applicability of historic metocean data for planning future operations.

Sun *et al.* (2009) investigated the motion and loading on the LNG ship with ice breaking hull. Analysis of ice breaking performance in various ice types was evaluated by model tests, while seakeeping and manoeuvring characteristics in high waves were investigated by a numerical tool.

An interesting concept for winterization of the transport, storage and production processes is possibility of applying unmanned systems and hence reducing the dependency on humans in such severe conditions.

4.4 LNG as Fuel

Alternative fuel for propulsion has come into focus especially the last decade when environmental issues and restrictions to emissions requirements have been addressed on the political agenda. Here LNG as fuel is an interesting alternative. There have

been ships propelled on compressed gas (CNG) prior to year 2000, also LNG carriers have been using dual fuel boilers for decades. However the first commercial LNG fuelled ship was the ferry “Glutra” in year 2000. The year after DNV came with the first rules for LNG fuelled ships (DNV Rules for Ships Pt.6 Ch.15, 2001). In 2009 the first international interim guideline for LNG fuelled ships was published (MSC 285.86). There is currently development of an IGF code for LNG fuelled ships in IMO. The rules are expected to come into force in 2013/2014. Until 2010 about 30 ships have been built with LNG fuelled engines. The number of ships propelled on LNG is increasing rapidly and in 2 – 3 years more the number of LNG fuelled ships with almost double. The new ECA requirements coming into force in 2015 will likely facilitate a large number of LNG fuelled ships in the years to come. The rules and regulations for LNG fuelled ships take the safety of LNG handling from the experience of LNG carriers. The safety philosophy will have to be adjusted to that LNG will be placed in other locations than previously experienced as well as the handling of LNG will be with ship crew not educated with the main purpose of transporting LNG but with using LNG as a commodity fuel alternative with a much more frequent schedule of filling and handling of LNG. This is an area that still may require further risk assessment studies to mature the safety understanding of the new LNG application.

5 CONCLUSIONS

It is concluded that the investigation of safety of LNG transportation indicates following areas where there is a need to determine the safety aspects and to develop consistent regulations for:

- LNG at offshore applications. Process and systems operability of LNG need further to be investigated. Also evaluations of sloshing loads at any filling levels. Semi empirical experience from LNGC operation cannot be generalized and the safety aspects should be further investigated. LNG spillage and safety handling and protection are areas where further investigation work may also be beneficial.
- There are many new LNG containment systems under development that do not fit into the established IGC code definitions of tanks. New generic regulations need to be established for how to handle new innovative containment system designs.
- LNG as fuel is a new area coming quickly as an attractive fuel alternative in shipping. LNG will be applied for any type of ships and LNG fuel tanks may be located in other areas than in cargo areas. This may challenge the established safety philosophies applied for LNG containment systems. A new IGF code is currently under development but the safety aspects should be revisited and evaluated.
- To examine the sloshing response of CCS and associated structures at actual seas, development of holistic analytical methodologies that combine CFD analysis of the liquid, fluid-structure interaction analysis, ship motion analysis, wave modelling, etc. is recommended.
- Innovation is required for the in situ inspection and monitoring of the hull structures and containment tanks to watch the performance of critical components and acquire data from structural variables such as stress, accelerations, fatigue, etc. These data together with operating variables (i.e. temperature, pressure and sea monitoring devices) could be effective for not only early detection of potential failure but also in further studies such as fatigue assessments, validation of analytical methods, etc.

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18th INTERNATIONAL SHIP AND
OFFSHORE STRUCTURES CONGRESS

09-13 SEPTEMBER 2012
ROSTOCK, GERMANY

VOLUME 2



COMMITTEE V.3
**MATERIALS AND FABRICATION
TECHNOLOGY**

COMMITTEE MANDATE

The committee shall give an overview regarding new developments in the field of ship and offshore materials and fabrication techniques with focus on trends which are highly relevant for practical application in the industry in the recent and coming years. Particular emphasis will be given to the impact of welding and corrosion protection techniques on structural performance, on the development and application of lighter structures and on computer and IT technologies and tools, which link design and production tools and to support efficient production.

COMMITTEE MEMBERS

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David Brennan
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Jose Gordo
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Liangbi Li
Stephen Liu
Tetsuo Okada
Florian Pires
Marc Yu

KEYWORDS

Welding of extreme thick plates, thickness effect, corrosion protection, composite material application, standard comparison, design for production and manufacturing, linking design and production.

18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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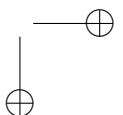
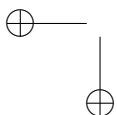
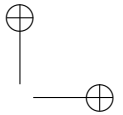
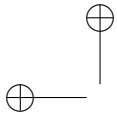
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18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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1 INTRODUCTION

After years of growth the global economic crisis has deeply affected the shipping industry. There are however clear signs of recovery in the last year. The shipbuilding industry has realised that, due to the crisis, new innovative designs and design and production methods are necessary to decrease operational costs, production costs and emissions, whilst meeting the changing rules and regulations. In this report ISSC committee V.3 discusses recent development in materials and fabrication technology.

Chapter 2 focusses on worldwide trends in materials and fabrication methods. Developments in fabrication technologies, such as welding and corrosion protection are dealt with in Chapter 3. Applications of composite materials are increasing. Some main areas of applications and research in those areas are described in Chapter 4. A comparison of current worldwide standards is made in Chapter 5. Chapter 6 gives an overview of current developments in the linking of design and production in computer applications, thus increasing the efficiency of ship building.

2 NEW TRENDS IN MATERIAL AND FABRICATION METHODS

2.1 World

After five years of extraordinary growth, the global shipbuilding industry has experienced a sharp turn-around of trend. From the second half of 2008, the global economic crisis has deeply affected this industry worldwide enduring an unparalleled collapse of demand for new ships, the situation of most shipbuilding yards is expected to remain difficult for some time as the order books continue to deplete affecting the entire value chain in shipbuilding, Clarksons (2010).

Nevertheless, there are clear signs that the recovery has started. Growing cargo volumes, improved earnings for ship-owners and also the slow increase of new order volumes are welcome and encouraging news, CESA (2010).

The strategic nature of the shipbuilding industry encouraged many countries to develop domestic capabilities to build ships without necessarily taking into consideration

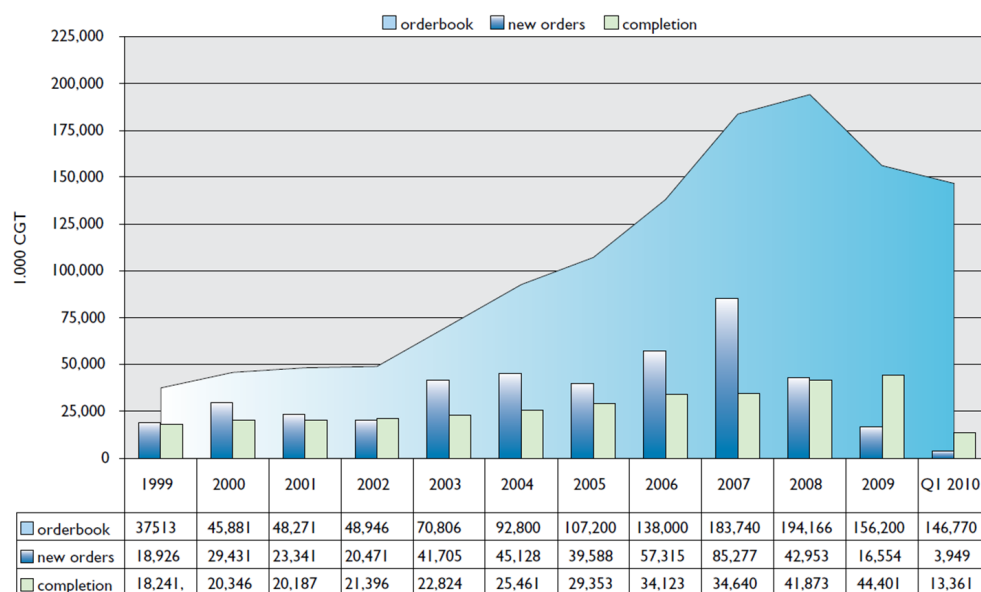


Figure 1: World Commercial Shipbuilding Activity – CESA (2010)

the developments in the world market. The most prominent example is South Korea, and more recently, the Republic of China which in 2009 accounted for 28% of the world production (compared to Korea 32% and Japan 21%).

The low ordering level in 2009 combined with higher deliveries brought the world order book down by 21% (see Fig. 1). Despite a large total order book, many yard's workloads are shrinking rapidly. New orders in 2009 totalled 16.5 million CGT which equals roughly 1/3 of the completions. Ordering has slightly improved during the first quarter of 2010. However, a much higher activity is needed to balance the rate of deliveries. The global merchant fleet today is relatively young; the average age of the container ship fleet is 10 years. The need for replacement due to age in this specific segment will, therefore, contribute less to the new building requirement in the coming years.

In an effort to maintain as much capacity as possible, sectoral programmes are implemented by several governments around the globe to maintain the national shipbuilding industry. Many market observers strongly criticise these moves as obstacles to the necessary market correction, which will prolong the current imbalance of supply and demand (see Fig. 2). The current crisis prompts the need to consider and adopt new designs to reduce the operational costs and lower emissions. Leading shipping companies recognise that low emission operations save costs, open quality sensitive markets and will be the key driver to profitability in the future. The innovations in the shipbuilding sector have the potential to greatly reduce the operating costs as the sector has already developed and demonstrated significant advances in green technologies.

The trend of ship production in different parts of the world is described in the following sub-sections. As pointed out in the previous report, shipyard production technologies are heavily influenced by the types of ships being built, size of shipyards, geographical aspect, etc. In this context, the European shipbuilding industry focuses on higher end sectors such as cruise ships and naval vessels. Production technologies with regard to thin plate thicknesses are therefore important. The Asian shipbuilding industry focuses on cargo ships, attaching importance to productivity in mass production in large shipyards.

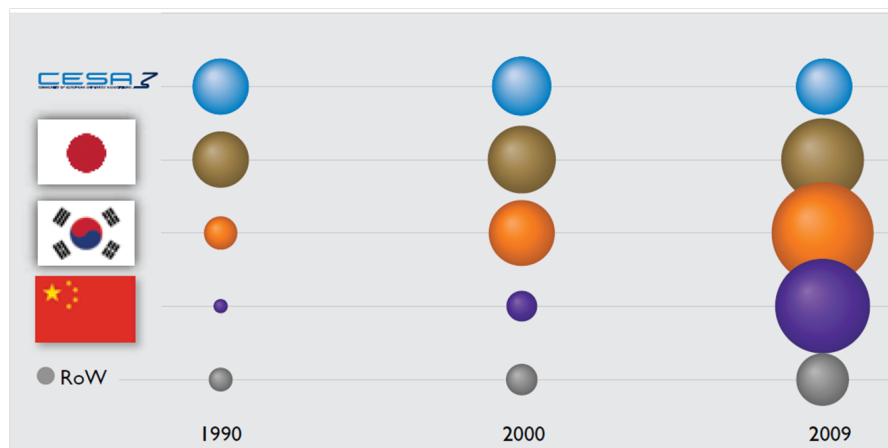


Figure 2: Massive capacities built up cause huge imbalance between capacity and demand (Production shown in CGT) – CESA (2010). The colors are related to the areas given by the flags. RoW stand for the Rest of the World.

2.2 Asia

2.2.1 Japan

After the economic crisis in 2008, the worldwide shipbuilding industry is operating in an increasingly competitive environment, and Japan is no exception, although Japanese shipyards produced more than 20 000 000 GT, a record high, in the year 2010. Under these rapidly changing circumstances, it is increasingly important to reduce shipbuilding costs by increasing productivity. In addition, the ship production process is affected by various international regulations to enhance safety and to protect the environment, requiring overall optimization of the production process. In response to these changes, Japanese shipyards are trying to make their production line more cost efficient and optimized, making use of advanced information technology, robotics, and so on. In this context, many achievements have been reported recently.

The application of laser arc hybrid welding usually requires considerable investment, and has not been widely introduced except for European yards building cruise ships made of thin steel plates. Koga *et al.* (2010) and Terada *et al.* (2010) extended this application to cargo vessels such as tankers, container ships, LNG and LPG carriers. By developing a simple welding carriage system and dispensing with large gantry crane equipments, the application of laser arc hybrid welding became practicable for various kinds of vessels. Currently the application is limited to butt welding of plates of less than or equal to 13 mm thickness, but its extension to fillet welding is also expected. This will result in considerable reduction in welding distortion, which, in combination with the application of laser cutting, 3D laser measurement, simulation of welding distortion and appropriate lifting plans for erection blocks, will lead to greater accuracy in the ship production process Yamamoto and Choshi (2010).

Miyazaki *et al.* (2009) has successfully applied the hydrogen gas cutting method, reducing the time required for cutting, preheating and piercing.

Line heating of steel plate to fabricate curved shell plating is one of the processes which is most labour intensive and difficult to learn, requiring a training period of more than 10 years. Automation of this process was reported by the committee in 2006, Borzecki *et al.* (2006). Based on this technology, Tango *et al.* (2011) has reported the application of an advanced fully automatic system, which was enabled through automatic evaluation of the distortion results, automatic corrective heating, automatic plate turn over and advanced robotics.

2.2.2 Korea

Since 2003, South Korea has become the world's top shipbuilding nation, but was surpassed by China in 2009 and 2010. Korean shipyards are making efforts to regain their world lead position. Tankers (17 million DWT), bulk carriers (16 million DWT) and container-ships (10 million DWT) still make up the largest share of deliveries, BRS (2010), but South Korea is focussing more on high-priced vessels and offshore facilities. Industry watchers consider that Korea is responding to the growing demand for technically-advanced ships with increased added value, such as drilling vessels and floating oil production facilities in response to the rapidly increasing oil prices. Whilst increasing their competitiveness in areas of offshore plants and high-value, specialized vessels, the companies are also investing heavily in alternative energy (e.g. Hyundai Heavy wind facility plant operation). Korean dockyards have been working to develop environmentally friendly shipbuilding technologies and 'green' vessels as the green wave reaches the global shipbuilding industry. Some Korean shipbuilders have

already developed hybrid ships that significantly cut carbon emissions and improve fuel efficiency. e.g. STX GD (Green Dream Project), ECO-Ship (STX Europe).

Hyundai Heavy Industry Co. Ltd. has established a method to control global bending distortion caused by the fabrication process for hatch-covers in a container ship. Lee *et al.* (2010b) measured the transitional behaviour of global bending distortion in the deck of a hatch-cover during fabrication by three dimensional measurement instruments. Ha and Yang (2010) have developed a modelling methodology by which global deformation after multi-pass welding can be analysed at the shell element level in one simulation.

In the field of welding automation, HHI (Hyundai Heavy Industry Co., Ltd.) and DSME (Daewoo Ship Building and Marine Engineering Co., Ltd.) announced the development of a corner-piece welding robot for LNG Carriers and a welding carriage with high deposit rate, respectively. Kim *et al.* (2010a) (HHI) has developed a Gas tungsten arc welding (GTA) robot system. They have verified the system and its performance through field testing on actual work pieces.

The welding position employed at the erection stage is usually the flat and vertical position. Application of submerged arc welding (SAW) and electrogas welding (EGW) for these positions makes it possible to achieve enhanced productivity and high quality. However, owing to their large size and weight it is difficult to apply these techniques in short and narrow regions. To overcome this problem, Kim *et al.* (2010c) (DSME) has developed a compact, lightweight, 4-axis welding carriage which perform 3D weaving.

Next to the developments in line heating systems mentioned in the Japan section, curved plates can also be manufactured using cold-forming techniques with a die system. However, the total number of curved plates with the same geometry is usually very small for ship structures. Therefore traditional fixed target surface machines are impractical. Paik *et al.* (2010) describe the concept of a changeable die system. In the publication the prediction of the spring-back characteristics of curved metal plates after cold-forming is discussed by means of the elastic-plastic large deformation finite element method. The algorithm provides accurate predictions of the spring-back deformation when compared to tests.

Hwang *et al.* (2010) and STX shipyard also have suggested a MPPF (multi-point press forming) that have a single side multipoint dieless tool for cold forming. They developed an integrated system for thick plate forming performed by the single side MPPF. To determine the piston strokes in multi-point forming from a set of scattered data points, the compensated position of each piston point was calculate by an integrated displacement compensation method which combines ICP (Iterative Closest Point) algorithm, DA (displacement adjustment), and FEA. DA was used to automatically calculate the spring-back compensation necessary for all hull plates. The DA method is incorporated into a commercial FE code through a batch-run interface to repeat the iterative compensation by the integrated system.

2.2.3 China

The Chinese shipbuilding has undergone significant expansion since 2000. Its order-book has increased from 10.6 million to 185 million DWT. Bulk carriers made up the largest proportion of yearly deliveries (41 million DWT), followed by tankers (15 million DWT) and container ships (4 million DWT), BRS (2010). Recently, China's shipbuilding has prospered in the building of supertankers, container vessels and engineering ships. In addition, there are new developments in China's equipment manufacture for ocean engineering.

But there are still some problems for shipbuilding in China. Firstly, shipbuilding costs are rising steadily. Labour costs have increased month by month, and according to most shipbuilding enterprises, the wages for workmen in China's shipbuilding industry at coastal areas by the end of December rose at a mean rate of 15 % compared with the beginning of the year. Secondly, the price of ship steel in 2010 has risen month by month, which increases the procurement costs of shipbuilding enterprises. Thirdly, international shipbuilding rules and regulations are renewed and updated more frequently, requiring China's shipbuilding enterprises to develop new hull forms and modern shipbuilding methods in order to comply.

2.3 Europe

European yards have been careful in their business development and have largely refrained from massive capacity expansions. The pursuit of numerous opportunities in specialised markets, which could be exploited through innovative solutions, have played a focal role for many years. With this approach, European yards have been able to double their turnover since 2005, whilst keeping the output in tonnage-terms stable. Thus, it is becoming more apparent that the focus on niche markets has placed the European yards as leaders in the building of complex hardware for a wide range of specialised maritime activities such as dredging, fishing, cruising and leisure, supply and support for harvesting offshore energies, research, environmental preservation, pollution control, etc. Despite this specialisation, the European yards continue to lose market share in shipbuilding production, CESA (2010).

The EU is confronted with twin challenges: sustainable growth and scarce natural resources. The maritime industries are pioneering the development of new markets with high growth potential, like wind and wave energy, food from the seas, pollution control, clean and safe transport of passengers and goods, deep-sea mining for minerals, etc. EU shipyards are now conducting research, development and innovation in order to adjust to the changed business environment and benefit from growth markets.

2.4 America

2.4.1 Brazil

According to Lloyd's Register, in 1980 Brazil was the world's 2nd largest shipbuilding nation behind Japan. However, the industry collapsed in the following decades due to local economic factors such as hyperinflation, high interest rates and the ending of state subsidies. By 1999, no ships over 100 tons were being built and the industry had shrunk to only 2000 workers nationwide, Paschoa (2010).

However, an amazing revival has occurred in the last decade in response to large deep-water offshore oil and gas discoveries. For political reasons, the Brazilian Government, through its state-sponsored oil company Petrobras and its shipping subsidiary Transpetro, have used these oil discoveries as a vehicle for job creation. Wherever possible the Brazilian government has required as many of the requisite vessels and oil rigs to be built within the country. This has resulted in a shipbuilding boom. Today, the industry has a national workforce of over 45 000 with approximately 80 booked orders for a variety of ships and rigs, França (2009) and Paschoa (2010). New developments in risers and anchorage systems are currently two important research topics, Andueza and Estefen (2011) and Rossi and Fernandes (2011).

3 FABRICATION TECHNOLOGY

3.1 Welding Thick Steel

Recently, various requirements for the use of thick steel plate in a number of industrial fields, including the shipbuilding industry, have been identified. Especially with the continual increases in marine transportation volumes on a global scale, the steel of container ships and LNG carriers has become thicker and thicker with the increased size of ships (An *et al.*, 2010), see Fig. 3. In the previous report, the introduction of YP460 steel plates (high tensile steel plates with the specified yield point of 460 N/mm^2) to reduce the maximum plate thickness was introduced. High-tensile strength steel has also been selected to meet the required structural strength in the joints of thick plates, Kim *et al.* (2010c), Funatsu *et al.* (2010), Kaneko *et al.* (2010).

However, the application of extremely thick steel plates raises additional issues, such as brittle fracture and fatigue strength, which still need to be addressed.

3.1.1 Welding of Extreme Thick Plates

Focusing on safety-related issues of extremely thick steel plate applied to large container ships, a national joint research project was organised by the Japan Ship Technology Research Association (JSTRA), and many research activities were carried out between 2007 and 2009, Sumi *et al.* (2010). This project tasked its 3 working groups with the following studies:

- Working group 1: Study of the arrest design of brittle crack propagation
- Working group 2: Study of prevention of brittle crack initiation
- Working group 3: Study of NDT technology for welding joints of extremely thick plates

In the activities of Working group 1, which are summarized by Yamaguchi *et al.* (2010), a new test method for brittle crack arrest toughness was established Kawabata *et al.* (2010), and a number of large-scale structural component model tests were carried out to simulate crack propagation and arrest from the hatch side coaming into the upper deck plating. In addition, the effects of structural discontinuities, in terms of welds Handa *et al.* (2010), full-scale structural component tests and ultra-wide duplex Esso tests were carried out, which confirmed the K_{ca} criterion of $6000 \text{ N/mm}^{3/2}$, Inoue *et al.* (2010). In addition, to ensure that the brittle crack does not propagate straight along the butt joint throughout the hull section, a butt shift of 300 mm in general is

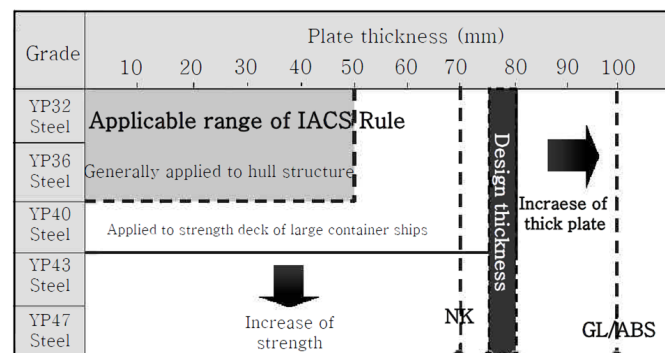


Figure 3: Recent history of thicker and stronger steel plates for large container ships
 – An *et al.* (2010)

Table 1: Recommendation to prevent brittle fracture accidents

	New ships	Existing ships
During new construction	Full length UT with allowable defect length of 25mm $K_{IC} \geq 3000 \text{ N/mm}^{3/2}$	Not applicable
After delivery	Full length UT every 10 years Visual inspection as far as practicable within the interval of not greater than 3 years	Full length UT after 10 years from delivery and subsequent every 5 years Visual inspection as far as practicable within the interval of not greater than 3 years

recommended between the hatch side coaming and the upper deck structure, based on the numerical simulations conducted under various conditions (Yoshinari and Aihara, 2009). All these findings were summarized in the Class NK “Guidelines on Brittle Crack Arrest Design” (Nippon Kaiji Kyokai 2009) as a guideline to ensure brittle crack arrestability for large container ships.

The brittle crack arrestability of ultra-thick plates has been also studied by An *et al.* (2010). In their study, crack arrest tests were conducted in order to investigate the crack arrestability of thick plates for shipbuilding steels, where test plate thicknesses were between 50 mm and 80 mm.

Working group 2 studied the prevention of brittle crack initiation. Fatigue crack growth from an embedded initial defect in the butt joint of the hatch side coaming was analysed. The critical crack size to prevent brittle fracture was identified for the purposes of determining allowable initial defect sizes for large container ships in combination with the results of Working group 3, which focused on the study of NDT technologies for welded joints of extremely thick plates. Several UT techniques were tested for accuracy by six Japanese shipyards, using test specimens with internal artificial defects.

All these results were summarized in the recommendation to prevent brittle fracture accidents, published by Association (2009). The recommendation proposes the control of the size of embedded defects in weld joints by UT, not only during new construction, but also after delivery, and also the control of the quality of weld joints to ensure that they exhibit sufficient brittle fracture toughness. Tab. 1 shows the recommendation for some typical trading patterns and conditions.

Shin *et al.* (2010) suggested a predictive equation for the prediction of the transverse residual stress at the thick FCA butt weldment of large container vessels. They used restraint degree, the yield strength of the base metal, the thickness of the weldment and welding heat input as the variables for the predictive equation. Restraint degree at the thick weldment of a container ship under different weld sequences was calculated by FEA in their study. On the basis of these results, an H-type specimen was designed to reproduce the level of restraint at the actual weldment of a container ship. Based on the FEA result, they proposed predictive equations for the mean value and the distortion of transverse residual stress at each location of the weldment using 3-D FEA and a multiple-regression method. The predictive equations were verified by comparison with those measured by XRD in the actual weldment of the container ship.

Lee *et al.* (2010c) suggest a finite element analysis (FEA) model to predict the residual stress in welded parts joined by FCA welding with more than 20 layers of weldment. The characteristics of residual stresses in FCW welds of high tensile strength steels

whose yield stresses were between 400 MPa and 500 MPa, respectively was investigated by both FEA and measurement. Three-dimensional thermal elastic-plastic analyses were conducted to investigate the welding residual stresses. EH40 and API2W-50 plates of 80 mm thickness were used as the base materials and a double 'V' butt joint configuration was used to join the plates. The joint for the specimen was welded with 28 layers. The residual stress was measured by X-Ray diffraction after the specimen was polished by chemical etching. The residual stress obtained by the FEA was also compared with that of experiment. Their study describes the 3-dimensional finite element model required to predict the welding residual stress in extremely thick plates of EH40 and API2W-50 joined by FCAW and discusses the comparison between the experimental results and numerical predictions of residual stress.

3.1.2 Thickness Effect to Fatigue Strength

Another problem associated with the application of extremely thick steel plate is fatigue strength. It is well known that increasing plate thickness causes a decrease in fatigue strength. Phenomena such as an expected larger stress concentration in the weld toe for thicker plates with identical weld profiles, larger stress gradients in thickness direction compared to thin plates especially in bending, possible larger residual stresses and increased probability of crack initiation due to larger areas of high stress can be causes for this decrease in fatigue strength. Some phenomena are already included in established fatigue rules and standards, such as DNV (2010), Hobbacher (1996) and IACS (International Association of Classification Societies) (2008). However, there are still some points which require further study, such as the thickness effects in extremely thick steel plate, thickness effects in actual ship structure details *etc.*

Polezhayeva and Badger (2009) studied the effect of plate thickness on fatigue strength through fatigue tests for the combinations of plate thicknesses 22 mm and 66 mm, in bending and tensile load, and for base material and butt welded joints. The authors proposed a thickness exponent of $z = 0.1$ for base material and $z = 0.2$ for butt welds, which is consistent with the IIW recommendations, Hobbacher (1996). On the other hand, some Japanese studies revealed far less of an effect of plate thickness in cases where the subject location for the fatigue strength assessment was just above a longitudinal supporting member, such as in the case of longitudinal stiffeners. Nakamura and Yamamoto (2007) carried out analytic research on stress concentrations in the welded joints of longitudinal stiffeners and web stiffeners, and concluded that the thickness effect is negligible. Fukuoka and Mochizuki (2010) carried out several fatigue tests and stress analyses, and also pointed out that the thickness effect for weld joints on I-section beams is much smaller than that of established rules and standards, and proposed new correction exponents. Im and Chang (2009) also investigated the fatigue strength of ultra-thick plates. Three types of joint, referred to as AW (As-Welded), UP (Ultra Peening) and TG (Toe Grinding), were cut from API 2W Grade 50 steel.

It is considered that the correction exponent included in the rules and standards are based on the results of small fatigue test specimen and not actual ship structural details. In addition, the mechanism for this effect is not yet clearly identified. Many factors are considered to influence the thickness effect, such as stress concentration factor, stress gradient in the thickness direction, residual stress due to welds and increased probability of crack initiation due to the larger area of high stress. Further study is considered to be necessary to reveal how each factor contributes to the thickness effect, and to establish reasonable and reliable thickness effect correction methods applicable to actual ship structural details.

3.1.3 Improvement of Fatigue Strength

In the committee's previous report, a new steel was introduced, called FCA (Fatigue Crack Arrestor) steel, which shows improved fatigue initiation life as well as improved crack growth life in welded structures. To maximise the benefit to be obtained from this steel, a new design S-N curve for welded cruciform joints made with FCA steel has been proposed, Konda *et al.* (2010). In this study, 66 small scale fatigue tests with FCA steel in association with 18 tests with conventional steel were conducted, supplemented by some large scale tests of relevant ship details. The proposed S-N curve has significantly longer life for the high cycle region of the S-N curve. As a result, the authors say that FCA steel is especially beneficial for details subjected to typical stress ranges from wave loading, leading to 3 times longer fatigue life, calculated for a typical long term stress range distribution.

Hara *et al.* applied FCA steel in the connecting areas between a cargo tank cover and the upper deck in the midship region of a 153 000 m^3 Moss type LNG carrier, and showed that FCA steel can be used without any scantling increase to relax or cancel the scope of weld toe grinding or to improve fatigue life, Hara *et al.* (2010). Takaoka (2010) presented examples of fatigue strength assessments from the application of FCA to crude oil carriers and bulk carriers. In this study, the fatigue life increased by 1.8 – 2.8 times on average.

In the previous report, the use of UIT (ultrasonic impact treatment) was reported as a very practical and effective method to enhance fatigue strength at welds. Takaoka (2010) studied the effective application of this method to ship structures, and found that its effectiveness is enhanced when UIT is carried out at locations under a tensile overall stress. They pointed out that such conditions occur when UIT is carried out on the weld joints on the upper deck of container and LNG carriers after launching. In addition, UIT is considered to be effective on accumulated fatigue damage, when it is carried out on aged vessels. IHI Marine United Inc. carried out UIT on an actual aged vessel and demonstrated its positive rehabilitation effect, Tango *et al.* (2011).

Due to the increase in the size of container ships and LNG carriers, and the use of higher strength and thicker steels, fatigue strength is becoming more of a concern. New developments such as FCA steel and UIT will help to improve the fatigue strength. However, in order to be able to fully take advantage of these developments, they need to be approved and included in rules and regulations.

3.2 Welding Aluminium

Aluminium is the material of choice for many ships and craft because of low weight, ease of fabrication and reasonable costs. However, welding of aluminium requires more joint preparation and cleanliness than is generally required for steel. Furthermore the need for shielding gas and the somewhat slower welding speeds make the process more expensive. Aluminium is more prone to distortion during welding than steel. A relatively new welding technique used for aluminium is Friction Stir Welding. It has found rapid application in the fabrication of structural panels. SSC-456 (2009) shows a comparison of the mechanical properties of friction stir welded and fusion welded aluminium plates. It was found that for a but-weld connection the tensile properties for friction stir welding were equal or better than for fusion welding. Also initial imperfections were smaller. For the compressive strength performance in the welded area friction stir welding is less good due to the occurrence of delaminations in the welded region, this could be resolved using a lap weld instead of a but weld.

When delaminations are prevented the ultimate compressive strength performance of the friction stir welding procedure is superior to fusion welding.

3.3 Corrosion Protection Techniques

Since 2008, research related to corrosion in ships and protection against corrosion has continued worldwide. Most of the efforts have been concentrated in three geographical regions, Australasia, North America, and Europe.

Most of the studies have focused in one of the following areas, a) corrosion behaviour, b) corrosion protection, and c) corrosion analysis. A summary of the developments will be presented in the following according to these three groupings.

3.3.1 Corrosion Behaviour

In terms of corrosion behaviour, studies in the last few years have included aluminium structures; differences between erosion-corrosion and corrosion; alternate alloys for the corrosion environment; elastomeric materials for bearings; anti-corrosive coatings; thermal spray coatings and direct metal deposition.

The specific strength of aluminium ships is higher than that of steel ships. Aluminium ships can travel at high speeds, have increased load capacities, increase ease of recycling, and have high anti-corrosion properties. For those reasons, the corrosion and mechanical properties of aluminium ships are continuously being developed. Kim *et al.* (2010b) reported on a number of electrochemical experiments undertaken to determine the optimum corrosion protection potential conditions for MIG welding to enhance protection for 5083-H116 (Al-Mg alloy) in natural seawater. For protection against stress corrosion cracking and hydrogen embrittlement in 5083-H116 aluminium alloys, the optimum corrosion protection range was reported as -1.2 to -0.7 V for the base metal and welds.

Zhao *et al.* (2008) studied the difference between synergistic erosion-corrosion and corrosion using a rotating disk apparatus and immersing mild steel specimens in a 0.05 wt. % SiC suspension. Techniques used in their study included scanning electron microscope (SEM), positron annihilation lifetime spectra (PALS) and X-ray photo-electron spectroscopy (XPS). The PALS results showed that both the size and number of vacancy defects depend on the cavitation and immersion time. Yet the size and number of vacancy clusters induced by cavitation erosion was much larger than that induced by corrosion damage alone. Reactions involving core level valence band electrons appeared to have led to greater oxidation of mild steel during cavitation erosion which was not observed in corrosion alone.

In an attempt to identify new materials for ship applications, titanium and super-austenitic stainless steels (SASS) were examined for marine diesel exhaust scrubbers operating with seawater as the used reactant that collects and neutralizes the sulphur dioxide of the exhaust gas. Aragon *et al.* (2009) conducted a series of corrosion resistance tests on welded samples of titanium and super-austenitic stainless steel alloys to evaluate their capacity in these applications. The authors concluded that super-austenitic alloys gave acceptable resistance characteristics in both parent metal and welded/HAZ areas, as long as there was no crevice in the test specimens. They also showed that high grade SASS (PRE > 40, Mo content > 6 %) could be a possible material provided that strict welding conditions are ensured and that the design of the scrubber be able to avoid any crevice configuration. Titanium samples were able to sustain the harshest corrosive conditions, including crevice geometry areas.

Elastomeric compounds, due to their favourable properties like sufficient hardness, toughness and natural resistance to abrasion and corrosion, are commonly used as bearing materials for the propeller shaft system of Indian Coast Guard Ships. They can be subject to unequal and non-uniform wear. Hirani and Verma (2009) analysed a sea-water lubricated journal bearing, investigating the actual geometric clearances of new and worn bearings recorded by the ship maintenance team and their effects, and duplicated in tests the operational data (load, speed and operating hours). Unplanned excessive radial clearance reduced the load capacity of the bearings and resulted in rapid and uneven wear. As such, bearing life can be enhanced by proper selection of radial clearance for the bearings.

3.3.2 Corrosion Protection

In terms of corrosion protection, recent studies have included cathodic protection, use of organic coatings, effects of surface preparation on epoxy coating performance effectiveness, effects of flash rust on the protective properties of organic coatings and active dehumidification during ship lay-ups.

To prevent salt-induced premature coating failure, International Maritime Organization (IMO) has adopted a Performance Standard for Protective Coatings (PSPC) specifying 50 mg/m^2 or less as the allowable NaCl limit for primary and secondary surface preparation for ship water ballast tank. Lee *et al.* (2010a) evaluated coating performance of epoxy coatings and established allowable soluble salt criteria, especially NaCl, in terms of adhesion strength and blistering resistance tests in immersion, condensation and cathodic protection environments. The authors recommended that the blistering resistance to soluble salt for each coating system be included in coating performance tests to verify and approve coating systems for IMO PSPC.

Since the implementation of the PSPC requirement in 2008, large numbers of vessels have already been delivered. Due to the preparation of hardware and software to fulfil the PSPC requirements, it seems that the shipbuilding industry is managing the stringent requirements effectively, although new facilities for blasting, painting and stock as well as increased labour time and enhanced quality management were necessary with the associated increases in cost, Seo (2010).

Based on actual experience, future revisions of the PSPC requirement have been proposed by the industry, e.g. replacing the requirement on surface roughness of “30 – 75 μm ” by “Medium” as defined in ISO8503-1/2, Lin (2010).

Amongst the many stringent requirements of PSPC, surface treatment using blasting is one of the most demanding factors, necessitating investment to the facilities as well as potentially creating a poor working environment and increasing industrial waste. As a promising alternative to this blasting, Yamagami (2010) established a new air mixed high-pressure water blasting technology, called Konki-Jet, which was demonstrated to fulfil the PSPC requirements.

Kim *et al.* (2008) examined the effect of the flash rust and surface roughness on the coating performance by evaluating adhesion forces and delamination areas through pull-off tests, visual inspection and electrochemical test. The rust layer on the substrate reduced the adhesion and accelerated the disbondment of epoxy coatings, but flash rust area ratios below 20 to 30 percent hardly affected the adhesion and performance of coatings. Anticorrosive pigments were observed to improve the barrier effect and protective performance of coatings. Brown (2010) proposed that dehumidifying systems should be used to control moisture on board and protect the ship from

corrosion during lay-ups. Dedicated dehumidification equipment is fundamental and effective in bringing air in the enclosed areas of the ship, such as ballast tanks, storage tanks, and control rooms, to a relative humidity not exceeding 50 %. According to Brown, desiccant type dehumidifiers are the most effective and efficient equipment for controlling moisture in large areas such as container ships.

Sorensen *et al.* (2009) prepared an extensive state-of-the-art review of the use of marine and protective coatings for anti-corrosive purposes. International and national legislation aimed at reducing the emission of volatile organic compounds (VOCs) have caused significant changes in the anti-corrosive coating industry. Meeting environmental regulations and reducing production costs remain a key challenge and a major driving force for new developments in anti-corrosive coatings. The authors pointed out that the next generation of high-performance anti-corrosive coatings face many challenges, and that the incomplete understanding of the physical and chemical mechanisms responsible for the failure of anti-corrosive coatings during service represents hindrances towards further progress. Thorough understanding and quantification of the degradation mechanisms by mathematical models may provide a useful tool in the development of new coating products and development of binders and pigments that may be capable of providing excellent protection against corrosion. Novel ideas which require further investigation and maturing include self-healing coatings.

Papavinasam *et al.* (2008) published an extensive review of the state-of-the-art of thermal spray coatings for corrosion protection. Thermal-spray coatings can be used in marine structures including offshore pipelines without external cathodic protection (CP). Al, Zn and ZnAl coatings protect steel by acting both as barrier coatings and as sacrificial anodes at local defects where corrosion could occur. Well-bonded, relatively dense, sealed coatings have the ability to provide effective long term corrosion protection (10-20 years), with minimum periodic maintenance. Practical examples of thermal-spray coatings can be found in Europe and in North America for corrosion protection of steel in urban, industrial, and marine environments, e.g over 40 bridges in Britain and the mile-long Pierre-Laporte suspension bridge near Quebec City.

In an attempt to deposit a corrosion resistant coating on a C71500 (70Cu-30Ni) alloy for marine applications, Direct Metal Deposition (DMD) technology using a CO₂ laser was developed. Bhattacharya *et al.* (2011) reported that a Cu-30Ni alloy (with a similar composition to the substrate) was successfully laser deposited on a rolled C71500 plate substrate. The Cu-30Ni clad specimen showed higher ultimate tensile strength but lower yield strength and percentage elongation than the C71500 substrate. The corrosion resistance of a DMD Cu-30Ni clad specimen was found to be lower than the C71500 substrate, but was found to improve in the case of the DMD Cu-30Ni clad/C71500 substrate specimen. The higher corrosion rate of the DMD Cu-30Ni clad specimens was attributed to the presence of porosity in the clad layers.

3.3.3 Corrosion Analysis

In terms of corrosion analysis, recent studies included several different approaches. The most relevant trend appears to be the development of models that are operational based, taking into consideration long term operations with cumulative damage and operational environment (region of navigation, sea water, temperature, etc.). Corrosion behaviour according to time-varying ultimate strength and strain rate has been investigated. Reliability of structural performance was examined under structural health monitoring. Advanced techniques such as the non contact EMAT measurement of aluminium alloy has proven to be a viable technique for characterising cor-

rosion and sensitisation. Finally, electrochemical impedance spectroscopy (EIS) also demonstrated good results in monitoring damage to organic coatings.

Guo *et al.* (2008) presented a semi-probabilistic approach to assess the time-varying ultimate strength of the deck plate of an aging tanker considering corrosion wastage. The authors proposed a non-linear corrosion model for deriving the time-varying probability density function of corrosion wastage of the deck plates. The model was validated using data from a total of nine sample tankers, designed in the 1970s, 1980s and 1990s. The results demonstrated that this procedure can be easily applied to assist the risk-based inspection.

When sensitized, 5XXX grade aluminum alloys are more susceptible to inter-granular corrosion (IGC) and inter-granular stress-corrosion cracking (IGSCC). The formation and growth of beta phase (Mg_2Al_3) along the grain boundaries is responsible for the susceptibility of these alloys. Conventionally, the degree of sensitization (DoS) is quantified by the ASTM G67 Nitric Acid Mass Loss Test, which is destructive and time consuming. Li *et al.* (2011) experimented with an electromagnetic acoustic transducer (EMAT) to measure the DoS in AA5083 aluminium alloy samples sensitized at $100^\circ C$ with processing times varying from 7 days to 30 days. Correlations between DoS and shear wave velocity, as well as shear wave attenuation allowed for easy DoS characterization in AA5XXX aluminium alloys. The authors successfully used EMAT ultrasonic measurements to discriminate low (5 mg/cm^2), medium (30 mg/cm^2), and high (60 mg/cm^2) levels of DoS for planar samples with accuracy about 90 %.

Melchers and Paik (2009) conducted laboratory experiments subject to pre-existing rusts to high levels of tensile strain to examine the effect of tensile strain on the rate of marine corrosion of steel plates. They exposed the steel to a natural marine environment, including the atmospheric and tidal zones. It can be concluded that low levels of tensile strains applied to previously corroded steel specimens have a relatively minor influence on the loss of adherent rusts from the surface of the metal and practically negligible effect on steel mass loss due to marine immersion corrosion. On the other hand, high levels of strain, approaching the yield strain of the steel, produced observable losses of adherent rusts and observable cracking of the adherent rust layer. Short-term marine immersion exposure tests of samples under these conditions experienced increases in mass loss due to corrosion by some 10-15 %.

Okasha *et al.* (2010) presented in their paper an approach for integrating the data obtained from structural health monitoring (SHM) in the life-cycle performance assessment of ship structures under uncertainty. Lifecycle performance of the ship structure is quantified in terms of the reliability with respect to first and ultimate failure and system redundancy. Structural Health Monitoring data obtained by testing a scaled model of a Joint High-speed Sealift Ship representing the worst operational conditions of sea state 7.35 knot speed and head seas were used to update its life-cycle reliability and redundancy. The results obtained showed that the dynamic load effects can be significant in rough operational conditions.

Regarding specific environmental effects on corrosion and corrosion wastage of ship structures, Guedes Soares *et al.* (2009) developed a new corrosion wastage model based on a reference non-linear time-dependent corrosion model by including the effects of different environmental factors such as relative humidity, chlorides and temperature. The inclusion of these factors that are especially relevant to the rates of corrosion in marine environment allows for more accurate predictions of the expected corrosion levels and better planning of the corrosion inspections along the life of the ships. The

paper proposed equations that serve as guide to ship owners and Classification Societies about which variables need to be monitored to allow more accurate predictions of corrosion wastage in marine atmosphere. The authors highlight that it is necessary to put in place monitoring programs to produce and collect the required data for validation of the proposed model in the long-term.

Combining the Support Vector Regression (SVR) approach with the Particle Swarm Optimization (PSO) method, Wen *et al.* (2009) established a model for prediction of the corrosion rate of steel under five different seawater environment factors, including temperature, dissolved oxygen, salinity, pH value and oxidation-reduction potential. The major objective of the paper was to compare between this method and back-propagation neural network (BPNN). The results illustrate that the predicted errors of SVR models are smaller than those of BPNN models for the identical training and test dataset, and the generalization ability of the SVR model is also superior to that of the BPNN model. The SVR does exhibit more limited extrapolation ability than the other methods.

Yan *et al.* (2009) used electrochemical impedance spectroscopy (EIS) combined with open circuit potential measurements and scanning electron microscopy (SEM) to characterise the corrosion process and products of two commonly used ship coatings, epoxy aluminium coating and chloride rubber iron red coating, and their composite coatings. These systems were immersed in 3.5 % NaCl solution. The authors demonstrated using potential-time measurement that the free corrosion potential of these three coatings with immersion time are more positive than that of metal substrate. The results also showed that the growth of the electrochemical area of the anode and corrosion takes place continuously. EIS results showed that corrosive species can penetrate into coatings and reach the coating/substrate interface rather quickly, causing the coatings to lose their shielding role and initiate the start of electrochemical corrosion. The penetration of the corrosive media also caused damage to the coatings by destroying the intermolecular cross linkage and rendering the coatings coarse, porous and brittle. Composite coatings exhibited synergistic behaviour between the two coatings offering better protection performance, demonstrating the effect of $1 + 1 > 2$ according to the authors. The authors also concluded that electrochemical tests along with surface analysis are adequate tools for studying the corrosive behaviour of organic coatings-substrate systems and for assessing their performance in corrosion environments.

4 COMPOSITE MATERIALS AND THEIR PRACTICAL APPLICATION

In recent years the application of composite lightweight materials has increased, both in the application of full composite ships and in a combination of composite parts with steel hulls. Composites have several advantages such as a light weight, large freedom of shape and corrosion resistance, which make them very suitable for application in maritime environments. Examples of the use of composites are e.g.:

- The hybrid high speed vessel midfoil (Navantek ltd)
- The ship hull of the patrol boats KNM Skjld of the Royal Norwegian Navy
- Mine counter measure vehicles (MCMV lansort Class) of the Royal Swedish Navy
- Visby corvettes (Royal Swedish Navy)
- Superstructure of the la Fayette class frigate (French Navy)
- Advanced enclosed mast/sensor (US Navy)

As can be seen from this list, most applications are in military ships. Commercial ship application is still lagging behind. Especially for the combination of composites and

metals, research on the connection between the two is being conducted (§4.1), as well as research on connections within the composites itself.

The main technical drivers for the application of composites are corrosion resistance, Bergen and Needham (2009), reduced costs and weight saving and magnetic signature, Faraday (2008). The reduced costs can only be seen when through life costs are taken into account. Initial front costs for composite materials are higher. Since most owners/operators are still front cost driven, this is not an advantage that is generally considered. The main advantage of weight saving that is normally seen is the increased stability or payload increase. Fuel saving due to weight reduction is normally stated as less important.

The major drawback for the application of composites is the SOLAS fire regulations. A lot of research is being conducted on how to provide and prove equivalent safety for fire (§4.2). Another large application of composites is in the repair of steel structures, mostly to increase the fatigue life. Details on research in this area will be discussed in §4.3. Now that the use of composites is increasing, attention is also starting to focus on optimisation of the design (§4.4)

4.1 Connections

From earlier studies it was seen that the connection between metal parts and composites can be a problem due to stiffness differences. Both experimental work and modelling techniques have been considered, Kabche *et al.* (2007); Boyd *et al.* (2008). Next to studying the behaviour of certain designs, also optimization of the joint design is a major research subject. The effect of variations in core thickness, laminate thickness, materials, overlaps and adhesives is also being studied, not only for metal-composite connections, but also for full composite connections, Song *et al.* (2008); Bella *et al.* (2010). In Boyd *et al.* (2008) a genetic algorithm, based on natural selection and genetics, is used for the optimisation of the connection of a composite superstructure to a steel hull used in the La Fayette frigate. Based on several criteria on strength, stiffness and weight the best performing designs are interlinked to create a new family of designs. Several optimisation rounds are conducted.

Detection of damage in the joints can be difficult. Palaniappan *et al.* (2008) discussed the use of fibre Bragg grating optical techniques to detect disbonds in composite bonded constructions. Cracks in the bond are detected by strain changes in the sensors. Noise levels are still a problem in this technique.

4.2 Fire Safety

The main concern for using composites in large load-bearing structures has to do with the fire performance. The Solas regulations are mainly based on metals. Other materials can be used providing equivalent safety is demonstrated. A lot of research has been going on the fire safety of composites. In contrast to metals a lot of processes play a role in the fire behaviour. A combination of thermal, chemical, physical and failure processes occur, with interactions between these processes. A good overview of the work done so far on the modelling of all these processes is given in Mouritz *et al.* (2009). The authors report on the recent modelling techniques and on the limitations of the models. Often the thermal analyses is decoupled from the fire/composite interaction, which is a simplification, since the composite material degenerates in the fire and, for example, the gases that are released in this process can ignite, thereby influencing the fire behaviour. A lot of research is focused on none-reactive (e.g. glass fibre) composites. Very little research has been conducted up to now on reactive fibre

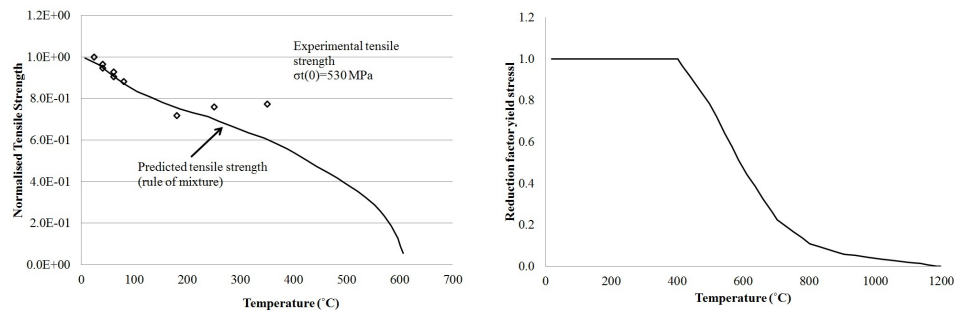


Figure 4: Strength decrease at elevated temperatures, left composite laminate, right steel Grade S275 (EN 1993 1-2) – Feih *et al.* (2007)

response. A lot of progress is reported on the modelling of the change in mechanical behaviour of composites under compressive loading subjected to a one-sided fire, Asaro *et al.* (2009). Gu *et al.* (2009) provide design diagrams and a quantitative methodology for fire protection design. In Asaro *et al.* (2009) it is shown that for moderate load levels the rate of degradation largely influences the panel response and not the temperature. For high load levels (close to normal failure loads) the temperature of the fire is the dominating factor.

Much less research has been conducted on composites in tensile loading conditions. This is because the softening behaviour in that case is far more difficult to model. However Feih *et al.* (2007) does address that issue. Thermo-mechanical models are described in this paper which predict time to failure in polymer laminates loaded in tension or compression subjected to one-sided fire. The mechanical models are based on the two-layer approach, in which part of the composite is not influenced by the fire (the virgin layer) and part (the charred layer) does have degenerate mechanical properties due to the fire interaction (see Figure 4). Limited verification with experiments is presented. It was concluded that compressive loading leads to earlier failure than tensile loading, and time to failure decreases with increased heat flux or mechanical loading.

4.3 Composite Patch Repair

Normally when a crack is detected in a metal structure it is repaired via welding or replacement of part of the structure. However, when hot-works are not allowed, welding is not an option. Also replacement can be a time consuming, costly option, since not all spare parts are available in-situ. Repair of corroded or cracked parts by composite patches is a good alternative. Composite patches are corrosion resistant; they prevent crack growth, can work as crack arresters, lower the stress concentrations and extend the life time of the structure. Furthermore the lightweight composite patches do not add much additional weight. Applications so far are mainly in aircraft. The application in bridges and other civil engineering structures is increasing. Maritime applications are still limited, although the advantages in offshore applications are becoming clear. A new FP7 EU project is started in January 2010 on composite patch repair for marine and civil engineering infrastructure applications (www.co-patch.com). Much research has been done on the fatigue behaviour of cracked steel plates, repaired by one-sided composite patches, Xiong and Shenoi (2008); Tsouvalis *et al.* (2009); McGeorge *et al.* (2009). It is seen that the patches can effectively slow down crack growth and extend the specimen life time. McGeorge *et al.* (2009) describes a recommended practice for patch repairs on floating offshore units. Xiong and Shenoi (2008) combine the research

on effects on the fatigue life of cracked metals with investigation of the static strength of the repaired specimen. No significant effect on the static strength was found with different patches. A large influence of patch thickness and material was seen in the fatigue life results.

4.4 Optimisation of Composite Design

Now that the use of composites is increasing, people are also looking into optimizing the design. This can take the form of optimization of certain parameters of composite structures, such as the use of shear keys to improve the shear behaviour of sandwich materials Mitra (2010). Other authors however, describe optimisation techniques to improve the overall behaviour of a structure, based on a combined optimisation of strength, stiffness and weight, Eamon and Rais-Rohani (2009).

Sriramula and Chryssanthopoulos (2009) address the uncertainties that are often result from fabrication and production of FRP composites. A deterministic approach often leads to severe over dimensioning, thereby diminishing the advantages of composites. The paper gives an overview of stochastic models, mostly with some validation, at both micro, meso and macro level.

Since the use of composites is still only limited and relatively recent, not much real time data is available yet on the expected life time in extreme service conditions. Miyano *et al.* (2008) present a new method based on temperature/time superposition for testing of composite responses. Tests are performed at a higher temperature at a higher loading rate to obtain long term prediction of the materials behaviour.

4.5 Recycling and Scrapping of Composite Materials

Patel (2010) forecasts that in 2040, 380 000 tons of fibre reinforced composites have to be disposed of each year. Recycling the material will be necessary to cope with this amount of disposal, both from an environmental point of view as well as from an economical point of view due to material scarcity. Due to the economical impact, more effort is being put into recycling of carbon fibre composites even though glass fibre composites production volumes are much larger.

Both mechanical, thermal and chemical recycling have been studied in the last few years. Pickering (2006) gives an overview of some of the methods investigated for thermoset composites (see Fig. 5). Thermoplastic composites can be reshaped again by heating and are not discussed here.

Most methods result in fibre shortening and a degradation of properties. Values of 60 % reduction in tensile and interface strengths have been found by Palmer (2009)

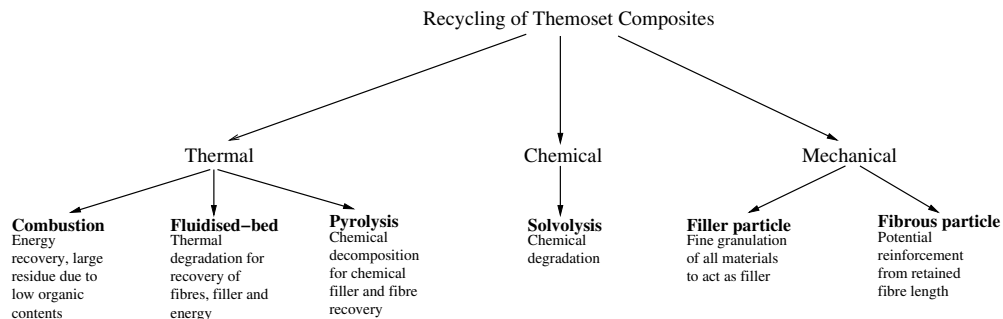


Figure 5: Recycling methods for thermoset composites – Pickering (2006)

for glass fibres, Job (2010) states values varying from 20-30 % for the fibre strength and 50% of the bonding strength for glass fibre composites. Carbon fibres show less decrease in strength, retaining up to 90% of the original properties. Nottingham university Piñero-Hernanza *et al.* (2008) has succeeded in recovering long carbon fibres with limited degradation (85-99 % of original properties) by chemical recycling using sub-critical and supercritical alcohols.

4.6 Metal Sandwich Materials

The main application of metallic/hybrid sandwich materials in ship structure is still in the production of different superstructure parts (vehicle decks, short deck panels, balcony, stairways, hatch cover, etc.) mainly exposed to local loading, Zanic *et al.* (2009); Kortenoeven *et al.* (2008). Inclusion of sandwich panels as the part of global hull girder bending carrying structure is under investigation, Romanoff *et al.* (2010).

Roland *et al.* (2006) present an overview of EU research projects dedicated to lightweight structures, their manufacturing and integration into the ship structure and summarise available solutions and trends. Feraris *et al.* (2006) give a brief review regarding the opportunities for the use of lightweight metallic structures in large high speed vessels. State of the art production technology and the possibilities offered by the new generation of High Strength Low Alloy steels and aluminium alloys have been discussed. Bohlmann (2006) presented experiences of using extra high tensile steel HT69 related to the design, construction and fabrication of large high speed craft. Reinert and Sobotka (2006) presented experiences in the development of all metallic I-core sandwich panels dedicated for inland waterway cruise ships.

The improvement of the strength-stiffness-weight characteristics of sandwich panels is continuously in progress. Structural optimisation of laser welded sandwich panels with adhesively bonded cores is presented by Kolster and Wennhage (2009), while Barkanov (2006) presented a methodology for optimal design of large vehicle decks and stair modules made of laser welded sandwich. Improvement of the shear properties of web core sandwich panel structures using different filling material is presented by Romanoff *et al.* (2009).

Thin steel sandwich panels need to be joined to one another and to conventional structures. The main focus in recent research has been on design and testing of appropriate joint shapes with respect to technological limits and strength requirements, Polic *et al.* (2011). Frank *et al.* (2011) investigated the influence of different laser welded T-joints geometries on fatigue strength.

5 STANDARDS

5.1 Standards Comparison

Okumoto *et al.* (2009) points out that fatigue cracks are influenced by structural stress concentration, construction tolerances, alignment, welding bead shape, as well as the exerted stress range and residual stress. In an actual ship structure, some construction deviation such as thin horse distortion and misalignment is inevitable. Such construction deviations are controlled under construction standards such as JSQS, and it is considered that strength is warranted by the feedback from actual structural damage of ships in service.

Although a long history of shipbuilding proves that this system has worked, simple standards such as JSQS do not accurately take into account the influence of design variations such as a wider application of higher tensile strength steel leading to increased

nominal stress, or different structural configurations. By quantitatively evaluating the influence of construction tolerances, the quality in terms of fatigue strength can be enhanced.

Actually, such design is required to some extent in the field of gas carriers. Comprehensive application of such design methodology can be found in Adhia *et al.* (1997).

On the other hand, some requirements of construction standards do not seem necessary in terms of strength, and have arisen as a result of empirical as-built records or aesthetic reasons. The unclear origin of such requirements causes confusion when applied to novel designs.

Comparison of standards was conducted in this committee in the 1997 term. After 15 years, a similar comparison was attempted. Results of the comparison of some typical attributes are summarized in Tab. 2. As a whole, no significant changes are observed. However, the following discussion points are raised with respect to the following items.

5.1.1 Distance Between Welds

IACS etc. allows no restriction on the distance between butt and fillet welds or scallops if it is not a strength member, whereas VSM does not. Taking account advances in materials and welding methods, this restriction is considered to be unnecessary even for strength members. Research into the effect of lapped welds on the material strength, including fracture toughness, is expected to reasonably abolish this requirement, or at least result in its application only to certain material chemical compositions.

5.1.2 Fairness of Frames

Required tolerances with regard to fairness of frames are stipulated in several construction standards, based on actual experiences of ship construction. However, as to the distortion of primary supporting members, it is not clear which requirements should be applicable. Especially, as to ferryboats and car carrier decks, whose girders usually have a long span, support small loads, and therefore have small scantlings and relatively large initial deformation, the requirements are critical for their efficient construction.

From this viewpoint, the newest revision of JSQS (JASNAOE, 2010) has introduced tentative requirements for the distortion of car deck transverses as Appendix-1. Somewhat relaxed requirements are stipulated for vertical distortion of deck transverse (standard range of $3L/1000$ to tolerance limits of $4L/1000$), lateral distortion of deck transverse (standard range of $3 + 2L/1000$, max 12 to tolerance limits of $6 + 2L/1000$, max 15), and distortion of web plate (tolerance limits of t). According to its technical background, it is confirmed through detailed structural analysis that even under the maximum distortion of tolerance limit, structural strength is not deteriorated and the distortion will not increase after delivery. These tentative requirements will be re-evaluated at the next revision of JSQS.

Table 2: Standards comparison – Part 1

Attribute	IACS	JSQS	CSBC	VSM	IRCN
Alignment					
Butt welds	Strength members: Limit: $a \leq 0.15t$, $max = 4\text{ mm}$; Other members: Limit: $a \leq 0.2t$, $max = 4\text{ mm}$	Strength members: Limit: $a \leq 0.15t$, $max = 3\text{ mm}$; Other members: Limit: $a \leq 0.2t$, $max = 3\text{ mm}$	Skin plates and longitudinal members: Limit: $a \leq 0.15t$, $max = 3\text{ mm}$; Bulkhead plates and interior members: Limit: $a \leq 0.2t$, $max = 3\text{ mm}$	Class. level B: Limit: $a \leq 0.1t$, $max = 3\text{ mm}$; Class. level C: Limit: $a \leq 0.15t$, $max = 4\text{ mm}$; Class. level D: Limit: $a \leq 0.25t$, $max = 5\text{ mm}$	Plating: Limit: $a \leq 0.15t$, $max = 4\text{ mm}$; Flanges: Limit: $a \leq 0.2t$, $max = 4\text{ mm}$
Cruciform fillet welds	Strength member and higher stress member: Limit: $a \leq t/3$; Other members: Limit: $a \leq t/2$ when the thickness of shlef plate (t_3) is smaller, t_3 should be substituted.	Strength member: Limit: $a \leq t/3$; Others: Standard: $a \leq t/3$; Limit: $a \leq t/2$	Longitudinal members within $0.6L$ and principal transverse supporting members: Limit: $a \leq t/3$; Others: Limit: $a \leq t/2$	Limit: $a \leq t/2$ when the thickness of shelf plate (t_3) is greater, t_3 may be substituted.	Limit: $a \leq t/3$ where t is the thicker of the two
Distance between welds					
Between butt welds on one plane	For cut-outs: Standard: $d \geq 30\text{ mm}$; For margin plates: Standard: $d \geq 300\text{ mm}$, Limit: $d \geq 150\text{ mm}$	Limit: $d \geq 30\text{ mm}$	Limit: $d \geq 30\text{ mm}$	Limit: $d \geq 50\text{ mm} + 4t$	Not specified
Between butt welds on two crossing plane	No restriction	No restriction	No restriction	No restriction (optional for satisfactory welding)	Not specified
Between butt weld and fillet weld	Strength member: Limit: $d \geq 10\text{ mm}$; Others: no restriction	Main structure: Limit: $d \geq 10\text{ mm}$; Others: no restriction	Main structure: Limit: $d \geq 10\text{ mm}$; Others: no restriction	For fillet weld first: Limit: $d \geq 30\text{ mm} + 2t$; For butt weld first: Limit: $d \geq 10\text{ mm}$	Not specified
Scallops over weld seams	Strength member: Limit: $d \geq 5\text{ mm}$; Others: no restriction	Main structure: Limit: $d \geq 5\text{ mm}$; Others: no restriction	Limit: $d \geq 5\text{ mm}$	Limit: $d \geq 10\text{ mm}$	Not specified

Table 3: Standards comparison – Part 2

Attribute	IACS	JSQS	CSBC	VSM	IRCN
Weld Gap					
Single vee butt	Standard: $G \leq 3 \text{ mm}$, Limit: $G \leq 5 \text{ mm}$; For $5 \text{ mm} < G \leq 1.5t$, $\max = 25 \text{ mm}$, weld up gap; For $G > 25 \text{ mm}$ or $1.5t$ whichever is smaller, partial renew	Standard: $G \leq 3.5 \text{ mm}$, Limit: $G \leq 5 \text{ mm}$; For $5 \text{ mm} < G \leq 16 \text{ mm}$, weld up gap; For $16 \text{ mm} < G \leq 25 \text{ mm}$, weld up with edge preparation or partial renew; For $G > 25 \text{ mm}$, partial renew	Limit: $G \leq 5 \text{ mm}$; For $5 \text{ mm} < G \leq 25 \text{ mm}$, weld up gap; For $G > 25 \text{ mm}$, partial renew	For $JWPS < G \leq t$, $\max = 30 \text{ mm}$, weld up gap	For $JWPS < G \leq 25 \text{ mm}$, weld up gap; For $G > 25 \text{ mm}$, partial renew
Butt weld insert plate size	Limit: $w \geq 300 \text{ mm}$	Limit: $w \geq 300 \text{ mm}$	Limit: $w \geq 300 \text{ mm}$	not specified	Limit: $w \geq 100 \text{ mm}$ or $10t$
Fillet weld	Standard: $G \leq 2 \text{ mm}$, Limit: $G \leq 3 \text{ mm}$	Standard: $G \leq 2 \text{ mm}$, Limit: $G \leq 3 \text{ mm}$	Limit: $G \leq 2 \text{ mm}$	Class. level B: Limit weld throat: $G \leq 0.5 \text{ mm} + 0.1x$, $\max = 2 \text{ mm}$; Class. level C: Limit weld throat: $G \leq 0.5 \text{ mm} + 0.2x$, $\max = 3 \text{ mm}$; Class. level D: Limit weld throat: $G \leq 0.5 \text{ mm} + 0.3x$, $\max = 4 \text{ mm}$	Limit: $G \leq 3 \text{ mm}$
Fillet weld correction	For $3 \text{ mm} < G \leq 5 \text{ mm}$, increase leg length by $(G - 2) \text{ mm}$; For $5 \text{ mm} < G \leq 16 \text{ mm}$ or $1.5t$, weld up; For $G > 16 \text{ mm}$ or $1.5t$, partial renew	For $3 \text{ mm} < G \leq 5 \text{ mm}$, increase leg length by $(G - 2) \text{ mm}$; For $5 \text{ mm} < G \leq 16 \text{ mm}$, weld up or liner treatment; For $G > 16 \text{ mm}$, partial renew	For $2 \text{ mm} < G \leq 5 \text{ mm}$, increase leg length by $(G - 2) \text{ mm}$; For $5 \text{ mm} < G \leq 16 \text{ mm}$, weld up or liner treatment; For $16 \text{ mm} < G \leq 25 \text{ mm}$, weld up; For $G > 25 \text{ mm}$, partial renew	For $G \leq 5 \text{ mm}$, increase leg length by $(G - 1) \text{ mm}$; For $G > 5 \text{ mm}$, weld up or liner treatment; For $G \gg t$, partial renew	For $3 \text{ mm} < G \leq 5 \text{ mm}$, increase throat by $G/2$; For $5 \text{ mm} < G \leq 25 \text{ mm}$ (main item) or 30 mm (secondary item), weld up; For $G \geq 15 \text{ mm}$ (main item) or 20 mm (secondary item), partial renew
Fillet weld insert plate size	Limit: $w \geq 300 \text{ mm}$	Limit: $w \geq 300 \text{ mm}$	Limit: $w \geq 300 \text{ mm}$	Standard: $w \geq 300 \text{ mm}$, Limit: $w \geq 150 \text{ mm}$	Limit: $w \geq 10 \text{ mm}$, $\min = 10t$

Table 4: Standards comparison – Part 3

Attribute	IACS	JSQS	CSBC	VSM	IRCN
Fairness of frames					
Shell plate	Parallel part: Standard: $2L/1000$, Limit: $3L/1000$; Fore and aft part: Standard: $3L/1000$, Limit: $4L/1000$	Parallel part: Standard: $2L/1000$, Limit: $3L/1000$; Fore and aft part: Standard: $3L/1000$, Limit: $4L/1000$	Parallel part: Limit: $3L/1000$; Fore and aft part: Limit: $4L/1000$	Limit: $0.2\sqrt{L}$	Not specified
Deck plate	Standard: $3L/1000$, Limit: $4L/1000$	Standard: $3L/1000$, Limit: $4L/1000$	Limit: $4L/1000$	Limit: $0.2\sqrt{L}$	Not specified
Inner bottom	Standard: $3L/1000$, Limit: $4L/1000$	Standard: $3L/1000$, Limit: $4L/1000$	Limit: $4L/1000$	Limit: $0.2\sqrt{L} + 3$	Not specified
Bulkhead	Limit: $5L/1000$	Standard: $4L/1000$, Limit: $5L/1000$	Limit: $5L/1000$	Limit: $0.2\sqrt{L} + 3$	Not specified
Accommodation	Standard: $5L/1000$, Limit: $6L/1000$	Deck: Standard: $3L/1000$, Limit: $4L/1000$; Outside wall: Standard: $2L/1000$, Limit: $3L/1000$	Not specified	Limit: $0.2\sqrt{L}$	Not specified
Others	Standard: $5L/1000$, Limit: $6L/1000$	Standard: $5L/1000$, Limit: $6L/1000$	Limit: $6L/1000$	Limit: $0.2\sqrt{L} + 3$	Not specified
Accuracy of dimensions					
Deviation of rudder from shaft C.L	Standard: 4 mm ; Limit: 8 mm	Standard: 4 mm ; Limit: 8 mm	Limit: 8 mm	Not specified	Not specified

5.1.3 Deviation of Rudder from Shaft Centreline

IACS, JSQS and CSBC require 8 mm deviation at maximum. But it seems that even if this maximum deviation is exceeded, no functions are deteriorated, and that it comes from rather actual as-built records. Actually, according to the technical background of JSQS (JASNAOE, 2010), this 8 mm limit was derived according to the histogram of as-built records of 48 ships. Considering that the original JSQS was published in 1964, the ships surveyed to determine this requirement might be built before 1960's. Ship's size and rudder types are totally different nowadays and the validity of this requirement is questionable. The background of the same requirement of IACS Rec.47 is not known. This kind of requirements causes confusion among surveyors and shipyards. Such requirements should be reconsidered, and may be better to be left to each shipyard's practice.

5.2 Effect on Structural Performances

Pretheesh *et al.* (2010) studied the effect of distortion on the buckling strength of stiffened panels through a parametric non-linear finite element analysis under an axial loading condition, and proposed a new strength parameter to represent buckling strength which takes into account the inelastic post-buckling behaviour of the structure.

6 LINKING DESIGN AND PRODUCTION IN COMPUTER APPLICATIONS FOR INCREASED EFFICIENCY

Throughout the engineering disciplines, many "Design for X" processes have been developed in order to correct the inadequacies of the designs during the ship design stages. DFX is the process of pro-actively designing products to optimise all the functions throughout the life of the product. This has been called "Design for X" where X is whatever the specific focus happens to be. So "Design for X paradigm" covers many areas such as Design for Production, Design for Manufacturing, Design for Assembly, Design to Cost, Design for Simplicity, Design for Maintenance, Design for environment, Design for Safety, Design for Life Cycle Cost, Design for Robustness, Design for Six Sigma, etc., Olcer *et al.* (2004); Papanikolaou *et al.* (2009).

The future challenge will be the multi objective optimisation of the ship and offshore structures (lean manufacturing, costs, safety, environment, etc.) to obtain a good synergy at a given time (see the report of the design method committee). Rapidly changing parameters such as the steel market prices make this optimisation in total time-frame design-build ship difficult.

6.1 Design for Production and Design for Manufacturing (DFP)

For most ships, productibility has become a major design attribute. If a ship cannot be manufactured or assembled efficiently, it is not properly designed. Any adjustment required after the design stage will result in a penalty of extra time or cost. Deficiencies in the design of a ship will influence the succeeding stages of production Larkins (2010); Olcer *et al.* (2004); Ou-Yang and Lin (1997); Papanikolaou *et al.* (2009); Storch *et al.* (2000); Bruce *et al.* (2006). There are two main principles for DFP for ships, namely:

1. all designs should drive for simplicity, and
2. all designs should be the most suitable given the shipyard facilities.

DFP in the context of shipbuilding can be understood as the following collection of principles and recommendations Larkins (2010); Caprace and Rigo (2010); Fanguy *et al.* (2008); Miroyannis (2006); Rodriguez Toro *et al.* (2004):

- Apply the ease of manufacturing: Designing for easy construction of parts, material processing and product assembly is a primary design consideration. Particularly if labour costs are a big percentage of the cost, problems in fabrication, processing and assembly can generate enormous costs, cause production delays, and demand the time of precious resources.
 - Avoid using thin plate to avoid distortions, reworking and straightening
 - Do not plan hull curvature into the structure (hull plating)
 - Eliminate cruiser sterns and cambered transoms
 - Maximize use of flat panels, straight frames, and reduce plate curvature
 - Simplify bow and stern shape by removing unnecessary curvature
 - Run strakes in the same direction as primary framing
 - Design for maximum use of high productivity tools such as automatic welding
 - Design bilge strakes with the same thickness as bottom plates
 - Make port side and starboard unit similar (symmetry)
 - Allow for large deck space to facilitate outfitting
 - Minimize lifting and handling of parts because it is labour intensive and non-value added
 - Minimize and optimize welding because it is the largest contributor to the total cost
 - Reduce the structural complexity
- Standardise as much as possible: Standardisation, as a means of reducing complexity and component variants, actually boosts the manufacturability of the product itself. It also increases the chances of automated assembly as it presents a repeated mode of assembly.
 - Minimise the number of parts
 - Standardise the parts to minimize the number of unique parts
 - Standardise the material and scantling types
- Use modularity wherever possible: Modular design Modular design or “modularity in design” is an approach that subdivides a ship into smaller parts (modules) that can be independently created. Besides reduction in cost (due to reduced customisation and learning time) and flexibility in design, modularity offers other benefits such as the reduction of lead time during production.
 - Design to facilitate assembly and erection with structural units, machinery units, and piping units

Several computer applications and prototypes that are using these principles are listed in the following section.

6.2 Computer Applications

Computer Integrated Manufacturing, CIM, systems for shipbuilding support the increase of productivity during the production stage by linking the design system with the production support system, Caprace and Rigo (2010). Many advanced CIM systems used in shipbuilding incorporate advanced production support systems. Such systems lead to improvements in the quality of production planning and scheduling, consequently enabling improved production flow. The systems also enable the introduction of automated facilities/robots by electronic data of the design information, Storch *et al.* (2000). The CIM software technology takes into account:

- Computer aided design (CAD)
- Computer aided manufacturing (CAM)

- Product data management (PDM)
- Enterprise resource planning (ERP)
- Computer aided process planning (CAPP)s
- Discrete Event Simulation (DES)

6.2.1 *Difficulties to Link Design and Production*

One of the major problems in the shipbuilding process today is the lack of interoperability between data systems and software applications. Ship data is complex and stored throughout multiple applications that do not automatically interface with each other. The data must then be manually integrated by gathering information from multiple sources and verifying individual results (Briggs *et al.*, 2005). Enterprise Resource Planning (ERP) systems have been developed for shipbuilding and are currently playing a major role in optimising resources in a shipyard's supply chain, value chain and information chain. These systems also link the necessary ship data to help decision makers retrieve information and make educated management decisions, Zhang and Liang (2006).

In order to overcome these issues, some authors are recently proposing solutions to improve the design and interoperability of production software.

A first example is provided by Borasch (2010). He proposed a digital method for outfitting called DigiMaus in order to link different IT-tools, design and construction applications such as CAD, ERP, project management software and visualization systems. This new tool has demonstrated a reduction of man-hours for prototypes or the first vessel in a series. One of the main functionalities is that the tool can show all outfitting components assigned to one selected production activity in only one GUI (steel, pipes, HVAC, engine, electricity, etc.). Moreover it is possible to check the current assembly state of the visualised area.

Similarly, Boesche (2010) developed a 3D-CAD catalogue to integrate 3D-Models of equipment directly into almost all used 3D-CAD Systems, in native format. Therefore shipyards are able to speed up their design for production earlier in the design process avoiding unscheduled problem during the production (interferences, bad accessibility, low maintainability, etc.).

Because many different participants and systems are involved in the very dynamic process with high modification rates, there is always a constant risk of errors being introduced in the different CIM software's. Koch (2010) recently developed a prototype to validate the engineering design and the production data based on data retrieval and rule-based analysis. This methodology has been shown to reduce the risk of costly errors often uncovered very late in the production process. Moreover, this rule based approach has the advantage of protecting customer know-how and provides a high degree of flexibility and sophistication.

6.2.2 *Linking CAD/CAM to Production*

The benefits of fully adopting CAD/CAM technologies have been proven in other industries as well as at smaller levels in the shipbuilding process. Three-dimensional (3D) modelling has been proven as the next necessary step in shipbuilding. One complete model can be developed and used by all designers and additions or modifications can be completed more effectively. Using the unified model, multiple optimisations can be completed prior to production. Benefits of using CAD/CAM technologies include the following Okumoto *et al.* (2006):

- Decreasing lead time: The time period from purchase order to delivery can be reduced.
- Effective production without backtracking: Trial and error work can be eliminated and manufacturing efficiency can increase. Labour cost is 30 % to 40 % of the total ship cost, but this can be reduced with effective production methods.
- Decreasing material cost: Material cost is 50 % to 60 % of the total ship cost. Simulations can be used to effectively optimize the model and reduce material costs.
- Non-skilled production: Skilled work can be replaced by systematisation and automation using information technology.

Most simulations have just been implemented in the design stage, mainly due to the complexity and experience required for ship production. There are major benefits in using simulation and early assessment in the ship production stage, as well as throughout the design. Simulations could be used to analyse and evaluate the production process, plan and assist with production, train workers, in skills such as welding, and confirm the safety of work operations. Implementation of applying CAD to ship production has been slow in the shipbuilding industry mainly due to the large cost associated with developing the models. Capabilities of computers and the cost reduction of CAD programs have made this technology very attractive for the shipbuilding industry, Okumoto *et al.* (2006); Okumoto (2009).

An example of making the design and production stages of shipbuilding more effective is the generation of real-time production indicators for the designers. Caprace (2010) proposed a fuzzy metric to assess the producibility of the straightening process during detailed design. Straightening is the process by which the welding distortions are reduced in order to improve the structure flatness for aesthetic or service reasons. This metric has been used in CAD software to compare the relative costs of different design alternatives of stiffened deck structures.

The same authors, Caprace and Rigo (2010), have more recently proposed a real time complexity indicator for practical ship design. The aim of this application, running on a CAD/CAM software, is to provide recommendations to designers in order to improve the design quality. In this approach the complexity indicators are made up of different components such as the shape complexity, the assembly complexity and the material complexity. Each of these components is computed with data coming from the 3D CAD model. Then they are gathered in only one indicator after calibration with real production data.

6.2.3 Optimization of Schedule, Flow and Resources

Shipbuilding production usually is a complicated process that requires a lot of individual planning due to its one-of-a-kind nature. Traditionally the planning activity is mostly an empirical procedure, but with the introduction of computerised systems such as linear programming, concurrent engineering, the critical path method (CPM), program evaluation and review techniques (PERT), Discrete Event Simulation (DES) and ERP systems particularly the administrative aspects have been covered increasingly well in an automated or semi-automated way, Das and Tejpal (2008).

Following the full-scale use of CAD systems a trend has developed towards using simulation systems that can model the physical and dynamic behaviour of products being designed. At the same time, various approaches have been made to apply simulation techniques to production planning and factory design problems. Many of

these systems focus on the generic description of processes or, more specifically, on logistics, manufacturing processes, or material flows through factories or warehouses.

Creating a simulation model based on generic process descriptions and properties takes a considerable effort and includes a wide range of potential configurations. This is acceptable for factory design and layout simulations for example, where the cost of creating the model accounts only for a small part of the total investment and where the involvement of trained experts can be easily afforded. However, when it comes to reflecting actual shipyard configurations and processes on a day to day basis (including capturing changes over time) this effort has been found to be quite high for production planners and engineers. Few shipyards in Europe and Korea (Steinhauer, 2010, 2011; Shin *et al.*, 2009) are able to afford specialists focusing on these tasks.

However, this issue can be addressed through the use of simplified scheduling models coupled with optimisation, Biman and Navin (2008); Moyst and Das (2008); Yamato *et al.* (2009); Wang *et al.* (2009); Dong *et al.* (2009). Some of these authors are using simple PERT methodology while others are coupling linear programming with an optimisation algorithm in order to solve the scheduling problem. A similar approach has been developed by Lödning *et al.* (2010) and Koch (2011) in order to simplify the creation of the production simulation models based on a rule based decision making module. Obviously these kinds of simplified methodologies are less expensive to implement and to maintain than complex DES models. It is especially convenient for small and medium size shipyards.

Outfitting and more specifically piping seem to have a renewed interest among researchers in recent years, probably because the assembly work of the pipe unit is currently carried out by experienced, skilled workers, using complicated two dimensional drawings. Wei and Nienhuis (2009); Li *et al.* (2009) developed an automatic schedule generation with the expectation of helping to reduce the on-site coordination and installation effort and increase the level of pre-outfitting, which reduces cost and lead-time. Both of the developments have been based on the Theory of Constraints (TOC).

Another major problem for tasks related to planning is the lack of precise product information required to execute reasonably reliable production simulation runs at early project stages. Trying to forecast production of a future project at an early point in time poses a problem due to unavailable or unstable design information. Therefore Steinhauer (2010) proposed a generic model for simulation data. This data model will cover the required data for production simulation in shipbuilding and it will be usable for other companies from maritime industries or related branches. The goal is to increase efficiency at shipyards already using simulation and to help other shipyards introducing simulation technology even if the required data is not available completely.

Beside these difficulties, the DES keeps the interest of many researchers around the world. Two categories of simulations can be defined:

- The layout planning related to shipyards under planning (Greenfield) or construction and shipyards that are making retrofitting or extension of existing workshops, Shin *et al.* (2009).
- Production planning related to shipyards in operation, Reyes *et al.* (2009); Pires *et al.* (2010); Pires and da Silva (2010).

To make the most use of the simulation, coupling the overall simulation of the steel construction stages with the outfitting simulation is expected to be far more effective

in improving the planning quality as well reducing the effort required in production planning and control. In 2006 SIMoFIT (Simulation of Outfitting in Shipbuilding and Civil Engineering, www.simofit.com) was founded as an interbranch cooperation between shipbuilding and civil engineering, Steinhauer (2007). Outfitting processes in the shipbuilding and the building industry bear a high resemblance to each other. The planners have to answer the same questions: how to find a practicable schedule with sufficiently utilised equipment and employees satisfying principal guidelines. In the inter-branch team of SIMoFIT methods for outfitting simulation are further developed and used in various fields.

Both the limited space available in shipyards and the growth in the size of blocks and sections force the shipyards to optimise the block splitting and the lock erection processes. Asok and Kazuhiro (2009) and Karottu *et al.* (2009) proposed systems for block splitting optimisation based on graph theory and respectively on a fuzzy logic and genetic algorithm. Similarly, Roh and Lee (2009) developed the block division method for dividing the structure into blocks using the relationships between the structural parts. A generation method for production material information is then developed that includes calculating the weights associated with each block. Finally, a simulation method for block erection is developed.

Complementary approaches have been developed by Seo *et al.* (2007) who focused on a process planning system using case-based reasoning (CBR) and theory of constraints (TOC) for block assembly in shipbuilding. Then Cha and Roh (2010) combined discrete event and discrete time simulation framework to support the block erection process in shipbuilding.

Further steps to production simulation would be the realisation of the virtual shipyard by the use of both production simulation and virtual reality. Nedess *et al.* (2009) and Lödning *et al.* (2010) presented virtual reality models for the shipbuilding industry ensuring a focus on better processes, increases in productivity and reduction in throughput time. These prototypes further support the finding and verification of assembly sequences by providing necessary information, e.g. about the next part to assemble. Also an automatic model preparation for collision control is procured.

7 CONCLUSIONS AND RECOMMENDATIONS

According to the increase in ship size of container ships and LNG carriers, thicker plating and new high tensile steels are becoming more widely utilised, causing more concern about fatigue strength. With regard to the thickness effect on fatigue strength, the mechanism is not yet clearly identified, and further study is considered to be necessary to establish reasonable and reliable thickness effect correction methods applicable to actual ship structural details. New developments to improve fatigue strength such as FCA steel and UIT have demonstrated their effectiveness through published papers. Rules and regulations should incorporate these new technologies.

Most of the research in the last years on the application of composite materials relates to fire resistance and recycling and scrapping. Composite patch repair is being increasingly used in many fields of application where hot works are not acceptable. With the increase in the use of composite materials, the demand for optimised composite design is also increasing.

It is obvious that DFP is critical to achieving a globally competitive shipbuilding business but the real question is how to apply DFP. The days of simplistic applications of DFP principles, such as minimisation of unique parts, are gone due to the scale

and complexity of the modern global shipbuilding business. The correct combination of shipbuilding technology, business process improvement, ERP technology, production simulation and advanced material technology, such as virtual reality will be the delimiters for future shipyards.

After 15 years, a comparison of construction quality standards has been conducted and no significant changes were observed as a whole. However, to promote more rational and effective quality control, research activities directed towards more strength oriented standards are recommended, as well as a study to remove the current standards, which have their origin in as-built records of very old ships, and to leave these requirements to the more rational practice of each shipyard.

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18th INTERNATIONAL SHIP AND
OFFSHORE STRUCTURES CONGRESS

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ROSTOCK, GERMANY

VOLUME 2



COMMITTEE V.4

OFFSHORE RENEWABLE ENERGY

COMMITTEE MANDATE

Concern for load analysis and structural design of offshore renewable energy devices. Attention shall be given to the interaction between the load and structural response of fixed and floating installations, taking due consideration of the stochastic nature of the ocean environment.

COMMITTEE MEMBERS

Chairman: Feargal P. Brennan
Jeffrey Falzarano
Zhen Gao
Einar Landet
Marc Le Boulluec
Chae Whan Rim
Jaideep Sirkar
Liping Sun
Hideyuki Suzuki
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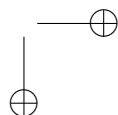
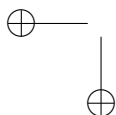
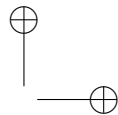
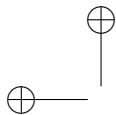
KEYWORDS

Offshore wind, marine energy, wave power, tidal energy, loading, design, testing, ocean current energy conversion (OCEC), ocean thermal energy conversion (OTEC).

18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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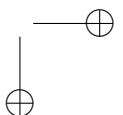
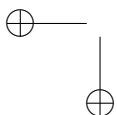
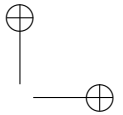
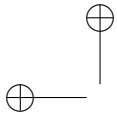
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1 INTRODUCTION

This report of the current committee describes recent activity of the international ship and offshore industry and the researchers that support it, with specific regard to current pertinent issues and trends relating to Offshore Renewable Energy. It is important to remember that the subject area is vast and developing rapidly, and that this report should be seen not only in the context of the entire ISSC 2012 proceedings but also as a continuation of past ISSC reports and complementing more generic IPCC (Lewis *et al.*, 2011, Wiser *et al.*, 2011) and other pertinent reviews. In addition, the committee chose to focus on areas commensurate with the expertise of the committee members to build on the vast knowledge base generated by previous Committee Two V.4 reports (ISSC, 2006, 2009). With this in mind, the present report has focused on offshore wind, which is by far the most technically and commercially developed of all the offshore renewable energy technologies.

In addition to a significant consideration of offshore wind power, this report, whilst updating developments in wave and tidal power, introduces ocean current energy conversion (OCEC) and ocean thermal energy conversion (OTEC), which have received much attention by IPCC and others but are further from commercial development, due mainly to the scale and investment required for concept demonstration.

An important element of ISSC work is to provide an expert opinion on the subject matter reported. Section 6 summarises the main features of the report and makes specific observations on the topics studied, particularly with respect to where further work is needed.

2 OFFSHORE WIND TURBINES

2.1 Summary of Current Activities

2.1.1 Commercial Installations and Wind Farms

Wind technology has come a long way in the past twelve years, from the first 220 kW offshore wind turbine that was built in Nordersund, Sweden in 1990, to the 1 GW London Array wind farm that was launched in March 2011 on the outer Thames estuary in the United Kingdom (Bilgili *et al.*, 2011).

According to the European Wind Energy Association (EWEA, 2011a), by the end of 2010, Belgium, Finland, Germany, Ireland, Norway, and Sweden had joined Denmark, the UK and the Netherlands, leading the global offshore wind capacity to 2,946.2 MW (approximately 0.3 % of the electricity demand in Europe). Major projects were: in Belgium, Belwind Phase 1 (165 MW); in Denmark, Nysted II/Rødsand II (207 MW); in Germany, Alpha Ventus (60 MW); in Sweden, Gässlingegrund (30 MW); and, in the UK, Robin Rigg (180 MW), Gunfleet Sands (172.6 MW), and Thanet (300 MW). The total installed offshore wind power in 2010 was 883 MW, a 51 % increase from 2009 (EWEA, 2011b).

In March 2007, the European Union set a target for 20 % of energy consumed across Europe to come from renewable sources by 2020. This challenging target would entail a total installed capacity of 40 GW of offshore wind power by 2020 and an average annual increase of 28 % (EWEA, 2010). For example, the UK would need to build 29 GW of offshore wind by 2020 to deliver its target of 15 %. This includes the Round 1 and Round 2 developments, with a total of 8 GW offshore wind power currently in operation or under construction, as well as Round 3, with another 21 GW to be installed starting in 2015 (Carbon Trust, 2008).

China's use of onshore wind power has increased rapidly in recent years, but its development of offshore wind farms has been relatively slow. Per April 2010, the offshore wind development pipeline stands at approximately 11.9 GW, and a total 650 MW capacity of wind power has been installed or is under construction off the coast of China (Qin *et al.*, 2010). The first commercial wind farm, Donghai-Bridge, was completed in February 2010, and is located 10 km offshore near Shanghai, with an average water depth of 10 m (Enslow, 2010). The wind farm consists of 34 3-MW Sinovel wind turbines. The support structure is based on a four-pile concept (Lin *et al.*, 2007).

So far, there is no offshore wind farm installed in Japan. However, the study by Ushiyama *et al.* (2010) suggested a roadmap with a long-term goal of installing 25 GW of offshore wind power by the year 2050. Under this plan, the country would begin construction of fixed offshore wind turbines in 2015 and of floating wind turbines in 2020. The aftermath of Fukushima has contributed renewed emphasis on this project. By the end of 2010, there were 822 monopile and 295 gravity-base wind turbine structures, out of a total of 1,136 wind turbines in the European offshore wind farms. Though steel monopile and concrete gravity structures were still widely used for water depths up to 25 m, wind farm projects involving jacket, tripile and tripod substructures have been completed in water depths ranging from 30 to 40 m (EWEA, 2011a).

Offshore wind turbines installed today are generally between 2 and 4 MW. The largest turbines used so far at sea have been 5 MW (Bilgili *et al.*, 2011). However, as shown by Snyder and Kaiser (2008), there is a clear trend toward increasing turbine size in offshore projects in order to achieve economies of scale. Larger wind turbines could be used offshore because of the lack of a number of possible constraints, such as aesthetics and noise limitations. On the other hand, designs need to address issues related to marine conditions, corrosion, and reliability (Fichaux *et al.*, 2009).

Offshore wind power remains relatively costly and risky because of the inherent uncertainties and often severity of the marine environment. The biggest concerns in the economics of an offshore wind farm are the construction and installation of the support structures, the connection to the grid, and operation and maintenance (Snyder and Kaiser, 2008, Sørensen, 2009). According to Morthorst *et al.* (2009), typical investment costs of recent offshore wind farms range from € 1.2 to 2.7 million per MW. The cost of support structures (mainly monopile and gravity-base so far) accounts for about 20 % of the total investment cost. For floating wind turbines, the cost is expected to be much higher. However, the cost for support structures is very dependent on the distance to shore and the depth of the water. New designs for substructures and foundations, as well as new installation vessels, are needed to reduce cost (Snyder and Kaiser, 2008, Fichaux *et al.*, 2009).

In terms of operation and maintenance (O&M), two philosophies are emerging to help reduce cost. The first is to limit the risk of failure while developing a simple and robust turbine. The second is to improve wind turbine intelligence and implement redundancy and preventive maintenance algorithms (Fichaux *et al.*, 2009). Under the harsh environmental conditions of wind and waves, access to offshore wind turbines becomes challenging or even impossible for extended periods (Breton and Moe, 2009). Various methods to provide better access under certain conditions are under consideration and development, including inflatable boats or helicopters (Van Bussel and Bierbooms, 2003). Advanced O&M approaches, based on remote assessments of turbine operability and the scheduling of preventative maintenance to maximize access during favourable conditions, are also being investigated and employed (Wiggelinkhuizen *et al.*, 2008).

It may also be possible to economize by using cables with a capacity slightly less than that of the wind farm. This is because offshore wind farms rarely generate at full capacity (Green and Vasilakos, 2011). Moreover, development plans that take into account interactions between projects and the onshore grid also minimize the cost of connecting offshore wind farms to the grid.

2.1.2 Current Research Activities and Test Sites

Current offshore wind technology has been successfully developed from the onshore wind industry and has resulted in a significant deployment of large wind farms in relatively shallow water (less than 30 m). However, in the future, new technology for wind farms in deeper water, especially in terms of support structures, needs to be developed (Michel *et al.*, 2011).

In order to encourage and support the production of wind power further offshore and in deeper water, costs must be reduced. Current research work aims to reduce the average cost per kW by enhancing component reliability, increasing wind turbine size and exploring novel substructures to support large turbines, including floating structures.

In connection with the Beatrice demonstration wind farm in the UK, the DOWNVInD project (Beatrice Wind Farm Website, 2011), funded by the EU Sixth Framework Programme (FP6), was carried out from 2004 to 2008. The project demonstrated the technological feasibility and commercial viability of deploying large offshore wind turbines in deeper water (at a water depth of 40 m). Two 5 MW wind turbines were installed on jacket substructures and provided electrical power to the Beatrice platforms. This project also focused on the potential environmental impact of the installation and operation of offshore wind farms. The overall environmental impact was reported to be non-significant by its operators (Talisman Energy (UK) Limited, 2006).

The 2006 – 2011 UpWind project (Fichaux *et al.*, 2011), also funded by the EU FP6, focused on the design of very large wind turbines for both onshore and offshore application. The project team developed design tools for the complete range of turbine components, addressing the aerodynamic, aero-elastic, structural, and material design of rotors. A conceptual wind turbine of 20 MW was designed, although more research needs to be carried out to demonstrate its economic feasibility (Fichaux *et al.*, 2011). Among other topics, the research focused on advanced control systems, aiming to reduce the applied structural loads and improve wind turbine design. The team also studied the design of support structures for offshore turbines, based on fixed solutions. They designed a reference jacket substructure to support a 5 MW wind turbine at a water depth of 50 m.

Prototype or small-scale tests of floating wind turbines at sea have been carried out for some of the concepts (Wang *et al.*, 2011), including the HYWIND prototype, the SWAY small-scale model, the Blue H concept, and the WindFloat prototype. The first full-scale spar floating turbine, called HYWIND, was installed by Statoil off the west coast of Norway in September of 2009, at a water depth of 220 m. It has been in operation for more than two years (HYWIND Website, 2011). This prototype is equipped with a 2.3 MW Siemens variable speed pitch regulated wind turbine mounted at a deep draft floating buoy. The 5300 m³-displacement hull is moored by three mooring lines consisting of steel wires and clump weights.

Another prototype using the concept of a floating spar filled with ballast was installed by SWAY off the west coast of Norway in March of 2011 (SWAY Website, 2011).

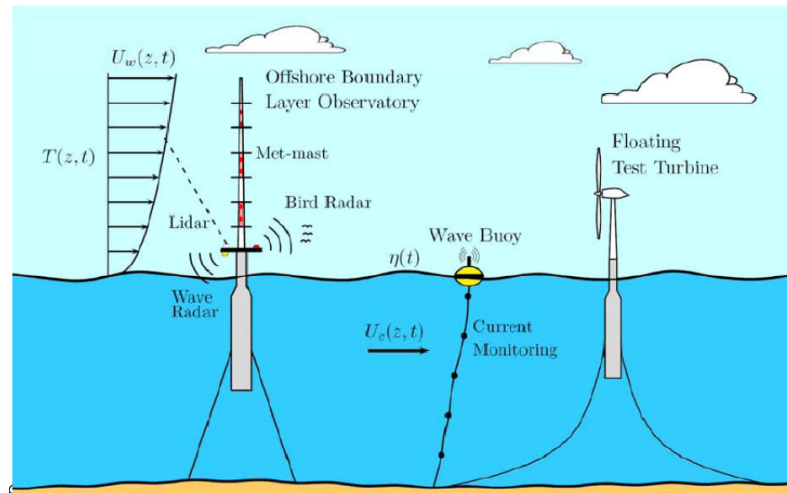


Figure 1: Components of the NOWERI (Reuder, 2011)

Unlike the HYWIND project, in this case, the spar is anchored to the seabed with a single pipe and a suction anchor. This system has the advantage of allowing the wind turbine to revolve naturally as the wind changes direction. An unforeseen sea state condition is reported to have led to its sinking in December 2011. A modified spar which has almost half the draft of a conventional spar was developed by IHIMU (IHI Website, 2011). The concept has ballast and a footing structure with a larger diameter.

A large-scale (3:4) prototype of a tension leg platform (TLP) was installed by Blue H Technologies off the coast of southern Italy in the summer of 2008 (Blue H Website, 2011). The hexagonal floating platform was placed at a water depth of 113 m and fitted with a two-bladed turbine rating 80 kW. The unit was decommissioned after 6 months at sea. A second proof-of-concept prototype, testing a 2 MW wind turbine mounted on a TLP, is expected to be built by 2012 and installed near the site of the future floating wind farm Tricase.

Principle Power Inc. is promoting a semi-submersible floating wind turbine system, called WindFloat, which consists of three columns with patented horizontal water entrapment heave plates at the bases (WindFloat Website, 2011). These structures aim to improve motion performance by using additional damping and entrained water effects. In addition, platform stability would be augmented by a closed-loop active ballast system. A 2 MW version of WindFloat was installed off the shore of Portugal in October 2011.

Recently, there have been developments in the offshore test sites for wind turbines. The Norwegian Offshore Wind Energy Research Infrastructure (NOWERI) is a proposed test infrastructure which consists of an offshore boundary layer observatory and a 250 kW floating test wind turbine, (Reuder, 2011) (see Figure 1). It will be built and testing will start at the beginning of 2013. It will simultaneously measure the environmental conditions of wind, waves, and current, and the dynamic responses of the floater. This can be used to validate numerical tools for floating wind turbines.

2.1.3 Novel Concepts

Future offshore wind turbines may be larger and lighter, and therefore more flexible. Offshore wind turbine size is not restricted in the same way as onshore wind turbines.

Additionally, the relatively higher cost of offshore substructures provides additional incentive to increase returns by building larger wind turbines (Wiser *et al.*, 2011). However, the development of large turbines for offshore applications involves research challenges, requiring continued advancement in component design and system-level analysis. New concepts, such as vertical axis wind turbines, gearless wind turbines, and other types suitable for large scale development are being considered to design more efficient, more reliable, less expensive, bigger, and easier-to-maintain wind turbines.

The vertical axis wind turbine (VAWT) design is known to work well for small scale wind turbines. However, Risø DTU has proposed the concept of a large floating VAWT, to be created through the 4-year long DeepWind project funded by the EU FP7. The floating VAWT shown in Figure 2 consists of a Darrieus type rotor, a long vertical tube rotating in the water, a generator mounted at the bottom of the tube, and an anchoring system (Vita *et al.*, 2010). Risø DTU has suggested three types of configuration: sea bed configuration, torque arm fixed configuration, and mooring fixed configuration. In the sea bed configuration, the shaft is hinged to the sea bottom so that it can tilt back and forth and to the sides, giving it two degrees of freedom. In the torque arm fixed configuration, a torque arm connects the shaft to the sea bed, so that the turbine system can move up and down in addition to back and forth and side-to-side, allowing it three degrees of freedom. In the mooring fixed configuration, three torque arms installed to the generator box are connected to the sea bottom by mooring lines. This system allows for two more degrees of freedom (sway and surge) than the torque arm fixed configuration. Risø DTU calculated the motions and forces of the sea bed configuration using HAWC2 aero-elastic code (Larsen and Hansen, 2008). The HAWC2 was originally developed to simulate horizontal axis wind turbines (HAWTs) and was modified for VAWTs for the project. The results show a strong coupling between the pitch and roll motions of the system, due to the system's gyro-motion hydrodynamic side force. Another coupling between the aerodynamic and hydrodynamic forces also exists, due to the dependency of hydrodynamic force and friction moment to the rotor's rotational speed. This means that hydrodynamic optimization should be included in the aerodynamic optimization. In addition to these results, Risø DTU found that the hydrodynamic load is dominant in the system's dynamics. They determined the hydrodynamic dominance using the tilt angle dependency on the current speed and the elliptical tower motion in the equilibrium in waves. Many facts must be

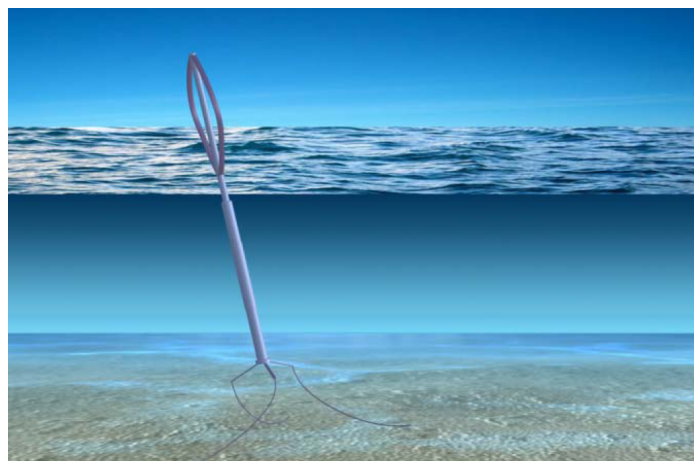


Figure 2: The DeepWind concept (Vita *et al.*, 2010)

clarified before this concept can be realized. However, it could aid in reducing cost in O&M and installation and increase up-scaling potential and suitability for deep sites.

Another concept for offshore vertical axis wind turbines is the NOVA (Novel Offshore Vertical Axis) concept (Cranfield Website, 2009). This project is funded by the Energy Technology Institute (ETI) in UK. The project aims to have 1 GW of offshore vertical axis turbines installed by 2020. The first phase of the project aims to demonstrate the feasibility of a unique vertical winged wind turbine (an aerogenerator turbine) compared to conventional horizontal axis turbines (see Figure 3).

Simultaneously, Siemens and General Electric (GE) are planning to develop larger low speed generators without gearboxes, to replace traditional gearboxes and high speed generators. This kind of wind turbine is called a “direct-drive” or “gearless” turbine. Since a gearbox is not necessary, a gearless turbine has fewer components and hence reduces maintenance costs over the long term. However, to drive directly, the shaft speed should be the same as the rotor speed, which means that much bigger generators are required, making the generator much heavier than gear type generators. In 2010, Siemens developed a 3 MW gearless wind turbine that uses a permanent magnet generator. In 2011, they developed a 2.3 MW turbine for low to moderate speed (Terra Magnetica Website, 2010). Siemens has also announced that it plans to finish developing a 6 MW gearless wind turbine in 2011 that is suitable for large offshore wind farms (ThomasNet Website, 2011). General Electric acquired a 3.5 MW gearless wind turbine from ScanWind in 2009 (Technology Review Website, 2009) and introduced a 4 MW gearless wind turbine optimized for offshore operation in 2011 (REVE Website, 2011). General Electric is also planning to develop up to 15 MW of direct drive wind turbines using superconducting magnets (Recharge Website, 2012). The superconducting technology could reduce the size and weight of the generator. STX Windpower B.V. also commissioned a 2 MW gearless wind turbine in Korea (Offshore Wind Website, 2011).

Some researchers are exploring very large turbines of more than 10 MW. The UpWind project studied the design limits of upscaling wind turbines (Fichaux *et al.*, 2011). They found that rotor diameters of 20 MW would be around 200 m, compared to about 120 m on today's 5 MW turbines (see Figure 4). The project foresees that the



Figure 3: The Aerogenerator X concept (Courtesy Wind Power Ltd and Grimshaw)

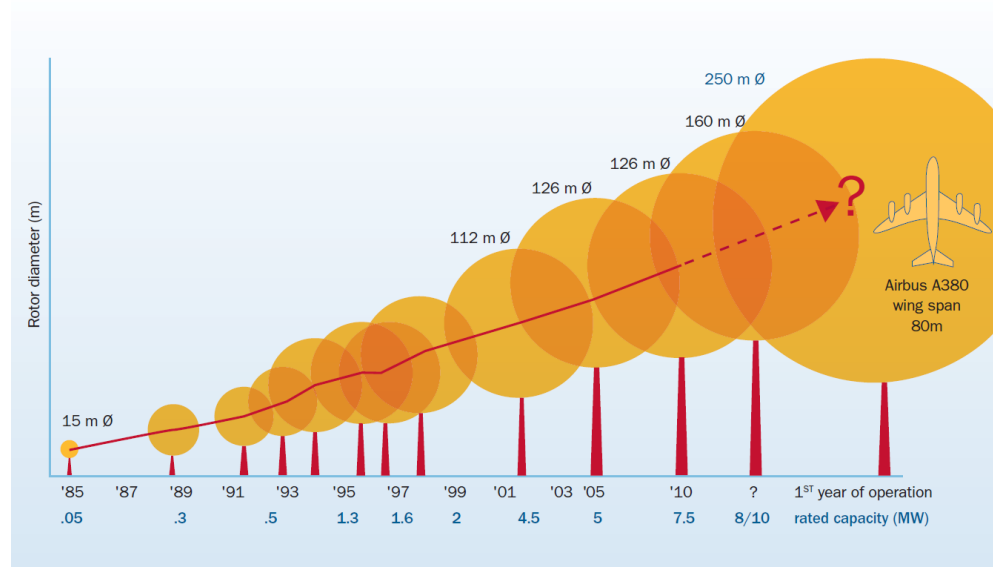


Figure 4: Trend of wind turbine size (Fichaux *et al.*, 2011)

wind turbine up to 20 MW is possible, provided some key innovations are developed and integrated. These innovations include blade material light and strong enough to endure a larger load and a control system incorporating distributed blade control and individual pitch control.

2.2 Fixed Solutions

The 2009 ISSC report introduced various support structures and foundations for offshore fixed wind turbines. These support structures include gravity-base, monopile, tripile, tripod, or jacket, and the foundation solutions are based on piles, gravity bases, or suction buckets (ISSC, 2009).

Currently, offshore wind farms are mainly built in shallow water (less than 30 m of water depth), using monopile or gravity-base support structures. In the next phase of offshore wind development, moderate water depth will be considered and jacket support structures are expected to play an important role for such water depths (Douglas-Westwood Limited, 2010), from 30 m until a water depth at which a floating wind turbine becomes cost-effective.

2.2.1 Jacket Support Structures

Jacket structures are the most common fixed structures used in the offshore oil and gas industry. They have been mainly used for water depths of less than 150 m, but the largest jacket structure was installed in water of 412 m deep (Chakrabarti *et al.*, 2005). However, for offshore wind applications, jacket structures are not considered to be economically feasible at such water depth; they are more suitable for moderate water depths.

In shallow water, monopile or gravity-base structures are the best solution for offshore wind power, due to their simplicity and low cost. When the water depth increases, the overturning moment due to wind loads acting on the rotor increases, and therefore a larger substructure is needed. On the other hand, larger monopiles implies larger wave loads. It is then natural to choose jacket structures that are more “transparent” to

wave loads. So far, there have been two jacket wind turbines installed on the Beatrice wind farm in the UK since 2007 (Beatrice Wind Farm Website, 2011) and six on the Alpha Ventus wind farm in Germany since 2010 (Alpha Ventus Wind Farm Website, 2011). The turbines on both wind farms are REpower 5 MW machines, and the jacket support structures are designed by OWEC Tower AS in Norway. The water depth is 40 m and 30 m, respectively.

The EU FP6 UpWind project designed a jacket support structure in 50 m water depth, using the NREL 5 MW turbine as a reference wind turbine (Jonkman *et al.*, 2009). This jacket structure is now used as a reference model in the International Energy Agency (IEA) Offshore Code Comparison Collaboration Continuation (OC4) benchmark study (IEA OC4 Website, 2011). A preliminary design of a jacket substructure was also made for a 20 MW wind turbine, which was developed in the UpWind project by scaling the NREL 5 MW turbine. However, the project concludes that more research work on both wind turbine and jacket structure is necessary, in order to achieve a cost-effective design for such large scale wind turbines (Vemula *et al.*, 2010).

Gao *et al.* (2010) analysed a jacket structure for a water depth of 70 m, designed by Aker Solutions ASA for a northern North Sea site. The jacket substructure (including transition piece, jacket, and piles) weighs about 1434 t, while the OWEC jacket on the Alpha Ventus wind farm in 30 m water depth weighs about 825 t (Seidel, 2007). All of these wind turbines have a cylindrical tower and a four-leg jacket, and a transition piece is required to connect them. Truss-tower wind turbines have been proposed by Long and Moe (2007), where the legs and braces extend from the sea bed up to the nacelle.

2.2.2 Design Challenges for Jacket Wind Turbines

The major challenge for offshore wind turbines is finding a cost-effective solution. This challenge must be considered for the turbine's entire lifecycle, including the phases of design, installation, operation, and maintenance. In deeper water, the substructure's contribution to the total cost of an offshore wind turbine increases. As wind turbines become standard components, ensuring an optimized and cost-effective substructure design is important to bringing down the total cost. Therefore, this report focuses on the substructure for offshore wind turbines.

As wind turbines become larger (from a rated power of 2–3 MW to 5 MW), dynamic loading in the substructure increases significantly. This creates a design challenge for jacket substructure. Aerodynamic loads may excite resonance and induce a high dynamic amplification for jacket responses, and fatigue design of jacket joints and especially welded joints needs to take these effects into account. Moreover, hydrodynamic loads also contribute to fatigue damage. Cast joints could be used, but the fabrication cost of doing so is significantly high.

From a structure point of view, in addition to the jacket and the piles, the transition piece is also an important element for design. Fatigue is also a challenging problem for transition pieces, due to high stress concentration. Moreover, the stiffness of the transition piece will influence the natural frequency of the complete jacket wind turbine and therefore the overall dynamics. So far, limited information has been published on this structural component. Transition pieces based on steel braces were developed by OWEC Tower AS and used in the jacket wind turbines in both the Beatrice and Alpha Ventus wind farms. The transition piece proposed by Rambøll AS (Vemula *et al.*, 2010) is a concrete structure, which is very stiff but quite heavy. The steel cone

developed by REpower (Seidel, 2007) is much lighter than the other two concepts, but has only been tested for a relatively steep batter angle and a narrow jacket top.

There is limited industrial experience reported so far for jacket wind turbine design. Current design practices deal with the design of wind turbines and jacket substructures separately. The design of the jacket substructure is based on the load information at the interface obtained by aerodynamic analysis from the wind turbine provider. However, due to the coupling of dynamic responses of the wind turbine and support structure, the design process of an offshore wind turbine needs to be carried out in an integrated manner.

The EU FP6 UpWind project proposed and demonstrated an integrated design methodology for the design for a jacket wind turbine in a water depth of 50 m (Fischer and de Vries, 2010). Through this methodology, an optimized substructure design has been achieved to compensate for the variability of sites on a wind farm, and a new wind turbine controller has been developed to actively mitigate the dynamic loads on the support structure. The integrated design method requires coupled analysis tools to address the wind- and wave-induced dynamic responses, which will be discussed in Subsection 2.2.3 under “Coupled Analysis”, see page 169.

Currently, the design method for offshore wind turbines is based on the partial safety factor method, which is also used for onshore wind turbines. The reliability-based design is not a common approach so far, but structural reliability analysis methods have been applied to calibrate the load and material factors used in the design method. Tarp-Johansen *et al.* (2003) proposed a calibration method of partial safety factors for extreme loads on onshore wind turbines, that could be further applied to calibrate the safety factors for combined wind and wave load effects for offshore monopile or tripod wind turbines (Tarp-Johansen, 2005). Sørensen (2011) carried out a reliability-based calibration of fatigue safety factors for the jacket substructure developed in the UpWind project. A reliability level corresponding to an annual failure probability of $2 \cdot 10^{-4}$ has been considered, which is normally used for unmanned offshore fixed structures in the oil and gas industry. However, a structural reliability method requires a detailed analysis of uncertainty, and for offshore wind turbines, more experience needs to be gathered to quantify the model uncertainty of the aerodynamic and hydrodynamic load calculation method, the structural dynamic analysis model, and the method for extreme value prediction or fatigue calculation and the statistical uncertainty related to the time-domain simulation.

2.2.3 Dynamic Analysis of Fixed Wind Turbines

Aerodynamics, Hydrodynamics, and Structural Dynamics

The design of offshore fixed wind turbines needs to address the load effects on the structure under environmental forces. Wind and wave loads are the most important of these load sources. The Blade Element Momentum (BEM) theory (Hansen, 2008) is frequently used for aerodynamic load calculation. Using two-dimensional coefficients, the sectional lift and drag forces and the moment are calculated based on the relative wind speed at various positions of the wind turbine blade, which includes the effect of the wind inflow speed, the induced velocity from the momentum theory, and the velocity induced by the motion and deformation of the blade. Engineering corrections (Hansen *et al.*, 2006) are usually applied to deal with the flow conditions that the BEM method is not able to solve, including the Glauert correction for large induction factors, Prandtl's tip loss correction, the dynamic inflow model, the engineering model for yaw or tilt conditions, and the dynamic stall model. The BEM method is still the

most common tool to obtain the aerodynamic loads on wind turbines, due to its computational efficiency. In recent years, there has been a significant development in more advanced numerical methods (Hansen *et al.*, 2006). This includes the panel method, the vortex method, the generalized actuator disc method, and the Navier-Stokes solvers (full Computational Fluid Dynamics (CFD)). However, the application of the full CFD to rotor aerodynamics is still very time-consuming, and not practical for wind turbine design in the industry.

Hydrodynamic loads on the substructure are also based on a two-dimensional method using Morison's formula, as the typical substructures such as monopile, tripod, and jacket consist of slender members. However, the MacCamy-Fuchs correction is applied on the inertia term of the Morison's formula for structures with a large diameter relative to wave length, such as gravity-base. For wave kinematics, linear waves with the Wheeler stretching model are commonly used for fatigue load analysis. Veldkamp and van der Tempel (2004) compared this model with the second-order wave theory considering irregular waves, and found that the difference in the fatigue loads on a monopile wind turbine is insignificant. However, for this type of wind turbines, the fatigue damage is mainly induced by wind loads. In extreme conditions, the slamming loads due to breaking waves are important to consider, since the aerodynamic damping for such condition is very low. De Ridder *et al.* (2011) carried out an experimental study of breaking wave loads on a monopile wind turbine. Large horizontal acceleration at the nacelle was observed in the order of 0.9 times the gravitational acceleration. CFD has been applied to study such phenomena, e.g. by Bredmose and Jacobsen (2010). However, more work needs to be carried out to implement nonlinear wave theory and slamming load calculation in coupled analysis tools.

In structural dynamic analysis, the complete structure of an offshore wind turbine is normally modelled as beams using the Finite Element Method (FEM), or the multi-body formulation. The geometric nonlinearity due to the rotating blades should be considered for such analysis. Since the wind turbine blades are long and flexible, natural modes of edge-wise and flap-wise bending have low frequency and can be excited by the wind loads. Moreover, due to the large mass of the nacelle and rotors on the top of the tower, the natural frequencies of the first fore-aft and side-to-side bending modes of the complete structure are also low, and these natural modes can be excited by wind loads as well. As an example, for the three-blade NREL 5 MW wind turbine (Jonkman *et al.*, 2009), the lowest natural frequency of the tower plus the rotor is about 0.32 Hz, while the blade natural frequency ranges from 0.67 to 2.02 Hz. It is important to consider the aeroelasticity in the structural analysis for wind turbines.

Design Load Prediction

Both ultimate limit state (ULS) and fatigue limit state (FLS) should be considered for the design of offshore wind turbines. ULS design needs to address the extreme loads or load effects corresponding to a return period of 50 years, see e.g. IEC 61400-3 (IEC, 2009). A full long-term analysis method could be used, which is based on a long-term distribution of responses obtained from the short-term distributions considering the probability of occurrence for all wind and wave conditions. Time-domain response analysis of offshore wind turbines is frequently applied to obtain the short-term distribution, which is further fitted by analytical distributions. Distributions of peak values or extreme values could be considered for each short-term condition. Agarwal and Manuel (2009) have applied this method to study the extreme response of a monopile wind turbine in 20 m of water. To limit the simulation requirement, a response surface

method is applied to obtain the short-term extreme value distribution as a function of mean wind speed, significant wave height, and spectral peak wave period. An inverse first-order reliability method is used to obtain the characteristic long-term extreme value. However, that study only considers operational conditions, since, for such water depth, the extreme wind turbine response is governed by wind loads, which give the largest thrust force at the rated wind speed due to the blade pitch controller. In deeper water, the contribution of wave loads will increase, and it is therefore important to include all of the load conditions in a full long-term analysis.

Alternatively, contour line (or surface) method can be applied to obtain the long-term extreme response, in which response analysis is carried out for a set of conditions along the environmental contours with a return period of 50 years. Then, the largest extreme response predicted under these conditions, with a proper selection of fractile, is used to approximate the long-term extreme response. This method has been frequently used to predict the long-term wave-induced responses for offshore oil and gas platforms (Winterstein *et al.*, 1993). An important assumption is that the long-term extreme responses are mainly due to the contribution from severe wave conditions. For offshore wind turbines, such assumptions must be checked carefully, since other conditions than extreme wind and wave conditions might give large responses. More research work needs to be done to validate the use of the contour line method for offshore wind turbines.

Since short-term time-domain simulations are normally based on a 10-minute period with several random seeds, when the short-term extreme value distribution is used, an extrapolation method must be applied in order to obtain the extreme value – for example, in 1 hour or 3 hours. Cheng (2002) compared various methods for short-term extreme value prediction and concluded that the peak-over-threshold method and the method based on statistical moments agree well with the reference method using the maximum value directly from simulations.

The current design codes, e.g. GL IV Part 2 (GL, 2005), IEC 61400-3 (IEC, 2009), and DNV OS-J101 (DNV, 2011), require a ULS check for a number of load cases, defined by various environmental conditions of wind, waves, and current, including both operational and extreme conditions with and without faults. This assumes that the 50-year extreme responses are captured by these defined load cases.

Currently, the design of offshore wind turbines with respect to FLS is normally based on the SN-curve approach. Since time-domain dynamic analysis is required, the fatigue loads are extracted directly from the load or stress time history by using the rainflow cycle counting method. For monopile wind turbines in relatively shallow water, design loads are mainly governed by wind loads. For jacket wind turbines, both aerodynamic and hydrodynamic loads need to be considered in the design. However, the relative importance of wind and wave loads is different for ULS design and FLS design. As shown by Cordle *et al.* (2011), both wind and wave loads are important to consider for extreme load cases. However, fatigue loading is found to be dominated by wind, with a relatively low contribution from hydrodynamics. Dong *et al.* (2011) drew a similar conclusion after carrying out a detailed fatigue analysis of joints in a jacket wind turbine of 70 m water depth. They found that the first four bending modes of the complete jacket wind turbine are excited by the wind loads on the rotor, which contributes significantly to the fatigue damage of tubular joints. However, the hydrodynamic response is quasi-static, since the jacket substructure is stiff. Moreover, the wind- and wave-induced responses have different frequency components, which lead the combined stress history to be broad-banded. Therefore, the fatigue damage

obtained from the combined history is much larger than the summation of the wind- and wave-induced fatigue damages.

Decoupled Analysis

Time-domain simulations of wind- and wave-induced dynamic responses of offshore wind turbines are usually performed using numerical tools. A decoupled analysis calculates the wind- and wave-induced responses separately and uses the summation to obtain the total responses, while in a coupled analysis, both wind and wave loads are applied simultaneously. Considering the large number of load cases, especially in a fatigue design check, decoupled analysis is still needed in practice, due to its computational efficiency. Such methodology is still used in the current design practice in industry, where the designs of wind turbine and jacket substructure are carried out by the turbine provider and the substructure provider, separately, with the exchange of necessary information.

Kühn (2001) has developed a simplified procedure for long-term fatigue analysis of a monopile wind turbine, based on the separate analyses of aerodynamic responses in the time domain and hydrodynamic responses in the frequency domain. Combined fatigue damage is calculated based on a quadratic superposition of separate wind- and wave-induced damages.

In the analysis of tripod wind turbines, Seidel *et al.* (2004) modelled the tripod substructure as a 6-DOF (degree of freedom) system with equivalent mass and stiffness matrices and hydrodynamic loading at the interface. The team performed time-domain simulations based on this equivalent model, using the standard aerodynamic code FLEX 5 (Øye, 1999). They applied a sequential coupling approach using time series of forces or displacements at the interface to analyze the dynamic response in the substructure. In general, they observed a good agreement between the sequential coupling approach and a full coupling approach. However, for jacket wind turbines, they found that this reduction method is not able to capture natural modes at higher frequencies, and the sequential approach over-predicts the responses at these modes (Seidel *et al.*, 2009).

Passon (2010) has compared the equivalent fatigue damage in jacket joints by using the sequential coupling methods with and without consideration of the inertia forces in the jacket structure. The method neglecting the inertia effect underestimates the fatigue damage significantly. He also found that the super-position of separate wind- and wave-induced responses provides accurate estimates of fatigue damages. Moreover, he studied the effect of local joint flexibility, and, found that, in most of cases, the method with no consideration of local joint flexibility overestimates the fatigue damage.

Gao *et al.* (2010) approximated the substructure of a jacket wind turbine with several vertical beams to achieve the equivalence in mass, stiffness and hydrodynamic loading, as well as the first and third bending modes. The sequential method was applied to dynamic response analysis due to wind and wave loads, which gave very similar global responses (e.g. the shear force and bending moment at the sea bed) in the jacket substructure and in the equivalent model. Based on this method, Gao and Moan (2010) carried out a long-term fatigue analysis considering these global responses, and they discussed the contribution of various short-term wind and wave conditions. However, this modelling technique needs to be verified with the fully coupled analysis of a jacket wind turbine, and the effect of torsional modes needs to be addressed.

A proper estimation of damping force is important for a dynamic response analysis of offshore wind turbines. Damping of an offshore fixed wind turbine consists of

aerodynamic damping, hydrodynamic damping, soil damping, and structural damping. Aerodynamic damping from an operational wind turbine is the most important contribution. For a 3 MW, two-blade, fixed-speed variable pitch wind turbine, Kühn (2001) estimates the aerodynamic damping as from 5 % of the critical damping in low wind speeds to 0.5 % in high wind speeds. However, the aerodynamic damping in the cross-wind direction is very small. Tarp-Johansen *et al.* (2009) have studied soil damping for a monopile foundation using a 3D finite element model, and suggest a damping ratio of 0.55 – 0.8 %. The structural damping provided by the monopile is very small, while that of a jacket structure with welded joints is higher. GL (2005) suggests a total 1 % of the critical damping for extreme wind conditions, in which the turbine is shut down and the aerodynamic damping is negligible.

Coupled Analysis

In a coupled analysis for offshore wind turbines, the aerodynamic loads, the hydrodynamic loads, and the structural responses are dealt with simultaneously in the time domain, as well as in the wind turbine controller. The phases between the wind and wave excitations and the structural responses are properly considered in such analysis, and various types of damping sources, including aerodynamic, hydrodynamic, soil and structural damping, are included in a correct manner. The dynamic wind turbine loads due to the change of rotor speed or blade pitch angle by the controller are also properly modelled. Both aeroelasticity and hydroelasticity are therefore considered, but the effect of the aeroelasticity is much more important than that of the hydroelasticity, since the blades are more flexible than the substructure.

Currently, most of the numerical tools dealing with coupled analysis of offshore fixed wind turbines apply the BEM method for aerodynamic loads, Morison's formula for hydrodynamic loads, and a beam model for structural members. For monopile or tripod offshore wind turbines, coupled analysis tools have been developed as an extension of aerodynamic codes for onshore wind turbines, and have been benchmarked in a code-to-code comparison and reported in the IEA OC3 study (Jonkman *et al.*, 2010).

Jacket support structures are more complicated in geometry. In order to analyze jacket wind turbines, an integration of existing aerodynamic, hydrodynamic, and structural analysis codes is needed. Such numerical tools exist. For example, Seidel *et al.* (2009) compared the coupled tool of FLEX 5-ASAS(NL) with the measurement of the jacket wind turbine in the DOWNVInD demonstration project, and the preliminary comparison showed a good agreement. Another example is the nonlinear aero-elastic code ADCoS-Offshore (Moll *et al.*, 2010).

More coupled analysis tools are now under development and need to be verified through code-to-code or code-to-experiment comparison. As a continuation of the IEA OC3 work, the ongoing IEA OC4 study is now comparing many of the existing analysis tools for a reference jacket wind turbine, developed in the EU FP6 UpWind project.

More details concerning the numerical tools for offshore fixed wind turbines are discussed in Section 2.5.

2.3 Floating Solutions

2.3.1 Fundamental Differences as Compared with Fixed Wind Turbines

Main Characteristics of Floaters vs. Fixed Offshore Structures

All loads on a fixed structure, both vertical and horizontal, must be carried by the structure and transferred down to the foundation at the sea bed. A common feature of

all types of floaters is that they utilize excess buoyancy to support the payload (tower and nacelle) and the horizontal loads are carried by the station keeping system. Fixed structures are stiff in the horizontal plane with a natural period normally well below 5 s, while all floater types naturally have surge, sway, and yaw periods generally longer than 100 s, due to the fact that they are “soft” in the horizontal plane. The natural period of a fixed structure is governed by the stiffness of the structure, while the natural period of a floater in surge, sway and yaw is governed by the station-keeping system.

Floater Design

Depending on the area and the sea state, ocean waves contain 1st harmonic wave energy in the period range of 5 – 25 s. For a floating unit, the natural periods of motions are key features, and in many ways reflect the design philosophy. As an example, the heave natural period for a spar is normally positioned above 25 s, while the natural period for a tension-leg-platform (TLP) is normally below 5 s, as indicated in Figure 5.

The fundamental differences among floaters are related to their motions in the vertical plane, i.e. heave, roll, and pitch. Floater motions are important for the choice of power take-off cables, umbilicals and mooring systems. Typical heave transfer functions for different floaters and a storm wave spectrum are shown in Figure 6. The figure is based on DNV RP-F205 (DNV, 2010b).

Floating structures such as spars, semi-submersibles, monohulls, and TLPs are well known by the offshore industry with respect to motion characteristics and critical components. When these structures are used to support wind turbines, new challenges may arise in the design caused by downsizing due to the smaller payload. This applies to the structure, mooring, power take-off cables, and umbilicals. The different floater types have different characteristics, as outlined in the following sub-sections.

Deep Draft Floater Response Characteristics (Spar)

A deep draught floater (DDF) is characterized by small heave motions. An example of a DDF is a spar platform. The main hull of a spar is a vertical cylinder which provides buoyancy. Fixed and floating ballast are often employed at the bottom to control the floating performance. The dominant loads are wind and wave loads. The spar also has a large area exposed to current forces. Low frequency (LF) vortex-induced motions (VIM) may increase the effective drag, leading to higher mean current forces. By adding strakes on the spar hull, possible vortex-induced cross-flow oscillation can be significantly reduced. However, the strakes will increase the mass and the drag forces on the spar. Deep draft floaters’ small heave motions are advantageous for the power take off cables, the umbilicals (instrument cables), and the moorings.

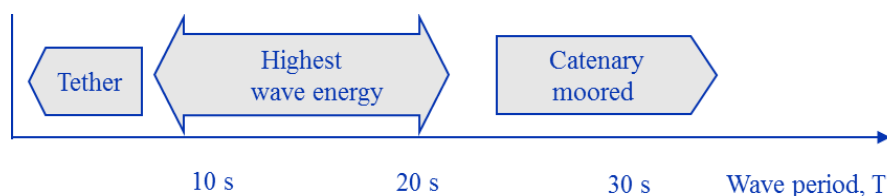


Figure 5: Typical heave periods for different floating solutions vs. location of highest wave

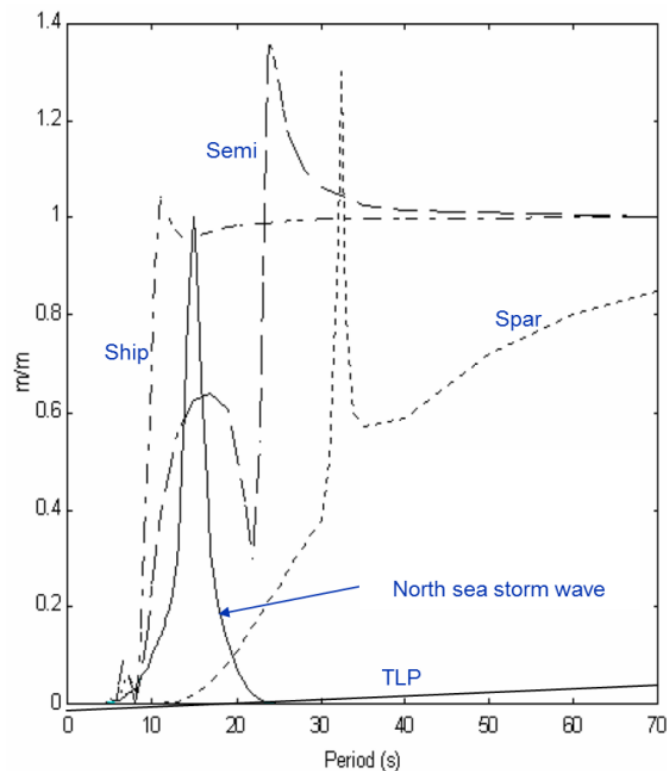


Figure 6: Heave transfer functions for different floaters and storm wave spectrum (DNV, 2010b)

Semi-submersible Response Characteristics

A semi-submersible is usually a column-stabilized unit, which consists of large diameter support columns attached to submerged pontoons. The pontoons may be of different designs, such as ring pontoons, twin pontoons, or multi-footing arrangements. Semi-submersibles have small water-plane areas, which give rather high natural periods in vertical modes. For offshore platforms, the natural period in heave is usually outside the range of wave periods, except for extreme sea states. This implies that a semi-submersible normally has relatively small vertical motions compared to a monohull floater (e.g. barge). However, its behaviour in extreme weather requires flexible, hang off systems or a hybrid arrangement for this concept. A semi-submersible may be equipped with a variety of mooring systems. For wind turbine support structures, heave plates are used to reduce motion (Cermelli *et al.*, 2009).

TLP Response Characteristics

A TLP differs fundamentally from other floater concepts in the sense that it is the tendon stiffness rather than the water-plane stiffness that governs vertical motion. The tendon system is a soft spring in surge, sway, and yaw motions, but stiff in heave, roll, and pitch motions. A TLP generally experiences wave frequency (WF) motions in the horizontal plane that are of the same order of magnitude as those of a semi-submersible of comparable size. In the vertical plane, however, the TLP will behave more like a fixed structure, with almost no WF motion response. The tendon stiffness forces directly counteract WF forces.

Table 1: Typical floaters and boundary conditions

Type	Surge , Sway	Heave	Roll , Pitch	Yaw
Deep Draft Floaters (spar)	C	C	C	C
Semi submersibles	C	C	C	C
Monohull (barge)	C	C	C	C
Tension Leg Platforms (TLP)	C	R	R	C

Higher order sum-frequency wave forces may introduce springing and/or ringing responses in the vertical modes. These effects may give significant contributions to the fatigue responses of the tethers. Set-down is the kinematic coupling between the horizontal surge/sway motions and the vertical heave motions. Set-down is important in the calculation of tether forces and power take-off cable responses.

Monohull Response Characteristics

A monohull structure might be shaped like a barge or a ship. Due to a large waterplane area, these structures are susceptible to large motions, in particular head sea and beam swell. Significant roll accelerations may occur and thus have an impact on the turbine. Such roll accelerations will also have large impact on the design of cables and the mooring system. Large bilge keels may be necessary to control motions. The selection of proper roll damping is important in predicting responses.

2.3.2 Comparison of Different Concepts

The selection of substructures for floating wind turbines depends on several parameters. The main boundary conditions are the environmental conditions (wind, wave, current) and the water depth at the site. These boundary conditions are somewhat correlated, especially with respect to wave heights, as waves will eventually be limited in height, due to finite water depth.

Existing floating offshore structures form the main reference base. However, floating wind turbine structures will be of smaller sizes and volumes even though the drafts may be of the same order of magnitude as the drafts of existing offshore floating structures. For concepts with multiple turbines mounted, the horizontal dimension may be larger in the horizontal plane. The floaters may either be compliant (C), or restrained (R) for the global modes of motions; surge, sway, heave, roll, pitch and yaw. Table 1 shows some typical floater types with a basis in floating offshore structures for easy reference. Restrained modes will not imply a total fixation, but displacements in the order of centimetres will be derived (e.g. an elastic stretch of a TLP tendon) compared to displacement in the order of metres for a compliant mode.

An overview of different floater types is also given in Ronold *et al.* (2010).

2.3.3 Challenges of Floating Wind Turbines

General

The main challenge of floating wind turbines is to reduce the unit costs of the produced energy, while at the same time maintaining an acceptable level of safety. All aspects contributing to the cost should be improved, such as:

- Development of design rules;
- Development and validation of design tools;
- Turbine design;
- Design of station-keeping;
- Design of power take-off cables and umbilicals;

- Design of foundations for cyclic loads;
- Determination of accidental load events;
- Requirements for floating stability and minimum compartmentalization;
- Fabrication, transportation, assembly, and installation costs;
- Corrosion engineering and control; and,
- Cost optimized operation, including maintenance and repair.

Combined Wind and Wave Loading / Wind Turbine Control

New floating wind turbine concepts must ensure that peculiar effects like Mathieu Instability (MI) and vortex-induced motions (VIM) for a DDF are unlikely to occur or can be controllable. For example, care must be taken in selecting eigenperiods in heave, roll, and pitch for a DDF. Concepts with abrupt changes in waterplane stiffness and metacentric height will also have focus on MI. Model testing will be the ultimate method for this, but state-of-the-art offshore design practices and methodologies provide sufficient guidance for early design stages. Reference is made to Ronold *et al.* (2010).

According to Roddier *et al.* (2009), the aero-hydro coupling of the wind turbine with the floater needs to be investigated in detail. Software must be validated against model and full-scale tests. A significant amount of work on the qualification of the turbine is needed. The turbine itself will most likely need to be improved and strengthened. Optimum floater design must be achieved in order to create a cost-effective solution for offshore floating wind turbines.

Berthelsen and Fylling (2011) present a design optimization approach which combines available response analysis programs for mooring system forces and vessel motions with a gradient search method for the solution of nonlinear optimization problems with arbitrary constraints. They considered the following design constraints: vessel motion, tower inclination, tower top acceleration, spar draft, mooring line load limitations, minimum horizontal pretension, and maximum horizontal offset.

Fatigue

Fatigue of a floating structure will be a larger challenge than for a fixed. Reference is made to Aubault *et al.* (2009), who state that the wind force is essential to the strength behaviour of the WindFloat, since its contribution to the bending stress of the structural members is significant. The effect of aerodynamics must be included in the detailed structural analyses.

Karimirad and Moan (2010) compare the structural response of the floating wind turbine in both survival and operational conditions, to show the importance of analysing the structural response in survival conditions to obtain lifetime optimization.

Station-Keeping

Optimizations of station-keeping systems may lead to non-redundant systems where a mooring failure may lead to a loss of position and possible conflict with adjacent wind turbines. If some of the structures in a wind park are planned to be manned during storms (e.g. substations), this might also influence the design requirements. Progressive drifting of floating units should be considered carefully. Reference is made to Suzuki *et al.* (2009a).

Floating Stability

For manned structures, existing offshore codes can be applied for stability check under both intact and damaged conditions. For unmanned structures, other and more

relaxed rules might be appropriate. For unmanned structures, additional compartmentalization is usually not required, unless more stringent requirements (governmental, operator) have been put forward. The need for a collision ring in the splash zone will have to be evaluated with basis in local legislation/requirements, manned/unmanned status, substructure material (concrete/steel/composites), size of service vessels in the area, and resistance to boat impacts.

Power Take-off Cables and Umbilicals

The design of cables for the transfer of electricity and other signals should be integrated with the design of the floater and global performance. Typical issues include the selection of the hang-off location, the motions at hang-off and relevant configuration (e.g. pliant wave, lazy wave, steep wave, or other). The design of umbilicals/cables should follow the same ISO codes as those used for the design of traditional offshore applications. Emphasis should also be placed on the cable installation phase, in order to ensure that the system is positioned as planned. Strakes and/or fairings may be needed to limit vortex-induced vibrations (VIV), or vortex-induced motions (VIM). VIV would typically involve the bending of long slender members either due to wind or to wave/current for the submerged members. VIM typically involves rigid-body motions (in-line, or cross flow) of the complete substructure, causing direct impact on mooring and tethers, as well as cables. Model testing is needed to check the susceptibility to VIM. Alternatively, computational fluid dynamics (CFD) testing is sufficient if the CFD software has been validated.

2.3.4 Coupled Dynamic Analysis of Floating Wind Turbines

For design purposes, dynamic responses of floating wind turbines in wind and waves need to be determined, which includes those of the rotor blades, tower, floater, and mooring system. Since a floating wind turbine has many natural modes of motions or vibrations with different periods that might be excited by the individual wind and wave loads or combined loads, it is important to obtain the dynamic responses considering the wind and wave loads simultaneously. Time-domain analysis is preferred due to the nonlinear aerodynamic loads and control action.

Numerical tools have been developed or are under development, to analyze the dynamic responses for various floating wind turbine concepts. The IEA OC3 study compared various numerical codes for a 5 MW spar floating wind turbine (Jonkman and Musial, 2010), while the comparison of a semi-submersible concept is now being carried out in the OC4 study. Typically, the BEM method with engineering corrections is used for aerodynamics, while the hydrodynamic loads are based on Morison's formula or potential theory.

The spar-supported floating wind turbine is the only concept that has been extensively analyzed. For analysis of the HYWIND concept, the integrated SIMO/RIFLEX/HAWC2 simulation tool has been developed and has shown a good agreement on dynamic responses with the model test results and the prototype test results (Skaare *et al.*, 2007, Hanson *et al.*, 2011). HAWC2 has also been applied to analyze a spar floating wind turbine, for example by Karimirad and Moan (2010), where the hydrodynamic loads on the spar are based on Morison's formula and the mooring system is modelled as nonlinear springs.

Jonkman (2007) has developed a fully coupled time-domain aero-hydro-servo-elastic simulation tool, FAST, with AeroDyn and HydroDyn, where the hydrodynamic loads obtained from WAMIT (Lee and Newman, 2006) are used. This tool has been used

for the dynamic analysis of three floating wind turbine concepts, supported by spar, TLP, and barge, respectively (Jonkman and Matha, 2011). The studies found that the platform motion-induced ultimate and fatigue loads for all turbine components in the barge concept are the highest among the three concepts, while the difference between the spar and the TLP concepts is not significant. Moreover, as compared with the onshore wind turbine, the dynamic responses of the blades in the spar and TLP concepts are not very different, while those of the tower base are 60 % and 30 % larger for the ultimate loads for the spar and the TLP concepts, respectively.

For a floating platform with a pitch-regulated turbine, the conventional land-based controller may give a ‘negative damping’ effect and induce large resonant pitch motions of the platform. This is because, for the wind speed larger than the rated wind speed, the blade-pitch controller is activated to obtain constant power output. However, when the relative wind speed increases due to the pitch motion of the platform, the wind thrust force decreases, leading to a ‘negative damping’ effect. The effect can be avoided by tuning the controller (Larsen and Hanson, 2007, Skaare *et al.*, 2010) or adding a tower-feedback control loop (Jonkman, 2007).

More details concerning the numerical tools for offshore floating wind turbines are discussed in Section 2.5.

2.4 Design Rules for Fixed and Floating Wind Turbines

Current standards for design of offshore wind turbines and their support structures essentially consist of the following four documents:

- IEC61400-3
- DNV-OS-J101
- GL (IV Part 2)
- ABS #176

The IEC standard 61400-3 (IEC, 2009) was issued in 2009, close to ten years after the decision was made to develop this standard. The DNV and GL standards, DNV-OS-J101 (DNV, 2011) and GL (IV Part 2) (GL, 2005), were first issued in 2004 and 2005, respectively. These two industry standards – mostly synchronized with IEC61400-3 – form the design standards which are used as a basis for the project certification services that DNV and GL offer for offshore wind farm projects. The ABS standard #176 (ABS, 2010) was first issued in 2010 and is also mostly aligned with IEC61400-3; however, it is the only standard among the four to address wind turbines in areas prone to tropical storms.

2.4.1 Status Regarding Design Standards for Floating Wind Turbines

The four standards available for the design of offshore wind turbines and their support structures are all encumbered with the common limitation that, in practice, they are restricted to the design of bottom-fixed structures only, as they do not cover floater-specific design issues, such as stability and station keeping. The IEC standard does not contain specific requirements for floaters. The DNV and GL standards do not explicitly exclude floating wind turbine solutions; however, they do not deal with floater-specific issues. For floater-specific issues, such as mooring, the GL standard references other GL rules, which are not dedicated to wind turbines and thus not calibrated for wind turbine loads. The ABS standard specifically excludes floating wind turbine installations.

The following subsections summarize the status for various stakeholders and publishers of standards, such as regulatory bodies and certifying bodies, with respect to standards for floating wind turbines and their support structures.

DNV

DNV has developed a 'Guideline for Offshore Floating Wind Turbine Structures', which was issued in 2009 as a technical report to supplement the existing DNV standard for bottom-fixed support structures, DNV-OS-J101. This guideline, less formal than an official standard document, addresses floater-specific issues, such as stability and station-keeping. It references DNV (2009) and Ronold *et al.* (2010, 2011). In 2011, DNV initiated a joint industry project with the aim of developing a new standard for the design of floating support structures for wind turbines.

GL

Extension of the current GL standard to further floater-specific requirements is under development, focusing on stability requirements and mooring applications.

BV

BV has developed a guidance note for the 'Classification and Certification of Floating Offshore Wind Turbines'. The guidance note, NI572, was issued in 2010 and appears to be produced by BV's Marine Division, to allow for classification of floating support structures. The document references BV (2010). BV appears not to have any design standard of their own for offshore wind turbines and their support structures.

IEC

A Korean proposal, submitted in 2010, for development of an authoritative standard for floating offshore wind turbines was considered premature by the TC88 committee and was therefore changed by the committee into a proposal for development of a technical IEC specification. This proposal has been accepted by a majority of the voting TC88 committee members and an IEC working group has been formed and has begun the task of developing the technical specification. In addition, DNV has proposed an extension of IEC61400-3 for floating wind turbines (DNV, 2010a). Much of this proposed extension is based on the DNV guideline, DNV (2009).

2.4.2 Discussion

Current standards for the design of bottom-fixed wind turbine structures reflect that it is cost optimal to carry out site-specific designs, for example resulting in individual pile lengths for the monopiles in a large wind farm of turbines supported by monopiles. The individual pile lengths match different water depths and different soil properties between wind turbine positions. Structural steel is also expensive.

New standards for floating wind turbines will be likely to be different, in that it will be cost efficient to use identical mass-produced units for all supporting floaters on a large wind farm. This means that structural design will be likely to be optimised for a fleet of floaters for site-specific environmental data rather than optimised for each individual support structure, as is usual for fixed support structures. In particular, in light of such a mass-production approach to support structures for floating wind farms, it becomes very important for new design standards to ensure sufficient safety against systematic errors in design.

Keeping cost low is important for the design of both fixed and floating wind turbine structures, since the wind farms on which they are used are often economically marginal projects. When new design standards are developed, it will therefore be a challenge to establish more accurately the safety level necessary for the wind turbines and their support structures on a wind farm. The consequence of failure of a single wind turbine will likely be smaller on a large wind farm than on a small wind farm.

Therefore, the required target reliability for the design of a wind turbine and its support structure would depend on the number of turbines on the wind farm. This issue of target safety level applies to both fixed and floating support structures.

For floating wind turbine structures, in contrast with bottom-fixed structures, low frequency response is an issue. This necessitates models to adequately capture environmental conditions in the low frequency range, beyond what is included in current standards for bottom-fixed turbines. This needs to include, but is not limited to:

- Adequate representation of wind in the low frequency range, where some of the commonly applied power spectral density models for wind are known to not provide a particularly good representation;
- Definition of gust events based on gust periods in excess of 12 seconds. The definition must cover expected events and reflect frequencies encountered for the dynamics of floaters;
- An alternative two-peaked spectral density model for floaters, which can be excited by swell. The uni-modal JONSWAP wave spectrum, which is commonly used for representation of wave energy; is insufficient, and
- Set-down effects for water level. Water level may be of significant importance for tension leg platforms.

The low frequency response of floating support structures means that they require longer simulation periods than those usually needed for bottom-fixed structures. A simulation period of load and response of 3 to 6 hours may be required to accurately capture nonlinearities, second-order effects, and slowly varying responses. This poses some challenges, as wind cannot be considered stationary over time scales as long as 3 to 6 hours (Ronold *et al.*, 2010).

2.5 Numerical Tools for Dynamic Analysis of Offshore Wind Turbines

This section describes the current development of numerical tools for analysing offshore wind turbines. There are two types of analysis of the dynamics of wind turbines—frequency domain and time domain. Frequency domain analysis has been used in the oil and gas industry, and is simple to use. However, it cannot take into account nonlinear dynamic characteristics and the response due to control systems, which are important in wind turbine system analysis. Time domain analysis is widely used for the design and analysis of wind turbine systems.

The IEA Wind Task 23 Subtask 2 project (Offshore Codes Comparison Collaboration, or OC3), compares the results of a dynamic analysis of fixed bottom wind turbines with various codes (Jonkman and Musial, 2010). The codes used are FAST, FLEX5, Bladed, Bladed Multibody, ADAMS, SIMPACK, HAWC, HAWC2, BHawC, and ADCoS-Offshore. A list of the participation codes and their capability is shown in Table 2. The codes were applied to monopole supports with rigid foundations, monopole supports with flexible foundations, and tripods. Various response parameters were selected for comparison, considering a number of defined load cases, including constant and uniform wind speeds and stochastic wind field, and regular and random waves. The main conclusion from this benchmark work is that these codes agree well in general for both the monopile and tripod wind turbine models. However, the differences in the response parameters between the codes are also observed, which are mainly due to the differences in the structural modelling, the wind field modelling, the implementation of the BEM code, and the discretization of substructure for hydrodynamic loading.

Table 2: Overview of the codes participated in the IEA OC3 benchmark study

FAST	FLEX5	Bladed	Bladed Multibody	ADAMS	SIMPACT	HAWC	HAWC2	BHawC	ADCoS- Offshore
Code Developer									
NREL	DTU	GH	GH	MSC + NREL	SIMPACT + SWE + NREL	Risø	Risø	Risø + Siemens	ADC + IWES
OC3 Participant									
NREL + CENER	DONG + SWE + Vestas	CENER + GH	GH	NREL	SWE	DNV + Risø	Risø	Siemens	IWES
Aerodynamics									
(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS	(BEM or GDW) + DS
Hydrodynamics									
(Airy* or UD) + ME	(Airy* or UD or Stream) + ME	(Airy* or Stream) + ME	(Airy* or Stream) + ME	(Airy* or UD) + ME	None	(Airy* or UD) + ME	(Airy* or UD) + ME	(Airy* or UD) + ME	(Airy* or UD) + ME
Control System (servo)									
DLL, UD, SM	DLL, UD	DLL	DLL	DLL, UD	DLL, UD	DLL, UD	DLL, UD, SM	DLL, UD	DLL, UD
Structural Dynamics (Elastic)									
FEM* + (Modal / MBS)	FEM* + (Modal / MBS)	FEM* + (Modal / MBS)	MBS	MBS	MBS	FEM	MBS / FEM	MBS / FEM	FEM
ADC – Aero Dynamik Consult Ingenieurgesellschaft mbH Airy* – Airy wave theory; (+) with free surface connections BEM – blade-element/momentum DLL – external dynamic link library			DS – dynamic stall GDW – generalized dynamic wake FEM* – finite-element method; (P) for mode preprocessing only MBS – multibody-dynamics formulation				ME – Morison's equation MSC – MSC Software Corporation SM – interface to Simulink* with MATLAB* UD – implementation through user- defined subroutine available		

The project is continuing in the IEA Wind Task 30 project (Offshore Codes Comparison Collaboration Continuation, OC4). This project will compare the results of jacket substructures (IEA Wind Task 30 Website, 2011).

As for the dynamic analysis of floating wind turbines, aerodynamics, hydrodynamics, control commands, and structural dynamics should be solved simultaneously, i.e. a coupled analysis of aero-hydro-servo-elastic. Cordle and Jonkman (2011) give a good summary of programs for the coupled dynamic analysis of floating wind turbines. The OC3 study has compared various codes for a spar floating wind turbine. The OC4 study will examine a semi-submersible floating wind turbine.

Among these numerical codes, one of the best known analysis tools is FAST, developed by the National Renewable Energy Laboratory (NREL) (Jonkman and Buhl, 2005). It was developed for fixed wind turbines, but it was extended to enable coupled dynamic analysis of floating wind turbines. It covers the structural dynamics, aerodynamics (AeroDyn), hydrodynamics (HydroDyn) and quasi-static mooring analysis. For structural dynamics, it uses modal and multi-body system dynamics (MBS) representation. For MBS, it has the option of using the ADAMS commercial program. The hydrodynamic forces include hydrostatic force, nonlinear viscous drag from Morison's equation, wave exciting forces, and radiation force. The hydrodynamic coefficients are calculated using WAMIT (Lee and Newman, 2006).

A few programs are combined with FAST to take advantage of the program openness. One of them is the CHARM3D-FAST combination. CHARM3D is a time domain floater and mooring line analysis tool developed by Shim and Kim (Shim and Kim, 2008, Bae *et al.*, 2011). CHARM3D also uses WAMIT to calculate the hydrodynamic coefficients and mean drift forces of floaters. TimeFloat, a time-domain software tool for analyzing floating structures, is also combined with FAST for a coupled dynamic analysis of floating wind turbines (Cermelli *et al.*, 2009, Roddier *et al.*, 2010). SIMPACK, a commercial MBS code, also uses AeroDyn and HydroDyn to simulate floating wind turbines (Matha *et al.*, 2011).

HAWC2 was developed at Risø DTU (Larsen, 2009). It was originally intended for

calculating onshore wind turbine response in time domain and has a structural formulation based on multi-body dynamics. It has been extended for dynamic analysis of both offshore fixed and floating wind turbines. The hydrodynamics is based on Morison's equation.

WindHydro is a coupled dynamic analysis program for floating wind turbines developed by Rim *et al.* (2010, 2011). It uses AeroDyn of NREL to calculate the aerodynamic forces and their own code to calculate the hydrodynamic coefficients and forces. DAFUL, a commercial multi-body and structure dynamics analysis program, is used for multi-body dynamics analysis considering the flexibility of the blades and the tower.

SIMO/RIFLEX is a time domain offshore simulation code and was extended to analyse wind turbines. Two means of calculating aerodynamic forces exist. One uses its own aerodynamic module (Fylling *et al.*, 2009) and the other uses HAWC2, which is for the dynamic analysis of fixed wind turbines (Skaare *et al.*, 2007).

There are two commercial codes for the dynamic analysis of wind turbines – GH Bladed and S4WT. GH Bladed, developed by GL Garrad Hassan, was developed for onshore fixed wind turbine dynamic analysis and has been extended to analyse offshore wind turbines by including hydrodynamic loads (GL Garrad Hassan, 2010, Henderson *et al.*, 2010). Flexible structural parts, such as the blades and the towers, are modelled by modal representation. The structural dynamics part was rewritten to incorporate MBS from FEM based code. Morrison's equation is used to calculate the hydrodynamic forces.

S4WT, developed by SAMREC, was also developed for onshore wind turbines and extended to analyse offshore wind turbines, both bottom fixed and floating. With this method, jacket and monopile modelling can be done in parametric modelling and floating turbines can be completed via user defined modelling (Heege *et al.*, 2010, 2011). Aerodynamic and hydrodynamic coupling analysis is also possible. Analysis can be done according to GL guideline and IEC codes.

3 WAVE ENERGY CONVERSION

Wave energy, like wind, suffers from the variability of environmental conditions, which may range from flat calm to severe waves. With small waves during a given period, limited energy can be produced. On the other hand, extreme waves may also limit the ability of the structure to capture energy and may even cause damage to the structure. The structure must be designed to the average wave conditions to be efficient and economical, but must also withstand the more severe conditions and possibly continue to operate. Although wave energy is much more plentiful than wind, exploitation of wave energy has lagged behind that of wind (Falcao, 2010). This is probably due to the steady progression of wind energy exploitation from land-based to near-shore locations and further out to sea. There are a wide variety of wave energy exploitation concepts, and more than 150 concepts can be listed to date in the following categories:

- Attenuator;
- Point absorber;
- Oscillating Wave Surge Converter;
- Oscillating water column;
- Overtopping/Terminator device;
- Submerged pressure differential;
- Other emerging technologies.

Until 2009, most of the concepts were still at an early stage of development, but the last few years have seen a considerable increase in the construction of reduced and full-scale demonstrators which may indicate a new trend that wave power could now become attractive commercially to developers and investors.

3.1 Review of Latest Developments

3.1.1 Full-scale Prototype Installations

The European Marine Energy Centre (EMEC) has seen a considerable increase in activity as a test site since 2009 for tidal energy converters, and a growing number of wave projects are planned for commissioning in the near future.

Aquamarine Power Ltd launched their nearshore "Oyster" device in 2009. Oyster 1 demonstrated the feasibility of using wave energy to pump high pressure water to an onshore hydro-electric turbine to generate electricity. The machine was removed in spring 2011 for analysis. A second Oyster wave energy device is due for commissioning, which will be followed by two further Oyster devices in 2012 and 2013, as part of a small array. Each Oyster 2 machine will have a generating capacity of 800 kW and will measure 26 m wide by 16 m high.

E.ON have deployed the first Pelamis P2 device. The second generation device, built by Pelamis Wave Power, arrived in Orkney in July 2010 and is undertaking a planned work-up programme of testing.

Finnish company Wello Oy will deploy their Penguin device in 2012. The Penguin is designed to capture rotational energy generated by the movement of its asymmetrically shaped hull. Constructed in Riga, Latvia, the device arrived in Orkney in June 2011. Approximately 30 m in length, the 1600 t device is expected to produce between 0.5 – 1 MW of power.

The Seatricity concept involves multiple floats travelling up and down with the waves, operating pumps to pressurize sea water, which is piped ashore to drive a standard hydroelectric turbine to produce electricity. The device is planned for deployment in 2012. Another development to take place in early 2012 in Portugal is the final testing and assembly of AW-Energy's first 3 × 100 kW WaveRoller power plant. The deployment is scheduled to take place in the waters off Peniche as soon as weather conditions permit.

Scotland-based AWS Energy has undertaken scale model testing in controlled tank conditions and to prove the manufacture, installation, maintenance, and durability of the flexible wave energy absorber membrane. A typical device will comprise an array of 12 cells, each measuring around 16 m wide by 8 m deep, arranged around a circular structure with overall diameter of 60 m. Such a device is capable of producing an average of 2.5 MW from a rough sea whilst having a structural steel weight of less than 1300 ts. The AWS-III will be slack moored in water depths of around 100 m using standard mooring spreads. A single-cell test apparatus is planned for deployment in UK waters during 2012.

The 150 kW PowerBuoy PB150 from Ocean Power technologies (OPT) was deployed at sea in 2011 for a series of test at a site approximately 33 nautical miles from Invergordon, off Scotland's northeast coast.

Port Kembla in Australia was the site of Oceanlinx Mk3 Pre-Commercial. The unit was deployed for three months, from February to May, 2010, and operated successfully during that time as one of the world's first grid-connected generators of electricity from

ocean waves. The device unfortunately broke free from its mooring in a storm and crashed into the breakwater and eventually sank.

There are many other studies reported e.g. Estefen *et al.* (2010) at different stages of development providing useful insights into structural and control challenges associated with this form of energy capture.

3.2 Current Research Activities

3.2.1 Numerical Predictions

Wave energy devices are varied but can be broadly classified into solid body, air-chamber, and overtopping devices. These types of devices lend themselves to analysis by well-established potential flow methods. Although simplified methods have been utilized with some success in parametric design studies, the most common analysis methods involve the use of floating free-surface body panel methods which are used extensively in ship and floating offshore platform design and analysis (Falnes, 2002). These numerical methods are implemented in a number of commercial computer programs and it is generally possible to incorporate air chambers into the analysis. Linear problems can be solved in the so-called frequency domain, but as nonlinearities and complex control strategies become important, time domain methods may have to be utilized. Frequency domain methods are relatively inexpensive to implement but have limitations. Time domain methods are more capable of taking into account nonlinearities and complex control strategies, but are more difficult to understand and time-consuming to implement. Significant research effort continues concerning wave prediction and the manner in which waves interact with tidal flow and sediment transportation (Previsic *et al.*, 2004, Warner *et al.*, 2010, Siddons *et al.*, 2009, Brown, 2010).

Numerical modelling of ships and floating offshore platforms has been affected by the advances in hardware and software computational capability. This has been particularly true with regards to the utilization of computational fluid dynamics (CFD) tools for hydrodynamic design. Although CFD has experienced tremendous development over the last several years it still has limitations with regards to analysing free surface problems.

3.2.2 Experimental/Concept Demonstration

Experimental verification of concepts is an important step before concepts are demonstrated on-site or at full scale, but model testing of wave energy devices suffers from the same limitations which floating body model tests do with the additional complications of scaling issues associated with the air chamber, and also power take-off and control mechanisms. It is critical to utilize large scale models in order to minimize the so-called “scale effects”. However, large-scale models are expensive to build and test and a limited number of facilities exist to test at very large scale. In order to eliminate scale effects and convincingly demonstrate a concept, full-scale demonstration is the eventual goal. However, full-scale demonstration is expensive and it may be difficult or impossible to measure or control the environment in order to convincingly demonstrate a concept.

3.2.3 Mooring Systems

Although not all devices are floating, fixed mooring is an important aspect for the floating devices. The mooring concepts utilized for wave energy devices are similar to those utilized in floating platforms. The mooring systems can generally be classified

into slack catenaries and taut moorings. However, in order to fully realize the potential of various concepts, the requirements of mooring systems may in some cases be more demanding than ships and floating offshore platforms. Moreover, if closely-spaced arrays of devices are considered, the interference of the mooring may result in a complex mooring arrangement (Falcao, 2010).

3.3 Power Electronics, Control

Power electronics for controlling devices and taking power are ultimately the most critical component. However, control and power electronics cannot be considered in isolation, but must be part of the overall system design. Moreover, the control and power take-off electronics must also consider an accurate physical model of the device in order to be fully effective. The issues associated with control and power take-off are similar to those which wind turbines experience, but are even more challenging due to the complex hydrodynamic interactions associated with random waves. The most significant issue is the fact that real sea waves are random. It is important to understand the amplitude and frequency variation of the sea waves. Optimal performance of wave energy devices occurs at resonance – i.e., when the device natural frequency and the excitation frequency are equal. Since the excitation frequency is constantly changing, some sort of active control mechanism is required in order to optimally capture the available wave energy.

4 TIDAL AND OCEAN CURRENTS ENERGY CONVERSION

This chapter presents an update of the previous reports of the ISSC specialist committee V.4, “Ocean, Wind and Wave Energy Utilization”, focusing upon different aspects of tidal energy conversion, with the addition of a section on ocean current energy conversion. Since 2009, many concepts have progressed to the stage of demonstrators being built and tested at model and at full-scale, which provide a valuable feedback in terms of structural reliability and a vast amount of in-situ experimental data for the design teams to review their initial design parameters. However, developers and suppliers, particularly in the area of electricity generation and power control systems, are still reluctant to make site data available to the wider research community, which does not favour the progress of research in this critical area of power control.

4.1 Tidal Currents Energy Conversion

4.1.1 Technologies

The 2006 and 2009 reports extensively covered the different concepts available at the time. Briefly, the mechanisms for tidal energy generation can be classified to date into four main categories, with the possible inclusion of an additional new technology based on vortex-shedding past cylinders:

- Horizontal axis tidal turbine (HATT);
- Vertical axis tidal turbine (VATT) ;
- Oscillating tidal hydrofoil (OTH); and
- Vortex-induced vibrating tidal cylinders (VIVTT).

HATT and VATT can be found in “open” or “ducted” configurations. Ducted configurations seem to be considered as they can present functional advantages, especially for maintenance, but also for the increased performance they seem to offer.

Depending on the depth of deployment of the converter, the structural foundation can follow one of the following types:

- Piled on the seabed with converter at a set depth;
- Piled on the seabed with converter at a variable depth (surface piercing);
- Moored from a floating structure;
- Guyed tower;
- Telescopic;
- Tethered; and
- Seabed standing under gravity.

4.1.2 Review of Latest Developments

Although tidal barrages are not the main consideration here, a brief mention should be made about the Severn Estuary project (UK). After the project of a tidal barrage was abandoned by the British government, an alternative project, the “Severn Tidal Fence Consortium” was set up to undertake a feasibility study to assess the potential for a large tidal fence system spanning the width of the Severn Estuary. The fence system would consist of a string of tidal stream energy converters spanning the estuary, with a free passage navigation gap. This new concept is claiming to have appreciable benefits when compared with tidal barrages, including reduced environmental impact, less disruption to shipping, and lower capital investment (Giles, 2010).

If vertical axis machines are mostly considered, it is interesting to note the deployment in 2009 of the 100 kW hydrofoil demonstrator, “Pulse-Stream 100”, by Pulse Tidal Ltd, in the River Humber in the UK, which is a shallow water site of 9 m depth. A 1.2 MW prototype is planned for commissioning in 2012 in the Isle of Skye Waters.

Among the latest developments, it is worth noting the “Tidal Flyer,” which is a novel, patented concept for extracting the kinetic energy from tidal currents developed by Open Ocean Energy Ltd in collaboration with HMRC of University College Cork. The fundamental concept of the device uses self-trimming tails in order to control the foils moving under water. The design ensures that the tail is always aligned to the apparent flow of the water and therefore keeps the main foils at the same angle of attack to the apparent flow and thereby creates maximum force within the cables. The system has been tested at model scale in the ocean engineering institute IFREMER. In a full system, the self-trimming tail oscillates from side-to-side to control the angle of attack of the main foils in each array. The resulting force in the cables is transferred to vertical shafts via the pulleys, which then feeds into the power take-off (PTO) system. This system has the advantage of operating at low current velocities of the order of 2 kn (1 m/s), which will have significant advantages in regard to the available sites for development and in the maintenance of the system.

A new technology based on the forces induced by a cylinder subject to vortex shedding is the VIVACE Converter developed by the University of Michigan under Vortex Hydro Energy LLC. This concept, which was, so far, at a laboratory scale experimental stage, is now being developed further as a demonstrator in the Detroit River.

Another similar concept is being developed in the University of Georgia (USA), which is a novel low-energy vortex shedding vertical axis turbine (VOSTURB). The rationale for this concept is to circumvent the inefficiencies and challenges of hydro-turbines in low velocity free tidal streams. VOSTURB aims to capture the energy of the vortices by installing a hydrofoil subsequent to a bluff body. This foil, free to oscillate, translates the vortex energy into oscillatory motion, which can be converted into a form of potential energy.

There continues to be fundamental concept development e.g. Bruder *et al.* (2011),

however the majority of research and development is now focusing on engineering optimization of components and devices e.g. Davies *et al.* (2011).

4.1.3 Full-scale Prototype Installations

Over the last few years, there has been a very significant increase of full-scale prototypes being commissioned. The UK has been particularly active, mostly at the European Marine Energy Centre (EMEC) based in the Orkney Islands. Installed as part of the Deep-Gen III project, co-funded by the UK government-backed Technology Strategy Board, the Rolls-Royce prototype tidal turbine was deployed in 2010 at the EMEC offshore test site off the Orkney Islands, Scotland. It is the first EMEC located project to both receive Renewable Obligation Certificates and to reach 100 MWh of supply to the grid.

The Atlantis Resources Corporation successfully re-deployed its AK-1000 tidal turbine on its subsea berth in the summer 2011, after initial trials in August 2010. This turbine has an 18 m rotor diameter, weights 1300 t, and stands at a height of 22.5 m. Voith Hydro and RWE Innogy commenced preparatory works in the summer of 2011 by installing the monopile for their 1 MW tidal turbine, which will be developed in 2012 through the joint venture company Voith Hydro Ocean Current Technologies.

Scotrenewables deployed its SR250 in March 2011 and reached the stage of power generation in September 2011. The testing programme involved towing the device to simulate tidal flow in a controlled manner before deploying the device in the Falls of Warne. DeltaStream, which is a gravity-based standing device developed by Tidal Energy Ltd, based in Wales, has a planned installation in 2012 in Ramsey Sound off the Pembrokeshire coast. Hammerfest Strom UK Ltd is planning to install the HS1000 tidal turbine at EMEC in 2012.

Outside the UK, the emergence of full-scale installations was enhanced in recent years by the support of several industry leaders across borders. For instance the Irish company, Open Hydro, partnered the French utility company EDF, and DCNS, the navy shipbuilder. The first 2 MW OpenHydro unit was towed from Brest on 31 August 2011 for deployment in 35 m of water on the seabed off the island of Brehat in Brittany.

4.1.4 Current Research Activities

General ocean engineering and coastal engineering developments are certainly useful in marine renewable energy in general, and particularly in tidal energy, but current methods still present some shortfalls, due to the specific design challenges of tidal energy converters.

Environment Modelling

Prediction tools for tidal and ocean currents modelling are available. In the case of tidal currents public entities, universities and private companies developed methods of prediction for a wide range of sea users (merchant service, fishing, and coastal management). These predictable data are of first interest for tidal energy. If the prediction of surface current is widely available, the prediction of the current profile at a given location is still very much in its infancy, and is, in fact, very complex, due to the fact that tidal currents are very much site-dependent (Lorke and Wüest, 2005, Liang *et al.*, 2008, Liang *et al.*, 2007). Such information, though, is critical to predict not only the power output but also the load cases, in terms of structural design. Data acquisition campaigns using Acoustic Doppler Current Profilers (ADCP) seem to be the only method currently available to collect large data sets, space- and time-wise, at

a given location. Such campaigns are, in practice, technically difficult in strong tidal spots, and costly to implement due to the multiple expectations from such data. If the basic knowledge of the current profile over few tide cycles can be sufficient for a device power prediction, the access to high resolution data sets becomes necessary for the analysis of turbulence levels at a given site.

Waves and Current Interaction

Three general approaches are considered to study the waves and current interaction:

- Semi-analytical methods based on Stokes development, Stream function, and Boussinesq in two dimensions;
- Numerical modelling of waves and current interaction in the spectral domain based on mitigation of waves modelling software (for instance, SWAN, TOMAWAC from EDF, WAVEWATCH from SHOM) and tidal software (for instance, MARS3D from Previmer). The results are spectral parameters of sea states and mean currents. The interaction with tidal devices are simplified;
- Numerical modelling of three-dimensional flow. Turbulence can be calculated based on basic approach by Navier Stokes, where turbulence can be calculated. In a simpler manner, Boussinesq (Bingham, 2009), or other methods based on wave dispersion (Belibassakis, 2011a). These methods are more able to take into account the detailed interaction with the tidal devices. The results are, at first, time domain parameters. Important studies in this area include Pinon *et al.* (2011), Rusu and Guedes Soares (2011), Mycek *et al.* (2011) and Maganga *et al.* (2010a,b).

Effects of Bathymetry

Smooth bathymetry effects can be taken into account by the different approaches presented above. Bathymetry effects are less accessible to semi-analytical methods. Some developments using diffraction and radiation context aimed to cope with sea bottom variation and the waves and current interaction (Belibassakis, 2011b).

4.1.5 Model Testing

In addition to prediction tools, model testing is an essential part in the selection of a turbine for a tidal energy converter, both to gain confidence in the performance and loads developed by the turbine. Most test campaigns focus on two main aspects: turbine performance, hydrodynamic loading, and dynamic behaviour of the overall device during deployment; and, in operation, when the scaling laws can be respected both for the turbine and the structure. An essential advantage of model testing is the access to data at a given angle of incidence where numerical models struggle to converge. There are inherent problems with model testing related to scale effects and the difficulty to reach the desired Reynolds number. The blockage effect (dimensions of model compared to tank section) must be considered with great care. One of the key aspects in selecting a test facility is the decision to use a water circulation channel or a towing tank. The main difference is the absence of turbulence in a towing tank, which can be of interest for some benchmark cases but in reality not pertinent as tidal flows can exhibit large levels of turbulence, which can be replicated in water circulation channels through the usage of variable size nets or honeycomb type devices. Generation of turbulence or sheared flow is easier in a flume tank and can be more realistic in many situations, such as development of the bottom boundary layer or free surface boundary layer, downstream effects of bathymetry, or downstream effects of other turbine wakes. Towing in waves and waves over current are not strictly equivalent,

depending on the Froude and Strouhal numbers. The kinematics of the flow on the turbine blades may be quite different. Wake measurements can be achieved through the usage of laser-based techniques, such as Laser Doppler Velocimetry (LDV) and Particle Imaging Velocimetry (PIV). LDV collects data in a punctual fashion, while PIV allows collection of slices of information in 2D, which greatly decreases the time required to assess the wake developed by a single turbine, and also when dealing with interaction effects, in the case of turbines in close arrays. The cost of PIV equipment is, however, restricted to a few research bodies with large public subsidies.

4.1.6 Deployment and Installation of Large/Full-scale Devices

Similar operations are routinely carried out in the offshore oil industry when installing sub-sea equipment. There is an existing fleet of vessels that could cater for the specific needs of the tidal energy industry, within certain limits. The technical capabilities of these vessels are very often far beyond what is required for the deployment of a tidal device, but also seem outside the range of budget available in the tidal industry. The most cost-effective concept seems to be the one of a barge towed by one or more tugs and service crafts. STX France Solutions, for instance, built a dedicated barge for the installation of the first 2 MW OpenHydro unit in 2011.

4.2 Ocean Current Energy Conversion

4.2.1 Resources

Ocean currents are strong, uni-directional surface currents located on the western boundary of the world's oceans. Ocean currents, especially the Gulf Stream and the Kuroshio Current, have been the topic of discussion over several decades as an energy resource. However, only one experimental ocean current turbine (OCT) has ever been deployed in the world. The experimental turbine was installed more than two decades ago in Florida current that is a portion of the Gulf Stream, off the coast of southeast Florida (Van Zwieten, 2011).

Global Circulation Models often fail to reproduce the structure of the current that is locally modified by topographic effect and by local winds. Re-analysis has to be adjusted to match the observed value (Duerr, 2010). The Kuroshio Current Power is estimated based on numerical models (Kodaira, 2009).

4.2.2 Design of Ocean Current Turbines

Many different designs are possible for Rotor Nacelle Assembly (RNA) of OCT. As with wind turbines, horizontal-axis rotor is dominant in the marine current turbine concept, because vertical-axis rotors are subject to cyclic loading even in uniform flow, and will result in fatigue loads (Senat, 2011).

Installation sites of ocean current turbines are mostly deeper than tidal current turbine installation sites. The RNA is supported just below the sea surface by floaters, and moored to the seafloor. Additional supporting structures are not necessary (Suzuki, 2009b). A contra-rotating type concept has been investigated as a more efficient concept and compared with a twin-rotor type concept. The comparison results show that the power-weight ratio of the contra-rotating type and the twin-turbine type are almost the same, and the ratio is comparable to that of bottom-mounted wind turbines (Takagi, 2011).

5 OCEAN THERMAL ENERGY CONVERSION

Theoretically, OTEC is a vast source of energy that is virtually limitless and sustainable. With the energy produced by OTEC, hydrogen and oxygen can be produced by

hydrolysis, the hydrogen and oxygen can then be liquefied, transported in cryogenic tankers to various destinations for use in space programs, fuel cell cars, industrial microchip manufacturing, power generation, etc. Fresh water can also be produced from seawater as a byproduct of the OTEC process, as the electric power produced during the process can also be used to power a reverse osmosis desalination process. Sea Water Air Conditioning (SWAC) has also been addressed for Pacific Ocean island buildings. Surface or small-depth sea water can be utilized as heat sources or sink for air-conditioning or heating by heat pump systems in temperate zones. High speed rail projects could very easily and efficiently be powered by alkaline fuel cell systems supplied by large-scale OTEC platforms (Energy Harvesting Systems, LLC, 2010). OTEC plants in the inter-tropical extraction belt could be considered to provide energy needed for oil and gas production, instead of burning a part of this production in thermal power plants.

5.1 Platform Design

The design of an OTEC platform depends on the weight and volume of the components on the structure, as well as the operating sea conditions. These platforms are usually quite large, in order to stabilize the structure against wave motion, improve its seakeeping performance, and reduce stress on the cold water pipe. Various designs of the OTEC platform and mooring systems have been considered by researchers over the past few decades. The simplest design is the rectangular barge type, such as the first MINI OTEC. The other four most complete OTEC design concepts offered in the 1980s were the spar OTEC plant by Lockheed Missiles and Space Co. (Trimble, 1975), the spar-buoy OTEC plant by Carnegie-Mellon University (Lavi, 1975), the submerged catamaran OTEC plant by the University of Massachusetts (Goss *et al.*, 1975), and the cylindrical surface vessel OTEC plant by TRW Inc (TRW Systems Group, 1975).

The grazing OTEC plant ship equipped with a propulsion system was proposed and designed by the Applied Physics Laboratory of Johns Hopkins University (Sasscer and Ortabasi, 1979). This OTEC plant is able to 'roam' the Pacific and Atlantic Oceans in order to seek for a high temperature differential. The OTEC tugboat concept was later proposed for the same purpose, but without the need to install a propulsion system. The surface ship design and the submerged cylindrical design have also been considered (Kamogawa, 1980) in order to meet the rough sea conditions around Japan.

A jacket-spar (J-spar) type of OTEC power plant was proposed by Srinivasan (2009) from Deepwater Structure, Inc. The J-spar configuration is able to suppress the alternate formation of Karman's vortex streets produced by underwater currents. Srinivasan (2009) also proposed the tension-based tension leg platform (TBTLP) in which an artificial seabed is utilized at an intermediate depth from the real seabed to support the TLP vessel tether system, thereby effectively enhancing the capacity of the tethers and reducing the sway and surge motion of the TLP.

5.2 Cold Water Pipe System

Ocean Thermal Energy Conversion platforms must have appropriate pipe technology to draw deep cold water for the process. The design, construction, and deployment of the cold water pipe (CWP) may be based on knowledge and experience gained from offshore risers. CWPs for OTEC plants are massive and subject to huge stresses at the joint between the cold water pipe and the OTEC platform. These stresses come from a combination of severe weather, wave action, and the length (more than 1000 m),

diameter (10 m for a full-scale 100 MW OTEC plant), and mass of the cold water pipe. It is worth noting that the Japanese have built a CWP (named TAKUMI) that upwells deep ocean water of $100,000 \text{ m}^3/\text{day}$ from 200 m depth and discharges it into the euphotic surface level to enrich the nutrients, in order to increase fish production in the surrounding sea area (Ouchi, 2009).

5.3 Heat Exchanger System

A major cost of OTEC power plants lies in the heat exchanger. The most common heat cycle suitable for OTEC is the Rankine cycle, using a low-pressure turbine. Two main types of the Rankine cycle heat exchanger are used in OTEC – i.e. the closed Rankine cycle process and the open Rankine cycle process. The earlier design of the OTEC closed-cycle heat exchanger was of the shell and tube type. The Alfa Laval plate heat exchanger was successfully applied in the 50 kW MINI OTEC pilot plant (Laboy *et al.*, 2011). Uehara of Saga University then developed a titanium plate-type heat exchanger (Uehara *et al.*, 1978). An experimental study for a 30 kW OTEC plant using the Uehara cycle was carried out by Ikegami *et al.* (2008), where a new embossed plate heat exchanger functioning as an evaporator and a condenser was adopted to decrease the pump power.

Other types of working fluids were also considered in the closed-cycle system. By taking into account the size reduction of the heat exchangers and the piping cost, ammonia was found to be the best working fluid in the OTEC closed-cycle heat exchanger. Recently, experimental studies and dynamic model simulations were carried out to investigate the efficiency of the OTEC heat engine when the ammonia/water mixture is used as the working fluid (Ikegami *et al.*, 2008, Sathybhama and Babu, 2011, Wagar, 2010). Hans Krock has adapted the Kalina Cycle for OTEC in collaboration with Recurrent Engineering, the patent holders of the Kalina Cycle (Energy Harvesting Systems, LLC, 2010).

In contrast, the open-cycle system used the vacuum flash vaporization of warm water to drive a low-pressure steam turbine. In order to reduce the impact of released non-condensable gases during the vacuum flash-evaporation process, a pre-aeration chamber could be installed below the flashing chamber so that gas molecules could be removed before entering the steam turbine (Energy Harvesting Systems, LLC, 2010). Such design will result in more efficiency, as well as the environmental benefits of oxygenated discharge water. Additionally, it could prevent the discharge of carbon dioxide and other greenhouse effect gases into the atmosphere (Kong *et al.*, 2010). Srinivasan and Sridhar (2010) have also proposed an OTEC engine that uses subsea technologies, such as the subsea condenser, subsea pump, submerged evaporator, and independent floating-pipe buoy platform to transport working fluid from the turbine outlet to the subsea condenser.

6 SUMMARY AND CONCLUSIONS

Recent developments in offshore wind technology have led to the deployment of large-scale offshore wind farms based on bottom-fixed support structures. There have also been in the past few years a number of concept demonstrators of floating wind turbines. However, the primary focus and major challenge today is in capital and operational cost reduction, which requires substantial research work to achieve cost-effective design, especially with regard to support structures and foundations.

Although design rules for bottom-fixed offshore wind turbines are in development, design rules for floating wind turbines need to be initiated and should cover a large

variety of floating concepts. Experience in the offshore oil and gas industry is valuable for such developments, but design requirements should be properly considered due to the unmanned nature of and economic constraints for fixed and floating wind turbines. Design tools, particularly rapid numerical tools for floating wind turbines need to be further developed and validated with model-scale and full-scale tests. In order to properly carry out structural design, these tools need to address not only aerodynamic and hydrodynamic loads, but also the structural responses of the rotor, tower and floating/mooring systems, as well as control strategy. In this respect offshore wind structural analysis can be more complex than for oil and gas structures however, machine loading can be monitored and controlled meaning greater scope for progressive life-cycle strategies.

Whereas to date monopole structures in relatively shallow waters (up to 40 m) have dominated, there is an increasing interest in jacket and tripod support structures for the 40 – 60 m water depths particularly for the 5 MW and larger structures. It should be noted that the structures installed to date have only been in operation for a relatively short period of time and not without some significant issues including integrity of grouted joints, internal corrosion and fatigue issues concerning transition pieces. The coming years will see a vast amount of field data and experience which will be important to learn from so that future designs can take full advantage of this experience and of the new emerging modelling techniques. In addition, focus will be on array optimization and cost reduction for large volume manufacture, deployment and operation.

The challenges associated with wave energy are similar or even greater than those associated with wind energy, mostly due to the wide variety of devices which have been proposed and to the fact that only a very few have been demonstrated in full scale. Although a larger number have been model-tested, many of these potentially good ideas have not been fully analyzed. Wave power has tremendous potential but for the moment is some years behind offshore wind. Cost reduction, particularly for moorings and deployment systems in addition to fatigue resistance are the primary challenges.

Tidal Stream is emerging rapidly as a serious commercial prospect with no fewer than ten large-scale concept turbine demonstrators since the last ISSC report. The predictability of the resource and relatively gentle manner in which turbines are loaded (compared to wave power) make it attractive. However the primary challenge is deployment in tidal flows greater than three metres per second and fixing to the sea bed. In this case the current costs of deployment and installation dwarf the cost of the supporting structure and it may be that tidal stream can only become economically competitive when array installations begin to use specially designed deployment vessels and machinery which have yet to be developed. Other issues are turbulence, resistance from debris and wake interaction effects, all of which are active areas of research and development. OCEC is related and is now receiving considerable attention; developments are likely to follow the Tidal Stream pioneering work as investments required are larger due to water depth and distance from shore.

Challenges for OTEC commercialization include its high cost, due to its large scale; finding an appropriate location that could leverage on the offshore oil and gas industry with respect to installation and specialized vessels, station-keeping systems, and support; and the concern over environmental risk (Cooper *et al.*, 2009, Cohen, 2009). Despite these challenges, it is reasonable that OTEC plants in one form or another will appear on our oceans.

7 ACKNOWLEDGEMENTS

The committee is grateful to Ms. Helle Slutz and Mr. Steven Spearman of the USCG for their work in editing the report for readability and typographical errors.

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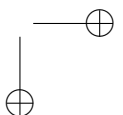
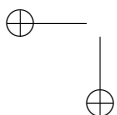
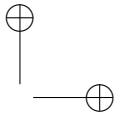
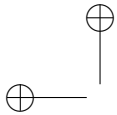
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18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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18th INTERNATIONAL SHIP AND
OFFSHORE STRUCTURES CONGRESS

09-13 SEPTEMBER 2012
ROSTOCK, GERMANY

VOLUME 2



COMMITTEE V.5
NAVAL VESSELS

COMMITTEE MANDATE

Concern for structural design methods for naval ships and submarines including uncertainties in modelling techniques. Particular attention shall be given to those aspects that characterise naval ship and submarine design such as blast loading, vulnerability analysis and others, as appropriate.

COMMITTEE MEMBERS

Chairman: Robert Dow
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Joep Broekhuijsen
Raphael Doig
Albert Fredriksen
Akihiko Imakita
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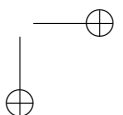
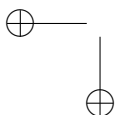
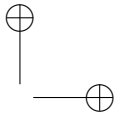
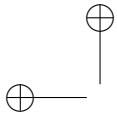
KEYWORDS

Naval ships, submarines, classification rules, design criteria, progressive collapse, lightweight materials, military load effects, shock, blast, underwater explosions (UNDEX).

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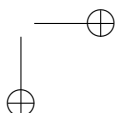
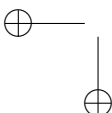
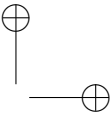
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1 GENERAL DISCUSSION – SIMILARITIES AND DIFFERENCES BETWEEN NAVAL AND COMMERCIAL STRUCTURAL DESIGN

1.1 Introduction

Naval ships and commercial ships have lived side by side for a long time, but under different technical regimes. During the last ten years, the world military situation has changed and with this, also naval ship procurement and design processes. The use of classification rules are being used more and more for naval ship design, and with this we see a closer relation between naval and commercial ship design.

This chapter will try to highlight the similarities and differences between the two, and extract the main areas that should be considered from two points of view:

- Areas where naval and commercial ships benefit from the same pool of technical knowledge
- Areas where naval ships are inherently different from commercial ships and where the commercial methods or thinking may lead to a less fit naval ship design

The other chapters of this report will explore this in more depth both with respect to typical military load effects such as blast loading and submarine hull collapse.

1.2 Some Historic Notes on Naval Structural Design

Looking at naval structural design over the last decades may illustrate some of the differences and similarities. During the cold war, a lot of effort was put into each structural design. Each new design was going to be the “formula one” of its type. Also speed seemed to be more important than today. The result on structural design was an optimised, weight sensitive thin plate structure, with high emphasis on details to enhance damage tolerance. Cost was not the primary focus here.

The most typical commercial development in the same time period was the significant growth in size, especially for tankers, bulk carriers, container vessels and cruise ships. The main focus here was production cost and thereby production friendly structural design details. Weight was less important, and the industry could live well with the minimum thicknesses specified by the Class Societies (to give an acceptable level of robustness, greater than necessary to withstand the rule loads).

It is now more than 20 years since the cold war ended. Navies and shipyards have been forced to adapt to a new situation. It is no longer so clear what the future job of the warship will be, with new missions like joint -peacekeeping missions, pirate operations etc. Underlying factors like speed seems less important, and cost seems to be more important. Also, during this period, Class Societies have entered the scene of warship design. The use of Class services and Class Rules as a technical standard for design and building of warships is now quite common. Through this new partnership, the naval and commercial shipbuilding practice meets. The end result of this seems to be a more pragmatic structural design, that may be less optimised, but with a general robustness as for other ship types. One may say that naval and commercial structural designs are starting to merge.

In this way one may say that we get the best of naval and commercial structural design. But what about the military loads and damage tolerance? This is further discussed at the end of this chapter.

1.3 Which Differences?

In order to get deeper into the subject it is necessary to establish the main categories for sorting of different hull design parameters.

- Different design values: in this case naval and commercial ship structures are based on the same load effect and formula, but they are at different points on the scale. For this reason, these items are categorised as similarities.
- Generic differences: in this case naval and commercial ships are subject to different type of load effects that requires different methods. These are further discussed under “differences”.

1.4 Similarities

Naval and commercial ship structural design has a lot of similarities for obvious reasons. They operate in the same environment and the laws of physics are the same, not influenced by ship types. Common structural parameters are listed in the Table 1.

We see from the table that most of the ships specifications are similar, and that the main differences are related to military requirements such as damage tolerance and survivability after damage.

Based on the above, it can be concluded that for the normal environmental loads and load conditions, the main structural elements are dimensioned in a similar way for both naval and commercial design. The differences that can be found are mainly related to the values and not the principles. This just reminds us that naval and commercial ships follow the same laws of physics, hydrodynamics etc.

The common link between all the loads listed in Table 2 is that they are the result of the ship’s safety under operational and environmental loads.

One example where a common problem may be treated differently in naval and commercial designs is “fatigue crack management” where a typical commercial approach will be to design the structure with a margin to avoid cracks, and naval approach may be to calculate how long operation can be continued with an existing crack so the ship can continue to “fight” after being damaged.

Table 1: Similarities and differences in specified use




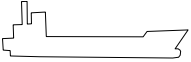

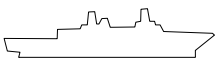
	Cargo ship 	Cruise vessel 	Frigate 
General specification: Similarities	Worldwide operation Open sea and coastal waters All weather High reliability, Year round operation Survive all weather and sea conditions Docking at planned intervals, plus emergency situations Grounding/collision damages	Similar to cargo ship	Similar to cargo ship
General specification: Differences	Moderate survivability Moderate damage tolerance Carry cargo	High survivability Moderate damage tolerance Carry passengers	High survivability High damage tolerance Carry weapon, sensors, personnel Military design requirements Enemy weapon damage Military loads Damage tolerant

Table 2: Structural similarities

Ship element	Cargo ship 	Cruise vessel 	Frigate 
Hull Bottom, Hull Sides, Bow	Sea loads and slamming loads. Speed and wave induced	Same*)	Same*)
Main deck	Local sea pressure, green seas, global hull girder loads	Same*)	Same*)
Watertight Bulkheads	Hydrostatic pressure	Same*)	Same*)
Superstructure	Deck loads, "sea loads", acceleration loads		
Internal decks	Local deck loads	Same*)	Same*)
Tanks	Local pressure, (filling, acceleration loads, pump pressure)	Same*)	Same*)
Foundations (engines, winches, etc.)	External loads, acceleration loads	Same*)	Same*)

*) Same load effect, but different values

1.5 Differences

When looking for generic differences between naval and commercial structural design, the most important differences are related to the military loads. Commercial ships are designed and operated to avoid damages. Those damages that cannot be totally avoided are termed "foreseeable damages", normally grounding, collision, and fire, and are in simple terms covered by double bottom, collision bulkhead, and fire insulation. Naval ships on the other hand, need to be better prepared for damage from enemy weapons in a warlike situation. This requires damage tolerance and survivability. A number of military loads are listed in Table 3.

The differences identified here are generic differences where there are little or no similarities between naval and commercial structural designs.

The items listed in Table 3 have one thing in common: they are the result of the Navy's performance requirements under warlike situations.

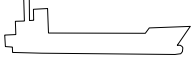


1.6 Military Loads

Based on the results from the Tables 1 - 3 it can be concluded that the main difference between naval and civilian structural design are the military load requirements, and these will be further commented here. A general picture of the survivability elements is shown in Figure 1.

The items in Figure 1 that affects the structural design of a frigate are mainly: weapon effect, damage, and recoverability. Some of these areas are covered in more detail in the current ISSC Committee V.5 report:

- military loads: Chapter 4
- residual strength: Chapter 5
- air blast: Chapter 6

Table 3: Generic differences between naval and commercial structural design

Load type	Cargo ship 	Cruise vessel 	Frigate 
Air blast	Not Applicable	Not Applicable	Relevant design case, local design considerations
Underwater Shock	Not Applicable	Not Applicable	Relevant design case, affects hull girder, local design and foundations
Fragmentation	Not Applicable	Not Applicable	Relevant design case, local protection
Residual damage requirement	Limited to the "foreseeable damage" loadcases	Limited to the "foreseeable damage" loadcases	Relevant design case, redistribution of strength elements
Magnetic signature	Not Applicable	Not Applicable	Relevant design case, limits on material selection
Stealth characteristics	Not Applicable	Not Applicable	Relevant design case, limits on hull shape
Damage tolerance (ruggedness)	Ruggedness based on normal scantlings are considered sufficient	Ruggedness based on normal scantlings are considered sufficient	Relevant design case, improved structural details

1.7 Submarines

It is difficult to make a comparison between surface vessels and submarines, as they have very different operational modes. However, the difference in military structural requirements can be seen from Figure 1. As the frigate is designed for strength in both ordinary and a number of military damage load cases, the submarine survivability relies mainly on its ability to avoid detection. For this reason the main dimensioning load case is to prevent collapse from external water pressure when diving and withstand operational loads. This is covered in the current ISSC Committee V.5 report Chapter 3 on submarine pressure hull design.

1.8 Relation to Rules and Regulations

It has been identified above that most of the structural design requirements for naval vessels are coming from the environmental and operational loads, and some from additional design requirements for military load cases.

When looking at Classification Rules from some of the major Class Societies that cover naval craft, we see that this is also reflected in the Rules. The large part of the

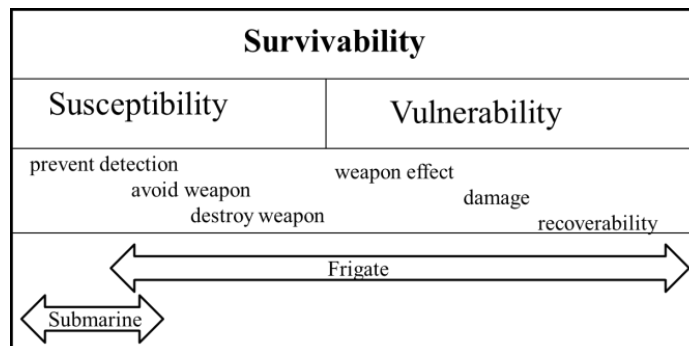


Figure 1: Survivability elements for a typical frigate and a submarine

structural requirements is similar for naval and commercial ships, although with some minor differences and different values. The specific military requirements to the ships structure represent a smaller part of the structural design criteria.

1.9 Concluding Remarks

From the discussion above it can be concluded that the larger part of the structural methods and calculations are common for naval and commercial ships, only with minor differences in characteristic values. This means that naval and commercial ship structural design can benefit from a common source of research and development of structural design methods. It also confirms the basis for using Classification Rules (so far, based on commercial ship experience) as a technical standard for naval ship structures.

Another conclusion that can be made is that the generic differences in structural design between naval and commercial ships are mainly related to the military load cases. For this area there is little common ground for exchange of methods and experience between naval and commercial structural design.

Seen in a broader perspective, the above conclusions raise some worrying questions for the naval community. The common knowledge basis for structural design through Classification Rules and Class Societies service experience is enormous. On the other hand, the knowledge basis for the military loads is small compared to this. As an example: a medium size Class Society like Det Norske Veritas is logging close to 6000 years of service experience per year for civilian ships. On the other hand, the corresponding service experience for naval ships is in the order of 100 years combined experience per year. In addition to this, the specific service experience on military loads is practically none. The question is then: How is the military loads taken care of in the future? How will the technical basis be maintained, and how will the personal knowledge and skills be maintained in the future? Having said this there is a wealth of knowledge and experience on military load effects which resides in Navy's around the world. This experience is the product of an enormous effort on shock testing of naval structures and equipment and this information has been distilled into standards and guidelines for the design of naval vessels against weapon effects. If in the future Naval Vessels are going to use Classification Society Rules for design then this information has to be made available to the Classification Societies.

It is advised that the next ISSC naval committee focuses on the military loads, vulnerability and residual strength of naval ships.

2 OPTIMIZATION OF NAVAL STRUCTURES USING LIGHTWEIGHT MATERIALS

2.1 Why Consider Lightweight Materials?

Lifecycle cost and mission capability are the standards to which any naval ship building program is to be evaluated. Cost and functionality are competing interests in a program where greater spending is thought to yield a vessel with better capabilities. However, some design parameters may be optimized for better performance at a lower cost. Structural weight is one such parameter in that decreasing weight lowers material costs and reduces the power demand throughout the service life. Reducing the power demand increases the vessel's fuel efficiency, endurance, speed, and/or tonnage carried. Furthermore, there may be auxiliary benefits of maintenance cost savings, corrosion protection, or stealth improvement from changing the structural material

from ordinary strength steel to lightweight options. The objective of this section is to discuss alternatives to ordinary strength steel construction of naval vessels for cost savings and mission capability improvement while maintaining a consistent level of safety compared to conventional designs.

“Lightweight” materials can be defined as those that have a greater strength to weight ratio than ordinary strength steel. When properly engineered and fabricated, lightweight materials provide the same strength at a lower total structural weight. However, stiffness, fatigue strength, flaw criticality, and fire protection are just a few of the design parameters that will change when designing with an alternative structural material.

2.2 Requirements and Decision Criteria for Naval Vessels

To be considered feasible, any new technology employed in ship building must be capable of withstanding the marine environment and of being fabricated in a conventional shipyard. Furthermore, seaway and military specific loads impose harsh conditions that further bracket material usage. The resulting loads put heavy fatigue demand on the structures that must be accounted for in the design. Extreme loading is another example of a common restriction where heat tempered 6xxx series aluminium lacks ductility such that the US Navy does not allow its use in hull applications; i.e. shock loading (ABS HSNC, 2006).

The following decision criteria can be used to evaluate the total value of a change in material system:

1. *Lifecycle Cost Reduction:*
 - (a) *Relative capital investment*
 - (b) *Operation: Reduced power demand (via fuel economy and smaller power plants)*
 - (c) *Maintenance: Inherent corrosion protection*
2. *Mission Capability Improvement:*
 - (a) *Increased: speed, endurance, and/or tonnage carried*
 - (b) *Improved stealth by thermal insulation or reduction of RADAR / magnetic signature*

2.3 Lightweight Materials as Means for Optimization

Table 4 offers a qualitative breakdown of how high strength steel, aluminium, titanium, and FRP compared to ordinary strength steel. There is considerable statistical variation in much of the data used to develop the table and the selection of grade, temper, as well as the geometric arrangement that is as important as the material selection. Therefore, the qualitative conclusions are somewhat relative. However, the data used are taken from a very appropriate range of alloys, grades, and tempers for metals and fibre types and lay-ups for FRP used the marine industry. Where distinctions in the data are made, thinner metals (i.e. less than 13 mm) and high quality FRP are presented. Naval projects tend to favour high quality construction using more refined (i.e. thinner) scantlings with optimized properties.

To provide a better linkage between weight savings and changes in material system, a sample calculation has been performed and summarized below in Table 5 and Table 6. Using mechanical properties for different material systems, the resulting flexural stiffness (EI), bending moment, shear force, and the weight per linear foot compared to an ordinary strength steel section were calculated for sections that have roughly the same maximum bending strength. The geometry is selected such that each section is

Table 4: Limit States/Mechanical Properties

Limit State Criteria	Ordinary Strength Steel (350 MPa)	High Strength Steel (550 – 690 MPa)	Aluminium		Titanium (Ti-6Al-4V Gd. 5)	Fibre Reinforced Plastic	
			Conventional Welded Plate (5xxx series)	Extruded Panels and Shapes (6xxx series)		Glass Fibre	Carbon Fibre
Strength	Average strength. Excellent strength above yield and ductility.	Excellent strength. Excellent to average strength above yield and ductility.	Low strength (Good strength if weight is considered). Good strength above yield and low to average ductility.	Low strength (Good strength if weight is considered). Low strength above yield and ductility. ¹	Superior strength (very good if weight is considered). Average strength above yield and low ductility.	Excellent strength (better if weight is considered). Brittle failure at design load. Very low ductility.	Superior strength (best if weight is considered). Brittle failure at design load. Very low ductility.
Deflection	Average (baseline).	Increased deflection ² due to thinner scantlings.	Equivalent / decreased deflection. ² However, stiffness is reduced at loads near yield due to a highly non-linear stress-strain relationship.		Increased deflection ² due to thinner scantlings.	Increased deflection. ² Potential for deflection creep under long term static loads.	
Vibration High-amplitude loads (i.e. blast) are not specifically addressed	Well controlled by minimum plate thicknesses; does increase weight. ³	Thinner scantlings are more susceptible to vibrational loads. ³	Low-amplitude, high frequency loading, i.e. propeller vibration, may endanger welding. ³	Beams and extruded members have equivalent stiffness that respond similarly to ordinary steel. ³	Thinner scantlings are more susceptible to vibrational loads. ³	Typical panel construction can lower natural frequencies into a range that is resonant to ship motions. Conversely, non-linear dampening effects tend to restrict low-amplitude, high frequency inputs (i.e. noise). Glass members will have higher inertias and better performance compared to carbon.	
	Noise is a common problem on metal ships. Welded metal joints transmit low-amplitude, high frequency loads very well and restricting <i>all</i> plates to non-resonant frequencies for all loads is not practical.						
Buckling	Average (baseline).	Thinner scantlings will have lower buckling resistance (global and local). ²	Average buckling resistance. ² However, a soft tangential modulus at high stresses will weaken the inelastic buckling resistance.	Beams and extruded members have equivalent stiffness that respond similarly to ordinary steel. ²	Thinner scantlings will have lower buckling resistance (global and local). ²	FRP panels, with or without-hat stiffeners, are difficult to associate directly with stiffened steel plates with-respect-to buckling. FRP panels are perfectly viable to resist buckling loads, but local buckling mechanisms and minimum skin thickness requirement may offset some strength weight savings.	
Flaw Criticality/Fatigue Strength ⁴	Average (baseline) flaw criticality and fatigue life.	Good energy absorption (crack arresting). Average fatigue life. ⁵	Low energy absorption (susceptible to cracking) and low fatigue life. ⁵		Average energy absorption. Low fatigue life.	Generally good energy absorption (crack arresting), however interlaminar peeling may or may not be a weakness. Good to superior fatigue life.	
Other	Limit states are well defined and excessive conservatism is easily eliminated.	Relationship to ordinary strength steel is established and some conservatism may be eliminated.	Some advanced limit state design criteria are available. Degraded performance, compared to steel, at high strain rates.	Significantly reduced strength once welded. Generally not allowed in high strain rate environments (i.e. shock applications) due to lack of ductility.	As a relatively novel material system, advanced material characteristics and responses to limit states are not well defined for marine applications.	Very poor resistance to abrasion. Impact resistance varies depending on the material system. Structure may be extremely weak in loading normal to the ply plane. Energy absorption in extreme loading conditions, including high strain rates, is very good (i.e. progressive failure), but will result in the loss of the structure.	
	Very good energy absorption under high strain rate (i.e. blast / extreme bending).						

¹ 6xxx series aluminium has not been accepted by the US Navy for use in shock loading applications due to its lack of ductility.

² Well-proportioned (compact) aluminium, titanium, or FRP members of the same weight as an ordinary steel member will have greater stiffness; glass FRP is the expectation in that it was equivalent stiffness. However, lightweight scantlings can meet the same strength requirements with much less weight. High strength steel, titanium, and carbon FRP scantlings tend to be thinner than steel and will have higher deflections under the same load while aluminium and glass FRP will have thicker scantlings with the aluminium having roughly equivalent and the glass FRP having less stiffness.

³ Metal framing is a very efficient conductor of vibration energy and care must be taken to ensure that modes of adjacent structures do not interact.

⁴ Flaw criticality is different from fatigue life in that flaw criticality represents the amount of energy that may be absorbed while a crack opens under a constant load and fatigue strength is the static equivalent maximum stress that a variable (cyclical) load may have such that the ultimate strength after many cycles, i.e. $N = 10^6$, is less than the yield strength of a single cycle load.

⁵ High-alloy steels and aluminiums hardened to increase their yield strength may exhibit poor fatigue strength. Welding and surface finish are major factors in performance.

compact, well proportioned against localized buckling, and optimized for maximum flexural strength and stiffness for the given weight. Standard shapes, i.e. wide-flanged beams readily available from mills, were used, and stiffened plates are assumed to be analogous to the beams. A wide-flanged beam is not a perfect analogy for a FRP member because use of open FRP shapes is rare and most structures are either “hat-stiffened” panels or closed box sections (Green, 2011). However, the FRP results are good for comparison purposes given that designers will try and maximize the section inertia similar to a wide-flange beam. All the results presented below should be interpreted as the upper bound of the strength to weight optimization because no other limit states are evaluated. No shear data is presented for FRP because it is likely that the FRP member will include a cored panel and/or different lay-up at the peak shear components. Shear in FRP sections is generally not a problem for distributed loads and will not impact the weight considerations.

Load uncertainty and consequence of failure are separate factors. The variation in bending moments is due to the fact that real sections were used and the percent difference is defined as:

$$\frac{(Value_{lightweight} - Value_{nominal})}{Value_{nominal}} \cdot 100\% \quad (1)$$

It is readily observed that lightweight sections with the same maximum bending moment capacity as ordinary strength steel will have much less flexural rigidity, except for aluminium which is almost equivalent, and lower shear strength; except for the titanium sections. Conversely, the sections optimized for maximum bending moment with the same weight show that lightweight members can be as stiff or stiffer. The major implication here is that the designer of lightweight structures has to calculate all of the limit states directly because one cannot assume that just because the strength

Table 5: Percent Difference of Lightweight Sections to Ordinary Steel Sections of the Same Strength*

	350 MPa Steel	550 MPa Steel	Aluminium	Titanium	Glass FRP	Carbon FRP
Flexural Stiffness (EI)	-34 %	-60 %	-8 %	-85 %	-89 %	-79 %
Bending Moment	5 %	3 %	0 %	7 %	4 %	4 %
Shear Force	-21 %	-39 %	-24 %	41 %	No data	No data
Section Weight	-24 %	-49 %	-75 %	-254 %	-383 %	-472 %

* Strength is taken as the initiation of yielding, or first ply failure for FRP given by the product of the allowable stress and the section modulus. The allowable stresses are base on Load and Resistance Factor Design (LRFD)

Table 6: Percent Difference of Lightweight Sections to Ordinary Steel Sections of the Same Weight

	350 MPa Steel	550 MPa Steel	Aluminium	Titanium	Glass FRP	Carbon FRP
Flexural Stiffness (EI)	0 %	0 %	81 %	60 %	-7 %	185 %
Bending Moment	43 %	96 %	91 %	533 %	565 %	802 %
Shear Force	43 %	96 %	8 %	340 %	No data	No data
Section Weight	0 %	0 %	-2 %	-2 %	-3 %	-3 %

criterion is satisfied that other criteria, i.e. deflection, vibration, or buckling, are satisfied by extension. Some commercial codes, i.e. the ABS Steel Vessel Rules, ABS SVR (2010) envelope limit states by requiring minimum scantling sizes based on experience and service history. For the maximum efficiency, a designer may start with a section that is optimized for strength, having a bending moment equal to ordinary strength steel, then increase the thickness until the stiffness of the member satisfies all of the other limit states.

The conclusions above are based on a sample calculation of a wide-flange beams that range from 500 to 1,000 mm in depth. An entire ship's hull will have much different results with less benefit from lightweight materials. Given that the "beam" components of a ship, i.e. the deck and shell plates, are very thin compared to their distance from the neutral axis, the offered inertia is almost solely based upon the area and the distance from the neutral axis; as evidenced by the parallel axis theorem:

$$I_i = I_o + A \cdot d^2 \quad (2)$$

Therefore, the stiffness of a lightweight ship will be based on the product of the global inertia and the elastic modulus of the structural material which are both directly proportional, first-order, to the area and the elastic modulus respectively. Materials that have significantly less modulus than steel, i.e. one third, and comparable weight savings, i.e. densities of 2.5 – 5.0 times less than steel, will likely yield a ship with much less stiffness for the same global strength. To counter act this loss of stiffness, additional material is required and will negate some weight savings.

2.4 Further Challenges for Mitigation of Weight in Naval Vessels

Table 7, included on the next page, offers some additional topics of interest which are discussed further below. While corrosion of steel structures, both ordinary and high strength, tends to yield an advantage to lightweight structures, fire loads, fabrication/repair issues, and weld ability tend to offset strength to weight advantages for lightweight materials in favour of steel.

2.4.1 Structural Fire Protection

Safety in a fire event is of primary concern for any vessel and naval vessels in particular. The major difference in naval vessels in a fire event to a standard commercial vessel is the naval vessel's force is active in fire suppression as opposed to commercial vessels relying almost solely on passive fire suppression systems. Conventional steel ship design practice has two features that are relevant to the discussion of lightweight materials: 1) steel is non-combustible and cannot add to the fire load at any ignition temperature and 2) when combined with insulation, steel decks and bulkheads form fire boundaries that restrict a fire's progression (IMO SOLAS, 2009). For a lightweight structure to have equivalent safety to a steel vessel, these principles must be replicated. FRP construction represents the largest departure from conventional structural fire protection so particular attention is paid to the establishment of FRP fire safety in this section. The same solution methods are applicable to other lightweight materials with different quantitative results.

The matrix material of FRP is hydrocarbon based and therefore combustible. Furthermore, temperatures in excess of 50 – 200°C (nominal 95°C) can render the laminate unstable (Hull and Clyne, 1996). Given that shipboard fires can reach temperatures several times larger than this critical temperature range, 935°C, protection must be given to the FRP to maintain the structure. The Swedish government has undertaken

Table 7: Other Behaviours Affecting Structural Weight in Naval Vessels

Other Properties	Ordinary Strength Steel	High Strength Steel	Aluminium		Titanium (Ti-6Al-4V Gd. 5)	Fibre Reinforced Plastic	
	(350 MPa)	(550 – 690 MPa)	Conventional Welded Plate (5xxx series)	Extruded Panels and Shapes (6xxx series)		Glass Fibre	Carbon Fibre
Corrosion	Poor corrosion resistance (baseline). Scantlings require increased thickness (weight) and/or continual maintenance to account for loss. Tertiary structures, i.e. fan intakes, on steel ships are often not steel because of prohibitive corrosion consequences.		Excellent resistance to corrosion. No protection is required.		Good resistance to corrosion.	Superior resistance: FRP materials are generally inert with respect to galvanic corrosion. However, some FRP systems may require protection from corrosive chemicals if used as integral tanks for fuel or other fluids.	
Response to Fire Loads (Thermal Stress NOT Applicable to Typical Structures)	Excellent resistance to thermal loading and good load capacity at extreme fire events (baseline). However, steel has high thermal conductivity and will rise in temperature quickly. The critical temperature is about 605° C. ¹		Poor resistance to thermal loading and almost no capacity at extreme fire loads. Also, aluminium has very high thermal conductivity and will rise in temperature quickly. The critical temperature is about 195° C. ¹		Superior resistance to thermal loading. The thermal conductivity is less than steel with a critical temperature of about 700° C. ¹	Very poor fire resistance: the glass transition temperature ² of the matrix is around 93° C for good material systems. However, FRP has very low thermal conductivity creating an insulating effect in a fire. If not ignited, FRP will resist energy transfer much better than metal.	
Structural Fire Protection	Non-combustible adding nothing to the fire load (baseline). However, some insulation is required to form substantial fire boundaries.		Non-combustible adding nothing to the fire load. However, insulation is required to form fire boundaries including a considerable amount required for the highest boundary grades.		Although not standardized, SFP will be similar to steel, but with less magnitude.	Combustible and toxic at fire event temperatures. Insulation can stop combustion and form fire boundaries, but the weight of insulation offsets some weight savings.	
Fabrication / Repair	Conventional with average material costs (baseline).	Weld quality requires greater QA/QC, but is conventional. High strength steels will cost more than ordinary steel.	Welding of aluminium is a conventional technology but requires skilled workman and detailed fabrication procedures. Both the unit price of the material and the labor costs will be higher than steel.		Lack of certified welders and quality requirements makes fabrication and especially repair difficult.	Fabrication and repair requires shipyards trained in FRP construction. The initial costs of both the labor and materials will be higher than steel construction. ³ FRP construction may be new to some shipyards, but there are conventional processes that may be adopted.	
Weldability ⁴ (Bonding for FRP)	Good (baseline).	Average: cracking may occur if quality standards are not met.	Below average: welding will always impact aluminium's yield and ultimate strength.	Poor: heat treated tempers will have a significant reduction in yield reduction.	Poor: welds are very susceptible to trace gas impurities. Extreme welding costs may result from high quality requirements.	Average-Excellent. Nearly the full strength of the parent material can be achieved in FRP bonds. However, issues with the matrix of the material system can reduce the overall strength of bonding. Also, bonded joints can be made flat and will not have heat distortion.	
Other	Steel hulls are magnetic and visible to mines and sensors based on magnetism. Degaussing is an expense and troublesome process on all ships, but steel ships require significantly more effort to protect.		Alloys other than the 5xxx series noted are restricted based on corrosion resistance and weldability.			Very good sound/thermal insulation. However, FRP can be degrading in the marine environment both below and above the waterline. Water impregnation can “blister” the surface finish or even induce structural delamination. Furthermore, exposure to direct sunlight can breakdown the resin and/or reduce the fatigue life of the unprotected FRP.	
	All metals conduct electricity. In terms of grounding of equipment or diffusing harmful current this is a benefit; however, electrically conductive material in-way-of RADAR and other sensors may change the way they operate. Also, welding of metals will distort the plating, also called the “hungry horse” effect, thus increasing the RADAR signature.						

¹ The critical fire load temperature is taken as half of the absolute melting point. At the critical temperatures noted, metals will begin to lose substantial stiffness and strength.
² The glass transition temperature is the inflection point where polymers transition from a solid to a semi-solid "rubbery" state. In the rubbery state, the matrix material will be non-structural.
³ Mass production of FRP structures, i.e. panels, structural elements, or even entire hulls, is possible and will greatly reduce the unit cost of FRP despite the higher cost of materials.
⁴ Residual stresses can reduce the yield capacity of any welded structure; however weldability is taken to mean any additional degradation of yield strength or loss of ductility due to welding.

the “LASS” (Lightweight Construction Applications at Sea) project to help spread lightweight technologies from specialized markets, such as small naval craft and large luxury yachts, to larger commercial ships. Structural fire protection is key in this discussion as previous IMO SOLAS requirements virtually excluded FRP construction given the concerns previously noted. Also, little or no large scale testing had been performed prior to this study to establish FRP’s equivalency to steel (RINA Conference, 2011).

The conclusion of the LASS project was that FRP could indeed be protected to meet the industry standards. First of all, with minimal insulation, 1 kg/m^2 , any FRP surface can be “fire restricting” in that the temperature to the FRP is below the level that would release volatile gases which would increase the fire load and spread toxic smoke. Secondly, with significantly more insulation than steel, it was demonstrated that it could maintain both its strength and stiffness as a SOLAS fire division up to the most restrictive boundary type: class A with 60 minutes of load. The real discussion of structural fire protection for lightweight structures is that the increase in insulation weight offsets some of the weight savings in the material’s strength to weight ratio. For class A boundaries, FRP would require 6.85 kg/m^2 more insulation to be considered equivalent to steel.

2.4.2 Capital Costs vs. Lifecycle Savings

Lightweight construction has higher capital cost that can be overcome by operation and maintenance savings when compared to steel naval vessel construction. Fuel costs are a large component of the lifetime operation expense and the costs are escalating at such a high rate that future prediction is difficult, and often underestimates the actual increases. Again the LASS project examined this part of the discussion and offers good insight. Central to the LASS study was comparison of three designs for a 128m high speed ferry operating at 42kts (Hellbratt, 2011). The size and speed of this vessel is appropriate for a discussion of naval vessels because the navies of the world have been looking to increase their littoral combat capability where smaller size and high speed are very advantageous (Hellbratt, 2011).

Two of the designs in the project were aluminium and FRP demonstrating a 50% reduction in structural weight when compared to the third steel design. The steel ship was the baseline for the study and showed a much higher operation costs due to the fact that the structural weight, with the inclusion of heavier machinery to propel the greater weight, changed the hydrodynamic properties of the vessel thus increasing the power demand. Because the weight savings were about the same for the aluminium and the FRP vessels, and both are similarly resistant to corrosion, the change to lightweight material in general for this study resulted in a 19 – 22% reduction in total lifecycle costs. One point to note is that the 25 year life assumption of the study was set by the expected life of the aluminium vessel and both the steel and FRP vessels would have additional value at the end of service that was not accounted for in the data above. Furthermore, the weight savings achieved with the LRFD limit state approach used in the beam example above implies that the weight savings of the FRP hull should have been higher than the aluminium hull which implies that even greater cost savings are possible.

2.5 Hull Monitoring

By definition hull monitoring systems are all systems which include stress measurement on board ship.

Different types are possible depending on the objectives. Three main objectives can be pointed out.

- The first is to acquire data for researchers (including classification society). Those data are necessary to validate the numerical or experimental tools used for the conception of a naval ship.
- The second is to use real time measurement to give information and eventually warning to the crew. Operator guidance systems are more and more numerous on ships but not so much related to the stress measurements. Operational limits are not numerous in the field of stress reductions (in some cases, speed restrictions in heavy weather in some incidences, but no more) and most of the time based on visual observation. More complex operator guidance based on measurements is being developed and validated in a few Navies.
- The third objective can be to obtain feedback about the navigation for the maintenance services and/or headquarters. This last possibility could be related to IMO recommendations for voyage data recorder (VDR). Usually in this last case only statistical data is stored for a long period with only 24-hour time series.

Submarines are instrumented for a long time (basic loads are most often considered as pressure variation due to immersion which are simpler to measure and to estimate than wave loads) and in some fleets systematically, with this kind of data recorder, but feedback is rare due to confidentiality reason.

Not so many naval ships are instrumented (not more than 5 %) but this number is likely to increase in the future. All types of naval ships are instrumented: Frigates, during experimentation phases of the monitoring systems, complex (from the point of view of the structure arrangement); also ships such as amphibious ship and high speed craft (in particular with composite structure) for which research has more funds.

Measurements systems in the naval structure domain are mainly strain gauge (classical or optical to avoid electromagnetic interference usually encountered in military environments). More usual measurements are accelerations but they need more post-processing to obtain usable data. Additional measurements useful for understanding the behaviour of the ship are navigation sensors (GPS, speed log, rudder compass, ...) which can be collected in most data acquisition systems, also a device to estimate the sea states (different methodologies are now available).

Depending on the objectives of the system of a particular ship the data recorded can be very simple (i.e. storage of rainflow matrix about one detail); or very complex such as full real time measurement (including high frequency range in order to observe impact) with real time presentation of results on the bridge.

Hull monitoring systems are never required, but nearly all classification societies have additional class notations to cover their use, because those systems are always fitted with a view to increasing the safety of the vessel, which of course is one of the main objectives of the classification society.

2.6 Conclusion

Lightweight materials do have great potential to save cost and improve performance for naval vessels. Some materials will come with restrictions that limit their application or have their weight savings reduced by additional concerns; however, optimization may be achieved in a logical and conservative manner. The cost savings demonstrated by the LASS project show a substantial benefit in fuel savings for a medium sized, high speed vessel that would be comparable to many naval ships. Furthermore, weight

savings could be used to carry more fuel, cargo, or weaponry to enhance mission capability or used to reduce power (fuel) demand. Also, the inherent corrosion protection of aluminium, titanium, and FRP can help reduce maintenance costs and operational time lost to repair. Lastly, FRP construction is known to restrict thermal and acoustic radiation and offers very flat surfaces which makes the vessel less “visible” to sensors: thermal, acoustic, and RADAR; resulting in appreciable stealth benefits.

3 SUBMARINE PRESSURE HULL STRUCTURAL DESIGN

3.1 Introduction

Pressure hulls are the main load bearing structures of naval submarines, commercial and research submersibles, and autonomous underwater vehicles (AUVs) whose primary load-bearing responsibility is to withstand hydrostatic pressure associated with diving. The most efficient pressure hull geometries are circular thin-walled cross-sections that transfer the normal pressure load to in-plane compressive forces. Thus, pressure hulls are typically composed of a combination of ring-stiffened cylinders and cones, with spherical or torispherical domes at either end. The ring-stiffeners prevent elastic buckling from occurring before yielding of the material, further increasing structural efficiency. Load bearing “watertight” bulkheads divide longer pressure hulls into more-or-less isolated compartments. Figure 2 is a schematic of typical pressure hull structure.

Pressure hulls are subject to several different load types such as those from weapons (underwater explosions), wave slap on superstructure and other sea loads, and the predominant hydrostatic pressure, which is the focus of this article. Other submarine structures, such as the hydrodynamic casing or outer hull, the control surfaces, the bridge fin and conning tower, and many decks, tanks and minor bulkheads, play an important role in submarine diving, manoeuvring, surfacing and sea-keeping; however, due to its paramount importance for safety, this article is primarily concerned with the structural integrity of the pressure hull itself. Dome ends are an integral part of the pressure hull but not specifically addressed in detail here. Ideally, hemispheres are the most efficient dome end, but for space and manufacturing reasons, torispheres are often used.

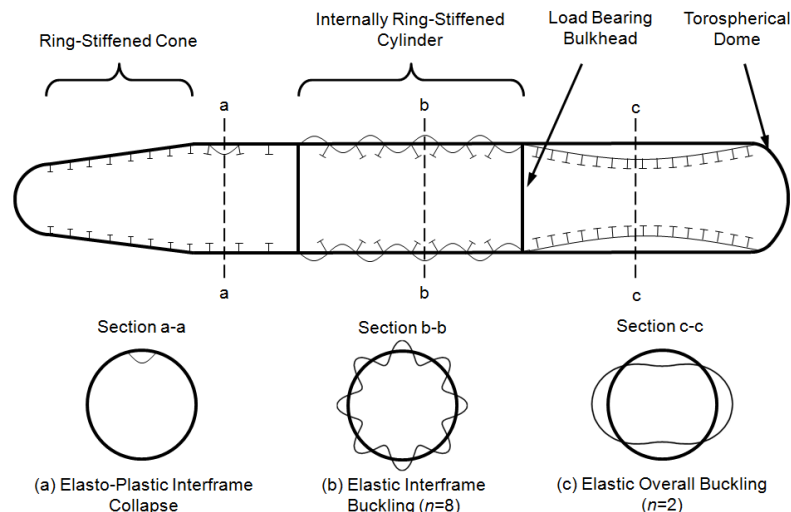


Figure 2: Typical pressure hull structure and buckling modes

Cylindrical shapes, rather than spheres, are used for submarine pressure hulls because they provide a good compromise between structural efficiency and internal space utilization. Ring stiffened cylinders are primarily designed by addressing two failure modes, interframe collapse and overall collapse. Interframe collapse is a failure of the plate between adjacent stiffeners while overall collapse is characterized by global failure of the frames and plating (Figure 2). A further source of pressure hull instability is frame tripping, which refers to torsional buckling of an inadequately proportioned ring-stiffener.

Experimental results of collapse tests are being used as part of a 50-specimen study (MacKay, 2006) to evaluate the effects of corrosion on collapse and to help develop partial safety factors for numerical models. Two of these are used in the round-robin study (Chapter 6).

3.2 Materials

Materials are not covered extensively here. Normally submarines are constructed of high yield strength steel ($> 500 \text{ MPa}$) to enable a higher elasto-plastic buckling collapse load. There has been some consideration of using composites to gain a higher yield strength to weight ratio, but manufacturing quality control is an issue for structure which fails mainly due to buckling instability, which is heavily influenced by imperfections.

3.3 Geometric Imperfections

Differential cooling after fabrication welding leads to local and global distortion of the pressure hull. Frame welding results in interframe dishing between frames, while longitudinal welding of shell sections causes global out-of-circularity (Faulkner, 1977). Of course, out-of-circularity (OOC) can also be partially attributed to the finite precision of cold rolling procedures for the shell plating and frames, as well as the accumulation of other fabrication errors. OOC imperfections are important for hull strength since they lead to destabilizing bending moments that hasten the onset of yielding and overall collapse. Pressure hulls are typically designed to accommodate a maximum radial eccentricity equal to 0.005 times the radius of the shell plating, or in the common terminology, 0.5 % OOC. Hulls are normally built to a tolerance of one-third of the design value, or approximately 0.17 % OOC (DPA, 2001).

The collapse mode of a pressure hull is influenced by the magnitude and shape of OOC: interframe collapse governs when initial imperfections are small, while overall collapse is dominant when imperfections are large and in the critical mode. On the other hand, the frame stiffness, relative to that of the shell plating, also plays a role in the mode of collapse: cylinders with closely spaced, heavy frames are more likely to fail by interframe collapse, while those with relatively weak frames will fail by overall collapse. Experimental and numerical investigations on ring-stiffened cylinders designed to fail by overall collapse have shown that 0.5 % OOC in the shape of the critical overall elastic buckling mode can result in a 15 – 25 % reduction in the elasto-plastic overall collapse pressure (Creswell and Dow, 1986; Bosman *et al.*, 1993; MacKay, 2006). When the pressure hull scantlings and OOC shape and magnitude are such that interframe and overall collapse occurs at approximately the same pressure, it is thought that failure mode interaction can significantly reduce the strength of the hull. One experiment showed that the “interactive” collapse pressure may be up to 14 % lower than either the interframe or overall collapse pressure (Creswell and Dow, 1986; Graham *et al.*, 1992).

Other types of initial geometric imperfections that may affect the strength of a pressure hull include the aforementioned interframe dishing of the hull plating between frames, misalignment of frame webs from the transverse plane (frame tilt), and deviations of the dome ends from the perfect spherical or torispherical shape.

3.4 Effect of Residual Stresses on Pressure Hull Strength

In addition to their effect on hull shape, fabrication procedures, especially cold rolling and welding, introduce locked-in, or residual, stresses to the as-built hull (Faulkner, 1977). The effect of residual stresses on interframe collapse pressure has not been extensively studied because the empirical design process for interframe collapse inherently includes fabrication effects. Nonetheless, it is generally accepted that, due to the dominance of bending and shear actions over direct compression, cold-bending of the shell plating does not significantly reduce interframe collapse strength (Kendrick, 1982). Cold rolling stresses are particularly important for overall collapse, since they can significantly modify the pressure at which frame yielding occurs. Cold rolling, combined with an overall $n = 2$ geometric imperfection, has been found to decrease overall collapse strength by up to 30 % (Faulkner, 1977; Creswell and Dow, 1986).

Residual stresses must also be considered when assessing the fatigue life of the hull. A submarine pressure hull is designed as a safe life structure from a fatigue perspective. Although an internally framed pressure hull typically only experiences compressive forces, residual welding stresses may cause the compressive load to cycle through a tensile range. Some empirical S-N curves for fatigue design of submarine hulls are presented in the UK naval submarine design standard (DPA, 2001).

3.5 Pressure Hull Design Methodology

There is a well-known discrepancy between shell buckling loads based on classical shell theory and observed experimental results. The disagreement between theory and reality has been attributed to several factors, including the general sensitivity of shell buckling to boundary conditions, load eccentricities, and geometric imperfections, as well as material related factors, such as anisotropies and residual stresses (Teng, 1996; Schmidt, 2000). Conventional shell design procedures, including interframe collapse predictions for pressure hulls, deal with analytical-experimental disparity through empirical methods. Typically, classical elastic buckling loads are plotted against the experimental values, with both buckling loads normalized using a slenderness parameter that accounts for the shell proportions and whether the shell has buckled in the plastic zone. An empirical design curve is then fit to either the mean or lower bound of the normalized experimental data, depending on the design philosophy. That type of design method is referred to as a “knock-down factor” approach, since the buckling load of the perfect structure is reduced to account for the effect of imperfections and material yielding. Hundreds of experimental results were collected for interframe collapse of pressure hulls (Kendrick, 1982), and were used to generate the empirical knock-down curves that are used in many design codes. The British (BSI, 1997) and European (ECCS, 1988) civilian pressure vessel codes use a lower bound curve, while the UK naval submarine standard (DPA, 2001) uses a curve fit to the mean of the experimental data.

Overall collapse pressures are typically estimated using analytical equations that consider bending stresses associated with OOC in order to predict the onset of material yield in either the frame flange or in the adjacent plate. Cold rolling residual stresses may be accounted for by using a larger safety factor for structures that are not stress-relieved. That is the approach taken for the British and European civilian design

codes. With the UK naval standard, overall collapse is predicted via a nonlinear elasto-plastic analysis of a single ring-frame (Kendrick, 1982). The analysis is carried out through a finite difference discretization of the ring in the circumferential direction, whereby material plasticity is tracked through several layers of the cross section. The governing equations are solved incrementally in order to predict the ultimate collapse pressure. Various correction factors are applied to the model in order to account for, for example, the finite length of a submarine compartment and interactive failure modes.

Kendrick (1982) presented an overview of externally loaded pressure vessel design criteria based on the BS5500 design code (BSI, 1997). The design methodology outlined by Kendrick (or a slightly modified version) was used in many contemporary codes e.g. ECCS (1988) and are still standard practice today e.g. DPA (2001). The BS5500 approach to design of pressure hulls is to proportion the structure such that: 1) interframe collapse is the critical failure mode, and 2) it is over-designed for overall collapse, which is difficult and computationally costly to predict accurately. Kendrick noted that the structural cost of avoiding failure by overall collapse is relatively small, and it is more economical to focus on predicting, and minimizing structural costs associated with interframe failure of the shell. The implementation of a more realistic overall elasto-plastic collapse model has allowed the UK naval submarine standard DPA (2001) to place roughly equal weight on interframe and overall collapse. This presents its own problems, as pressure hulls having similar predicted interframe and overall collapse pressures may have real interactive collapse pressures that are significantly less than either of the calculated values, as described above.

Collapse pressure predictions are related to the allowable working pressure, and deep diving depth, of a pressure hull through deterministic safety factors that were developed through a combination of experiments and past experience with pressure hull design. Some design codes use a single safety factor to account for all uncertainties (BSI, 1997; ECCS, 1988), while other codes use a partial safety factor (PSF) approach (DPA, 2001). Typical PSFs account for uncertainties associated with the predictive model (e.g. experimental scatter in the interframe design curve), deviations of the as-built hull from the design drawing, and loading.

3.6 Application of Numerical Methods to Pressure Hull Structural Design and Analysis

The conventional pressure hull design process described above is characterized by a conservativeness which has its roots in the necessity to analyse the simplest and most pessimistic geometry, which is, in turn, required due to the complexity of shell stability theory and the reliance on empirical design methods. The implementation of numerical methods, i.e. nonlinear finite element analysis (FEA), in pressure hull design procedures would address some of the inherent conservatism and inflexibility of the traditional methods by allowing strength calculations to be based on the elasto-plastic collapse limit state, rather than first yield criteria, of a complex pressure hull structure, including realistic modelling of geometric imperfections, the effects of fabrication procedures and in-service damage (e.g. due to collision or corrosion). Furthermore, numerical methods would allow the pressure hull to be designed as a whole, rather than by component, with inherent modelling of the interaction between structural components (e.g. ring-stiffened cylinders, domes and bulkheads) and modes of failure (e.g. interframe and overall collapse).

Numerical methods have traditionally been a complementary rather than an integral

aspect of pressure hull design and analysis. The BOSOR series of finite difference codes for axisymmetric shell structures (Bushnell, 1975) have been widely used to determine buckling loads and stresses in pressure hulls, e.g. Kendrick (1982), Moradi and Parsons (1993). Nonlinear FEA is currently used in pressure hull design and analysis in indirect ways, such as granting tolerance concessions to in-service structures, “validating” empirical design methods, identifying failure modes and weak structural features, determining the effects of in-service damage, and for general research purposes e.g. Creswell and Dow (1986), Graham *et al.* (1992), Morandi *et al.* (1998), Keron *et al.* (1997), Lennon and Das (1997), MacKay *et al.* (2006), Radha and Rajagopalan (2006). Despite its widespread informal use and accepted benefits, the direct use of nonlinear FEA in the design of pressure hulls is not currently supported by design codes, primarily because the accuracy of the method, which is required in order develop a partial safety factor, has not been quantified.

The numerical methods required to predict elasto-plastic collapse of submarine pressure hulls are well-established and readily available in commercial software packages. MacKay *et al.* (2011) conducted a survey of numerical models used for pressure hull analysis. Those authors found that a typical numerical model was based on a shell finite element discretization and a quasi-static incremental nonlinear analysis using Newton-Raphson iteration schemes with arc length solution methods. It was common to include geometric imperfections by either assuming a worst-case shape and amplitude, or by mapping OOC measurements on to the FE model when the analysis was aimed at predicting the response of a real structure or test specimen. Numerical material models accounted for plasticity, but residual stresses were sometimes neglected. In cases where some effort was applied to addressing residual stress effects, the methods used varied from explicit simulation of the fabrication procedures that lead to residual stresses, to the use of “effective” stress-strain curves to account for early yielding brought on by residual stresses.

Graham (2008) and MacKay *et al.* (2011) used the numerical methods described above to estimate the accuracy of FE collapse predictions. Graham modelled the collapse of several legacy test specimens that were used in the development of the UK naval submarine design standard (DPA, 2001). The test cylinders were constructed from cold rolled and welded steel so that they incorporated many of the imperfections associated with real submarine hulls. Graham simulated cold rolling procedures before performing collapse analyses, but welding residual stresses were not modelled. His analyses of thirteen test specimens gave collapse pressures within $\pm 6\%$ of the experimental values. Graham (2008) later extended his FE analysis to a fourteenth test specimen, over-predicting the collapse pressure by 8.5 %.

MacKay *et al.* (2011) used nonlinear FEA to predict the collapse pressures of twenty-two small-scale ring-stiffened cylinders. The test specimens were machined from aluminium tubing, so that residual stress levels were negligible and were neglected in the analyses. A statistical analysis of the experimental-numerical collapse pressure comparisons showed that the FE models were accurate to within 11 % with 95 % confidence. By way of comparison, the mean interframe design curve in the UK naval submarine standard (DPA, 2001) is accurate to within 20 % with 95 % confidence (MacKay *et al.*, 2011). They also found that neither the choice of FE solver, nor small differences in how the modelling was performed (e.g. in the mapping of measured OOC to the FE models), were found to significantly affect accuracy.

Experimental-numerical comparisons like those described above can be used to develop a partial safety factor that can be applied to FE collapse predictions in a design setting.

Graham (2008) suggested using the maximum discrepancy between FE predictions and experiments to directly determine a PSF. In the case of his analyses, the FE models overpredicted the experimental collapse pressures by at most 8.5 %, leading to his suggested PSF of 1.085. MacKay *et al.* (2011) proposed using a simple statistical analysis of experimental-numerical comparisons to develop the PSF. The actual value of the safety factor would depend of the degree of statistical confidence that is deemed necessary to ensure an adequate safety margin. For example, when MacKay *et al.* (2011) applied their statistical model to Graham's (2008) results, using a high level of confidence (99.5 %), the resulting PSF was 1.17. That means that, using Graham's numerical procedure, we can be 99.5 % confident that a future collapse prediction will not over-predict the actual collapse pressure by more than 17 %. The same procedure showed that Graham's "lower bound" PSF of 1.085 gives only a 90 % level of confidence (MacKay *et al.*, 2011).

A design procedure incorporating FE collapse predictions must be based on the same numerical methods that were used to generate the PSF, regardless of whether the PSF is based on a lower bound or a statistical approach. That requirement would likely result in a set of numerical modelling rules to specify the type of finite element, material model, boundary conditions, modelling of geometric imperfections, solution methods, etc., that are compatible with the PSF. It may even be appropriate to specify the actual computer programs used to generate and solve the FE model. That is the position taken by proponents of Verification and Validation (V&V) theory for numerical models (Thacker *et al.*, 2004; ASME, 2006). V&V is a developing field aimed at standardizing procedures used to ensure that numerical models are sufficiently accurate for their intended purposes.

As we have seen, much progress has been made with respect to standardizing numerical models for pressure hull collapse predictions, and furthermore, a significant amount of experimental-numerical data have been generated in support of quantifying the accuracy of the FE models. The most pressing needs, if FE methods are to be incorporated in hull design, are consensus regarding the best way to incorporate residual stresses in the analysis, further expansion of the experimental-numerical database in order to improve overall confidence in the FE results, and a set of rules defining the shape and magnitude of geometric imperfections for design.

As a final note, it is not expected that numerical methods will completely replace conventional pressure hull design curves and equations. The traditional analytical-empirical methods will likely be retained because of their simplicity and efficiency of use, as well as their value for use in iterative design procedures such as optimization routines and reliability analysis e.g. Radha and Rajagopalan (2006), Morandi *et al.* (1998). Numerical modelling is more likely to complement than to replace the conventional methods, as in a hierarchical design procedure, whereby analytical-empirical methods are used to conduct parametric studies of design variables, and to determine the nominal dimensions of the structure. Nonlinear FEA is then used to determine the design strength, either in a deterministic or probabilistic (i.e. reliability) setting.

4 MILITARY LOADS

Structural design for military loads was deeply described in chapter 6 of ISSC (2006) committee V.5 for Naval Ship Design. This description, which is still valid for nowadays understanding of this subject, included a review of every kind of load to be taken into account in any naval design: weapon effects (above and under water), fragments and penetrations as well as structural aspects of residual strength.

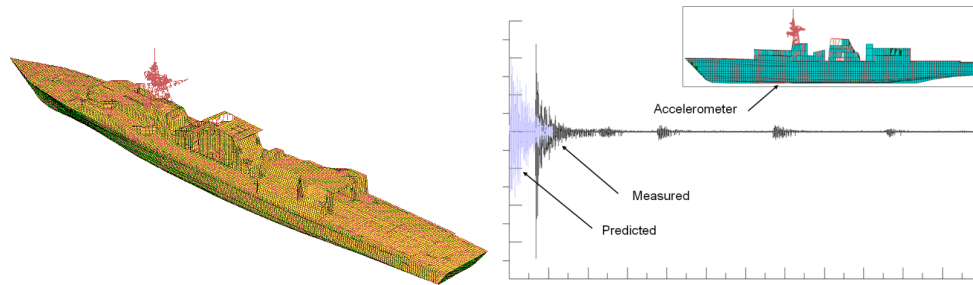


Figure 3: Full ship FE model for the USA prediction correlated with FSST measurement.

Now in ISSC 2012, this chapter for military loads tries to give a brief review about recent developments presented in public domains. It needs to be mentioned that most of substantial information about weapons effects and military loads remains classified within navies and is not available in the public domain.

4.1 Under Water Weapons Effects

Recent developments in underwater weapon effects are mainly focused in approaches to substitute the explosive loading is full scale shock trials (FSST).

One of these approaches is the FSST simulation by means of complex codes. The effort of the community is basically to correlate the results of the simulations with those obtained from FSST.

Regarding codes for underwater explosion analysis,, the commercially available program USA (Underwater Shock Analysis) applies the Doubly Asymptotic Approximation method DAA developed by Geers (1978). It is used by organizations in a number of countries. Although originally interfaced with the finite element system STAGS, it now is interfaced with a number of other finite element systems, including NAS-TRAN, ANSYS, ABAQUS, LSDYNA and TRIDENT. USA has been applied to full ship global finite element models like the one shown in Figures 3 and 4, and verified and validated both theoretically and experimentally. Figure 3 shows USA predicted versus experimentally determined acceleration time histories at forward keel location.

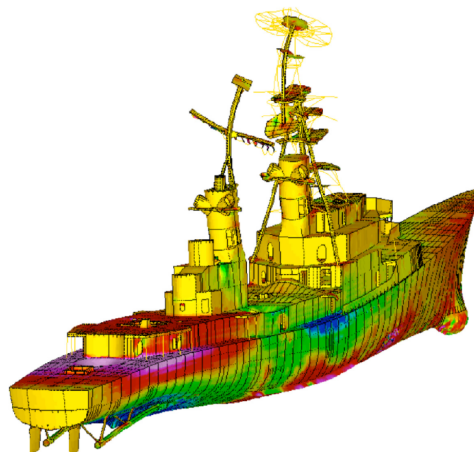


Figure 4: Structural model of the destroyer Lütjens. Deformation under lateral blast.

DYSMAS is another system that is being used in conjunction with structural finite element analysis procedures. This system was created by the Naval Surface Warfare Center (NSWC) and its German Ministry of Defense partners (IABG) by uniting the NSWC GEMINI solver with a modified version of the Laurence Livermore National Laboratory (LLNL) structural dynamic code DYNA3D. A major effort is now in place to increase the computational efficiency through parallel processing, Ferencz (2008). Figure 4 shows the structural domain for an existing DYSMAS production simulation model.

Another approach in recent years to substitute the full scale shock testing of warships is the use of Air Guns. Weidlinger Associates company has successfully developed and patented (Patent N° US 6,662,624 B1, dated Dec. 16, 2003) this alternative shock test methodology that avoid the use of explosives.

The method consists on an array of air guns (air reservoirs) which are positioned close to the vessel hull and generates a high pressure shock pulse over the length of the array arrangement. The way the high pressurised air is released from the air guns can be controlled by software to generate the desired shock effects on the ship structure and systems.

This is considered an environmentally friendly method since the energy released by the air guns is directly focussed on the ship, instead of explosives which energy is radiated spherically to the ocean. Thus, air guns shock testing reduces the risk for damages to personnel and to sea environment.

Another advantage is the cost savings since air guns testing can be performed on the naval base harbour with commercial equipments commonly used in oil prospection.

This methodology has been already used for shock testing of UK decommissioned Type 42 Destroyer as well as for Canadian decommissioned submarine. Results of these experimental activities basically look for benchmarking and proper correlation with explosive testing. References for every aspect described here, were presented mainly in SAVIAC (2008 and 2009) restricted publications.

4.2 Asymmetric Threats

In accordance with NATO AAP-6(2008), an asymmetric threat is defined as a threat emanating from the potential use of dissimilar means or methods to circumvent or negate an opponent's strengths while exploiting his weaknesses to obtain a disproportionate result.

In the case of naval ships, the asymmetric threat becomes highly critical when the ship is in a dangerous foreign port. Terrorist attacks, by means of small/medium caliber projectiles or explosive charges carried by any kind of vehicles or suicides, are the threats that a naval ship shall be prepared to resist.

Countermeasure to avoid structural damage is basically to improve the ballistic protection of critical areas in both aspects 1) extension of exposed area to be protected and 2) protection level in terms of the intensity of the expected maximum impact.

Some general ballistic protection structural aspects and techniques were presented within chapter 6 of ISSC (2006) committee V.5.

5 RESIDUAL STRENGTH AFTER DAMAGE

As design and analysis tools facilitate structures optimized to anticipated loads, it has become ever more important that designers include considerations related to ultimate strength and residual strength after damage. This is true for commercial ships

and marine structures which will encounter accidental or incidental extreme loading events but is especially relevant for naval ships which are intended to put themselves into scenarios which include targeted aggression and expectations for operations after damage.

Committee V.1 produced a comprehensive report on residual strength after damage for commercial ships for the 17th ISSC report (2009) and has provided a follow-on report specifically addressing offshore structures for the 18th ISSC report (2012). This most recent effort on the part of Committee V.1 will include considerations for loads produced by terrorist actions and, as such, will cover items of interest for those seeking information on residual strength after damage in naval scenarios. The subject of provision of residual strength after damage in naval ships was addressed by Committee V.5 in its 16th ISSC report (2006). This chapter of the 18th ISSC Committee V.5 report is intended as an update to this last report and supplementary to the work of Committee V.1.

As residual strength after damage assessments for both commercial and naval ships hold much in common, it is worthwhile to review recent work which is relevant to both. Most Classification Societies include processes for evaluating residual strength after damage for commercial marine structures including ships and provide classification notations which document the extent to which such a consideration has been made for a specific platform. As an example, ABS has published Guides for such processes for tankers (ABS, 1995) and bulk carriers (ABS, 1995). As was described in Committee V.1s initial report, these processes include definition of the damage scenarios, establishment of the operation goals after damage and assessment of the vessels ability to meet those goals. Vhanmane and Bhattacharya (2011) assess the extension of the classification processes as represented in the International Association of Classification Societies Common Structural Rules approach to ultimate strength. Their conclusion is that the approach presented by the CSR is adequate to address such considerations in the early design phases and can be followed up by more specific evaluations after the design is mature. Such evaluations are covered extensively in the Committee V.1 17th ISSC report (2009). More recently the Royal Institute of Naval Architects sponsored a conference on The Damaged Ship (2011) in London. Although much of the work addressed stability after damage, a number of relevant structural papers were presented. Amongst these, most papers were intended to provide either practical or analytical approaches to evaluation of strength after damage. Quinn and Hills (2011) provide an overall review of the MOD(UK)s organization structure for addressing incidents while Wang (2011) provides similar insight into the organizational structure of a classification societies parallel approach to rapid response damage assessment. Sahid (2011), Kwon *et al.* (2011) and Martin (2011) provide proposed analytical approaches to strength after damage assessment while Mangriotis (2011) provides a similar process for refloating grounded ships but including considerations from on-site survey. Fone *et al.* (2011) provide experimental loading data which can support analysis. Ellam *et al.* (2011) and Harman *et al.* (2011) describe anecdotal applications of assessment processes in the case of actual incidents. Finally Marshall (2011) provides an interesting proposal targeted at integrating strength after damage requirements from the new Naval Safety Code with military operational considerations tailored to the projected employment of a specific naval platform.

In general, addressing residual strength after damage for a naval platform begins with a structural vulnerability analysis at a point where the design is relatively mature and has been developed to handle normal expected environmental and peacetime op-

erational loads. It starts with a threat assessment and the potential resulting failure effects under various operating scenarios. This is usually restricted information as it is very sensitive in nature and may reveal weaknesses to a potential adversary. The structural analysis will usually include the affects of underwater explosions and air explosions at an estimated distance from the vessel or a contact mine hitting a particular location on a vessel. The failure mode analysis may initially find that structural connections are key points of failure due to the loading and unloading and again loading as such phenomena as pressure pulse waves pass the ship in a rapid time sequence. Therefore, structural connections are often designed to handle the maximum ultimate stress anticipated at the beam end connection. Additionally, the hull girder strength assessment will include the maximum transverse and vertical whipping moment from the weapons effects as well as the wave bending moment component. The vulnerability analysis will also take into consideration the spacing of the transverse and longitudinal watertight bulkheads with special consideration for two or three damaged compartment scenarios. Particular attention must be given to the relationship of the separation of engine room bulkheads and the ability of the vessel to sustain damage and continue to operate. In performing the structural vulnerability analysis, the ultimate strength analysis will clearly show the sequence of failures as they relate to buckling and yielding of the individual structural elements and potential effects on critical systems providing electrical power, fire fighting systems and ballast/de-watering system key to successful damage control.

Insofar as analyses methods are concerned, current approaches have been adequately addressed in the prior reports of ISSC Committee V.1. However, as with any problem involving a large number of uncertainties, analyses methods have had to incorporate a large number of assumptions, the effects of which have been unknown. These include extent of damages, the resulting structural geometry, the nature of the post-incident loadings, and post deformation material properties. Efforts have been focused on ensuring these results are conservative.

Ongoing work by Underwood (2011) is attempting to advance current methods from analysis of discretized stiffened plate elements as they perform under loads after removal of damaged elements to the use response surfaces which more accurately represent the post-incident geometry and modified material properties. Initial results seem to indicate significant differences in prediction of ultimate failure. Concurrently Benson *et al.* (2011, 2010, 2009) have examined application of ultimate strength analyses methods to lightweight aluminium naval vessels and have been able to develop a compartment based simplified progressive collapse analysis method for such structures, this methodology incorporates overall compartment level collapse modes as well as interframe collapse modes. It is worthwhile to note that an international Damaged Ship Structural Workshop was held in 2011 at the Naval Surface Warfare Center Carderock, MD USA in 2011 but the work presented there has yet to be published.

Insofar as incorporation of residual strength considerations into existing naval codes are concerned, several classification societies produced naval rules which include requirements for such assessment.

Lloyds Register Approach to Residual Strength Assessment (RSA) new requirements for residual strength introduced in Volume 1, Part 4, Chapter 2, section 7 of the Lloyds Register Naval Ship Rules (2011) are intended to verify that the residual strength assessment is adequate to ensure the ship will structurally survive in the event of an incident that impacts the hull girder. The ultimate strength of the hull in the damaged condition is determined using elasto-plastic methods and the damaged ultimate

strength is compared with the still water plus wave bending moment to ensure a small safety margin exists. In the LR Naval Rules, direct calculation techniques using short term values are required to predict extreme wave bending moments for a range of sea states. For each sea state, the assumption is that the mean period of the sea state is close to the peak of the ships wave bending moment response and hence maximise the bending moment response. From this information, it is possible to derive a residual strength wave bending moment relationship which is proportional to wave height and ship length. Naval ships that comply as defined in Volume 1, Part 4, Chapter 2, section 7 of the LR Naval Ship Rules will be assigned a RSA notation.

Germanischer Lloyd's Approach to Residual Strength after Damage, GL Naval Rules (2011) address Residual Strength in Section 21 of Hull Structures and Ship Equipment (III-1-1). The character and extent of each investigated case, as well as the assumed environmental conditions are defined by the Navy. Buckling and yield capacities of undamaged components are analysed and if the strength capacity of the intact hull components for remaining tasks defined by the Navy is sufficient, the class notation RSM is assigned. Minimum requirements are defined for plane plate fields, curved plate fields, stiffeners and girders (buckling), secondary stiffeners and primary members acting as columns. Proof of overall strength is done by applying the bending moments and shear forces to the cross section consisting of the components which are still intact. If more than one member/column is forming the residual hull cross section, effects of second order are considered. The conditions which have to be satisfied for non-linear calculations (ultimate load/ultimate strength) are defined in the rules. Materials of elements which are relevant for residual strength are not to be of lower class than Material Class III.

RINAs Approach to Residual Strength After Damage, RINA rules for Naval Ships (2011) deal with Military Notations in Part F, Chapter 1 – Additional Class Notations. Section 1 illustrates a specific confidential notation - STRU-DAM -, which can be assigned to ships in order to certify that measures are taken to increase their residual strength after damage to hull structures from an assigned explosion. This implies that structural analyses are carried out and that the ultimate strength of the damaged hull complies with specified requirements. Confidential input data include explosion location, mass and type of the charge, the equivalent TNT weight. The specific analysis method is left to the designer but must be approved by RINA.

DNVs Approach to Residual Strength after Damage, Residual strength after damage in DNV naval rules (2011) is handled by the DNV class notation CBT-H (Combat Survivability – Hull). The class notation covers hull girder strength in a given sea state after a hull damage. Calculations are done by the designer and verified by DNV. Damage size is provided by the Navy based on their internal (classified) evaluation of threat, weapon type and possible damage size. DNV gives default values for damage radius if the Navy does not want to specify damage size. The damage is to be considered anywhere at the ships cross section above waterline. The residual strength evaluation shall as a minimum cover midship section and quarter length forward and aft. Flooding related to the damage is to be considered when calculating the hull girder bending moment. Ship structure within the damage area is to be considered damaged to the next main structural element. The strength of the hull girder with the removed structure is calculated using FEM analysis considering yield and buckling. More sophisticated ultimate capacity models may be used on a case by case basis. The loads are calculated based on a direct calculated simulation for a low speed and a specified sea state. These parameters are normally to be defined by the Navy, but

default values may be found in the DNV Rules. The acceptance criterion is fulfilled when the damage strength (material yield or buckling) is higher than the static and dynamic loads in the given sea state. No safety factors are used in the calculation. If the Navy requires a more detailed result, the ultimate hull girder capacity may be used to determine the capacity with possible additional limits on permanent deformation. The design parameters for the analysis are agreed with the customer before the work is done.

6 BENCHMARK STUDIES

Two round-robin benchmark studies, relevant to naval platforms were undertaken during this committee's mandate. These consisted of numerical simulations for comparison of the effects of various solution parameters. Experimental data were also available for comparison in both cases. The two studies were; the simulation of the response of a flat square plate to an air blast load, and the simulation of collapse of a ring-stiffened cylinder under hydrostatic pressure. The latter included an intact cylinder and one with corrosion damage. These two problems are important topics for naval vessel structural design and are also complex, nonlinear failure calculations. As such, while the parametric comparisons are not comprehensive, the presentation of these two studies should be instructive for those seeking guidance on performing these types of calculations.

6.1 Square Plate Subject to a Blast Load

This round robin test compared the prediction results for a uniform air-blast load against a square plate. Experimental results for the center permanent set exist for the comparison (Houlston *et al.*, 1985). The plate is shown in Figure 5 with dimensions of $508 \times 508 \times 3.4 \text{ mm}$ which would be typical of side shell construction in a naval vessel. The boundary conditions were nominally clamped by bolting but as can be seen in the figure of the deformed plate, there was some slippage so conditions were not ideally clamped. The material was steel with $E = 207000 \text{ MPa}$, $yield = 350 \text{ MPa}$ and $Et = 20875 \text{ MPa}$ (strain hardening modulus).

The response of blast loaded plates in air and in water is described by Rajendran and Lee (2009). They give a complete review of the four important aspects of the blast damage phenomenon. (1) The detonation process or rapid chemical reaction of

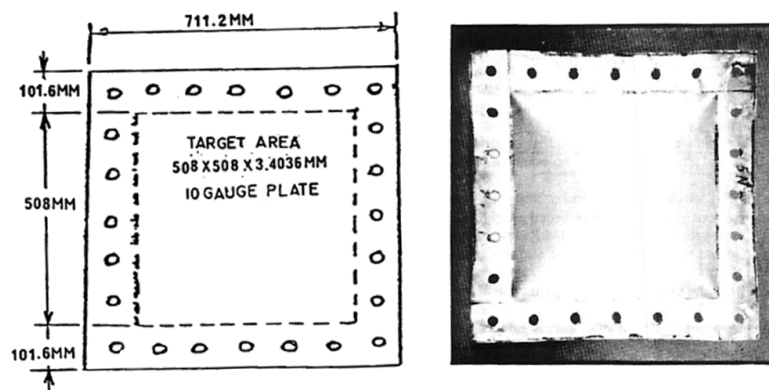


Figure 5: Square plate dimensions and final deformed shape

the explosive, (2) the shock wave propagation in the medium in which the detonation takes place, (3) the interaction of the shock wave with the plate and (4) the response of the plate to the input shock loading.

For the pressure-time characteristic and impulse of the shock wave in air they make reference to the Friedlander equation. For fully clamped rectangular plates without strain rate effects reference is made to the analytical method of Jones (1989) for the deflection-thickness ratio as given in equation 3.

$$\left(\frac{\delta}{t}\right)_r = \frac{(3 - \xi_0) \left\{ (1 - \Gamma)^{1/2} - 1 \right\}}{2 \{ 1 + (\xi_0 - 1)(\xi_0 - 2) \}} \quad (3)$$

where

$$\Gamma = \frac{2\rho_p V^2 a^2 \beta^2}{3\sigma_y t^2} (3 - 2\xi_0) \left(1 - \xi_0 + \frac{1}{2 - \xi_0} \right) \quad (4)$$

$$\xi_0 = \beta \left\{ (3 + \beta^2)^{1/2} - \beta \right\} \quad (5)$$

$$\beta = \frac{b}{a} \quad (6)$$

b and a are half the breadth and length of our square plate. Applying this equation our experiment gives a maximal mid point deflection of 29.8 mm . This is in good correspondence with the numerical results for the clamped plate.

The air blast load was assumed to act uniformly over the plate with a measured load history given in Figure 6. The simulations were done by finite element analysis with parameters varied as indicated in the results shown in Table 8. The experimental result shown at the bottom of the table indicates a permanent central deflection of 37.0 mm . Matching this value by the numerical comparisons is somewhat difficult due to the uncertainty of the experimental clamped boundary condition. Also of note is the equivalent linear static result using the peak pressure as a static load. The effects of dynamic behaviour and nonlinear material and geometry are very significant for this problem.

An example of the displacement time histories is given in Figure 7, indicating only small differences in results for material nonlinear representation or mesh size.

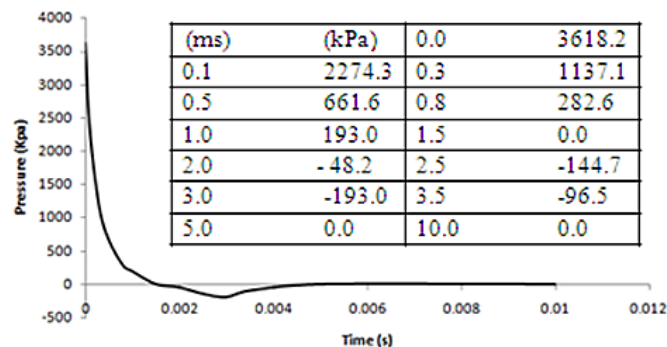


Figure 6: Load time history

Table 8: Results of Round-Robin test for blast load on a square plate

B.C.s	Mesh	Code	Solution	Nonlinearity		Nat'l	Central	Max Stress	
			Method	Material	Geometry	Frequency	Deflection	Principal	V Mises
	(mm)					(Hz)	(mm)	(Mpa)	(Mpa)
CL*		Experiment					37.0		
CL	25.4×25.4	TRIDENT	linear				113.0		
SS	10×10	ABAQUS	implicit	x-sh	×	64.2	36.2	355.7	342.2
SS	5×5	ABAQUS	explicit	x-pp	×	64.2	36.0	272.6	236.6
SS	12.7×12.7	LS-DYNA	explicit	x-sh	×	62.7	36.0		
SS	12.7×12.7	LS-DYNA	implicit	x-sh	×	62.7			
SS	5×5	ABAQUS	explicit	x-sh	×	64.2	36.0	292.5	261.9
SS	5×5	ABAQUS	implicit	x-pp	×	64.2	36.0	406.8	352.4
SS	5×5	ABAQUS	implicit	x-sh	×	64.2	36.0	406.8	352.4
SS	20×20	ABAQUS	implicit	x-pp	×	64.2	35.8	226.1	210.7
SS	10×10	ABAQUS	explicit	x-pp	×	64.2	35.7	292.3	262.2
SS	10×10	ABAQUS	explicit	x-sh	×	64.2	35.6	295.8	264.9
SS	20×20	ABAQUS	implicit	x-sh	×	64.2	35.4	251.6	319.8
SS	10×10	ABAQUS	implicit	x-pp	×	64.2	35.3	211.6	350.0
SS	20×20	ABAQUS	explicit	x-pp	×	64.2	35.0	396.0	350.0
SS	20×20	ABAQUS	explicit	x-sh	×	64.2	35.0	397.4	350.0
CL	5×5	ABAQUS	implicit	x-sh	×	118.8	34.3	401.8	352.0
CL	10×10	ABAQUS	implicit	x-sh	×	118.5	34.1	401.5	351.2
CL	5×5	ABAQUS	explicit	x-pp	×	118.8	34.1	396.8	350.0
CL	5×5	ABAQUS	explicit	x-sh	×	118.8	34.0	399.2	352.5
CL	10×10	ABAQUS	implicit	x-pp	×	118.5	33.8	384.9	335.7
CL	10×10	ABAQUS	explicit	x-pp	×	118.5	33.7	329.8	333.0
CL	10×10	ABAQUS	explicit	x-sh	×	118.5	33.7	346.2	341.1
CL	5×5	ABAQUS	implicit	x-pp	×	118.8	33.4	227.3	350.0
CL	20×20	ABAQUS	implicit	x-sh	×	118.5	33.3	339.1	351.2
CL	20×20	ABAQUS	implicit	x-pp	×	118.5	33.0	307.0	380.0
CL	12.7×12.7	Dytran	explicit	x-sh	×	117.0	33.0		
CL	12.7×12.7	LS-DYNA	explicit	x-sh	×	115.74	33.0		
CL	20×20	ABAQUS	explicit	x-pp	×	118.5	32.8	194.0	230.9
CL	20×20	ABAQUS	explicit	x-sh	×	118.5	32.7	187.0	235.1
CL	42.3×42.3		explicit	x-sh	×	115.5	31.0		
CL	10×10	TRIDENT	implicit	×	×	117.2	30.0		
CL	5×5		implicit	×	×		30.0	478.0	
SS	12.7×12.7	ANSYS	implicit	×	×	64.3	29.0		
CL	25.4×25.4	TRIDENT	implicit	×	×	117.0	29.0	438.0	
CL	12.7×12.7	ANSYS	implicit	x-sh	×	117.4	26.0		
CL	12.7×12.7	ANSYS		x-sh	×		23.0		
	42.3×42.3			×	×		23.0		

SS – Simply Supported boundary,

CL – Clamped Boundary,

pp – perfectly plastic,

sh – strain hardening

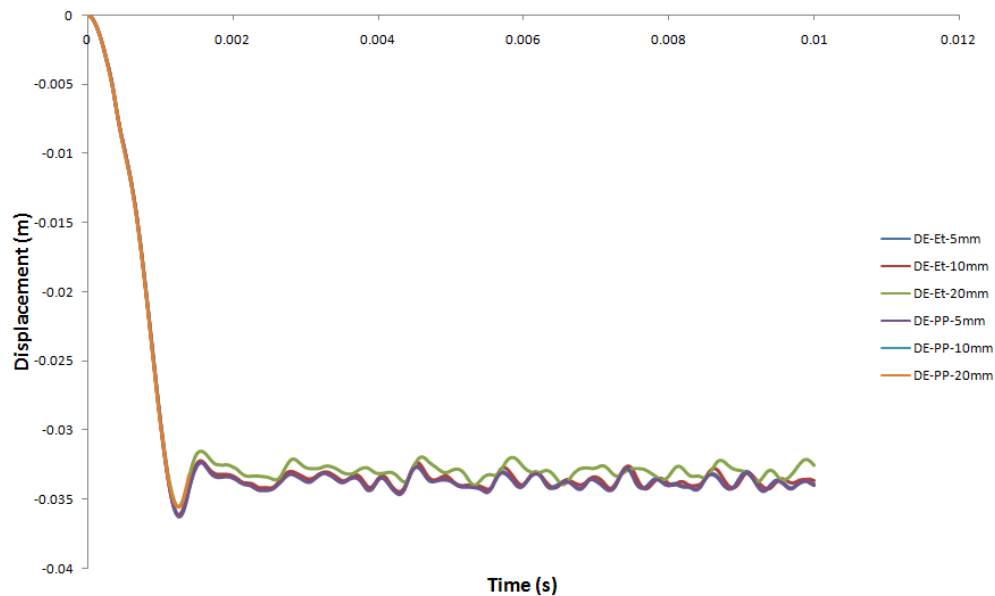


Figure 7: Results of displacement vs. time for explicit solution and strain hardening (Et) and perfectly plastic (PP) nonlinear FEA with differing mesh size

Performing a structural response analysis to a large blast load requires consideration of several factors. First of all it is a nonlinear dynamic impulse problem which requires modelling of nonlinear material and nonlinear large displacement behaviour within a time integration scheme capable of modelling short duration, rapidly changing impulse response. Most finite element programs will allow this type of analysis but the analyst must be aware of the effects of the different solution parameters and options that are available to him, to produce reliable results.

The time integration scheme can be either implicit (equilibrium performed at the current time step) or explicit (equilibrium is carried forward from the previous time step). Implicit requires more computations per time step than explicit but remains stable with larger time step sizes. Explicit generally requires smaller time step sizes to provide a stable solution, with time steps being less than 1/10th of the natural period of the structure. Another consideration in choosing a time step is that it must be small enough to accurately represent the load time history that it is modelling. For this reason, explicit solutions are often chosen for impulse problems, as the time step must be very small to accurately represent the load, and hence is usually small enough to meet the stability criteria of an explicit solution which requires less computation than an implicit solution. For this case study, there was not a great deal of difference between the two solution types and unfortunately, solution times were not reported. Solution time is less important than it used to be with modern computers.

Nonlinear material behaviour is essential for this problem as the material, particularly at the plate boundaries very quickly surpasses yield. Choices of modelling the nonlinear material region as perfectly plastic or including strain-hardening are options but did not show much difference in solutions. Nonlinear, large displacement non-linearity is very important for a supported plate problem like this as it allows the membrane effects (similar to suspension cable problems) to come into effect, which greatly increases the plates ability to withstand the load. The single linear analysis

shows significantly greater displacement response because the membrane effects are not allowed to develop.

The effects of boundary conditions are also an important consideration in this problem. As mentioned, the actual experiment did not have completely clamped response. Analyses were undertaken with both simply supported (SS) and clamped (CL) boundary conditions. SS gave somewhat better comparison to the experimental results, however, because the plate yields so quickly at the boundary, forming plastic hinges, the CL case very quickly becomes SS anyways. In general the SS analyses gave better results but differences were small.

Mesh size is always an important consideration in finite element calculations. In this case, the smaller the element size, the better the results, although differences between the mesh sizes chosen were not great. The 5 mm size is on the order of the plate thickness (3.4 mm) and in general, one does not want to have elements that are smaller in area dimensions than thickness.

The choice of finite element code had some effect, but no definite trends. In general it is important for a novice to this type of problem solution to experiment with the available parameters until he is satisfied that he has correct and converged results. Comparison to published solutions such as this one, are often a valuable resource in developing solution procedures.

6.2 Ring-Stiffened Cylinder Subject to Hydrostatic Pressure Load

This case study consisted of a round robin whereby the participants generated collapse predictions for two experimental models (Mackay and Pegg, 2010). Those models were tested under a joint project of Defence Research and Development Canada and the Netherlands Ministry of Defence that examined the effect of corrosion thinning on pressure hull strength and stability (Mackay, Smith *et al.*, In Press). The test models are small-scale aluminium ring-stiffened cylinders, their nominal dimensions are shown in Figure 8. The two models chosen for the case study are nominally identical, except for a patch of artificial corrosion on one of the specimens that was introduced by machining away some of the shell material (Figure 9).

The participants were allowed to use any method to predict the strength of the cylinders, including analytical, empirical or numerical methods, or some combination thereof. Each participant reported the predicted collapse pressure and yield pressure of each specimen, as well as predicted pressure-strain histories. The experimental results were withheld until after the participants submitted their results.

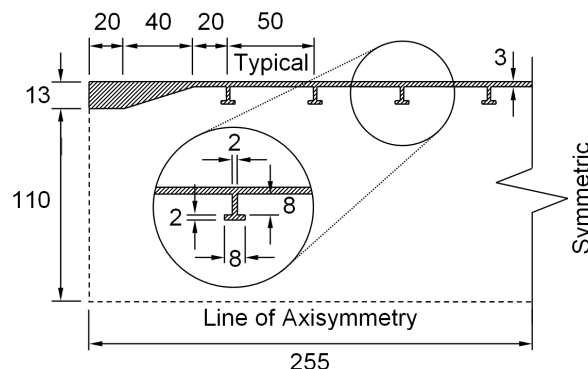


Figure 8: Nominal dimensions of test specimens



Figure 9: Photographs of the two ring-stiffened cylinder test specimens

6.2.1 Measured Specimen Geometry

A coordinate-measuring machine was used to measure the radii of the specimens at stiffener and mid-bay locations. Measurements were taken at 36 circumferential locations (10° intervals) on both the inside and outside surfaces. The specimen out-of-circularity (OOC), shell thickness at mid-bay, and combined stiffener-shell height were derived from those data.

A statistical summary of the measured radii is given in Table 9. The as-built cylinders showed good agreement with the design drawings, as indicated by the mean measured radii, none of which exceeded $\pm 0.1\%$ of the specified value. The near-perfect circularity of the machined cylinders is indicated by the coefficient of variation (i.e. standard deviation divided by the mean), which falls below 0.1% , and the maximum values of OOC, which fall well below the standard design value of 0.5% of the mean radius.

Fourier decompositions of the measured radii were performed in order to determine the contributions of the various modes (i.e. n -value, or number of circumferential waves) of imperfections. Mean Fourier amplitudes for both cylinders at ring-stiffener locations are shown in Table 10. The $n = 0$ and $n = 1$ modes represent the mean radius and the offset from the true centre of the data, respectively. Modes $n \geq 2$ describe the

Table 9: Measured radii of experimental specimens

Specimen	Radius of Stiffener Flange ^{a,d} (mm)				Outer Radius of Shell ^{b,d} (mm)			
	Nominal	Mean	St. Dev.	OOC ^c	Nominal	Mean	St. Dev.	OOC ^c
Intact	110	109.927	0.061	0.104%	123	123.010	0.030	0.078%
Corroded	110	109.948	0.073	0.155%	123	123.010	0.026	0.043%

a. Inner radius at stiffener flange.

b. Measurements taken at mid-bay and stiffener locations. Excludes radial measurements taken at corroded regions.

c. OOC is taken as the maximum absolute value of the deviation from the mean radius, expressed as a percentage of the mean radius.

d. Measured mean radii, standard deviation and OOC are the calculated using the raw measured radius less the $n = 1$ Fourier component to account for the offset of the measurement apparatus from the axis of revolution.

Table 10: Summary of Fourier decomposition at stiffener locations

Specimen Name	Mean Fourier Amplitude, A_n (mm), at Stiffener Locations ^{a,b}						
	$n = 0$	$n = 1$	$n = 2$	$n = 3$	$n = 4$	$n = 5$	$n = 6$
L510-No6A	109.925	0.026	0.077	0.019	0.004	0.003	0.001
L510-No10A	109.948	0.039	0.091	0.038	0.005	0.003	0.001

- a. Fourier amplitudes for $n > 6$ are negligible.
 b. Fourier amplitudes are based on inner shell radii at the stiffener flanges.

Table 11: Measured shell thicknesses of experimental specimens

Specimen Name	Shell Thickness in Undamaged Region ^a (mm)				Shell Thickness in Corroded Region ^b (mm)			
	Nominal	Mean	St. Dev.	COV	Nominal	Mean	St. Dev.	COV
L510-No6A	3	3.100	0.056	1.80 %	-	-	-	-
L510-No10A	3	3.082	0.086	2.79 %	2.6	2.612	0.019	0.71 %

- a. Shell thicknesses are calculated by subtracting outer and inner shell radii, using the raw measured radii less $n = 1$ Fourier components to account for the offset of the measurement apparatus from the axis of revolution.
 b. Shell thicknesses in the corroded region are calculated by subtracting the raw outer and inner shell radii. Thickness data in the corroded region are based on 10 measurement locations.

geometric imperfections. The results of the Fourier decompositions show that the machining process resulted in a dominant $n = 2$ imperfection at the stiffener flanges.

Thickness data for the specimens, derived from the measured inner and outer radii, are summarized in Table 11. The average measured values of shell thickness in the undamaged regions were within 4 % of the nominal value for all specimens. The average shell thickness in the corrosion patch of the corroded cylinder is within approximately 0.5 % of the nominal value. In general, the shell thicknesses were quite uniform, with no individual coefficient of variation (COV) significantly greater than 3 % for the undamaged shell regions.

The actual magnitude of shell thinning for the corroded cylinder, based on the average thicknesses listed in Table 11, was 15.2 %. That value is somewhat greater than the nominal value of 13.3 %, mainly due to the above-nominal thickness in the intact region of the model.

Participants were provided with the raw data measurements of all geometric quantities (Mackay and Pegg, 2010).

6.2.2 Measured Material Properties

The test models were machined from 6082-F28 aluminium alloy tubing. Tensile coupons were machined from a test cylinder. The results of coupon testing for specimens taken from the circumferential, axial and shear (45°) directions are presented

Table 12: Measured material properties determined from coupons taken from a cylinder specimen that was not pressure tested

Direction	Yield Strength, 0.2 % Offset (MPa)	Tensile Strength (MPa)	Young's Modulus (GPa)
Circumferential ^a	233	302	68.3
Axial ^b	258	328	74.3
Shear (45°) ^a	209	272	65.5

- a. Reporting the mean values based on three tensile coupon specimens.
 b. Reporting the mean values based on four tensile coupon specimens.

Table 13: Results of Round Robin

Code	Mesh	OOC	Material		Undamaged Cylinder			Corroded Cylinder		
			E	σ	Yield	Collapse	Mode	Yield	Collapse	Mode
	(mm)		(Mpa)	(Mpa)	(Mpa)	(Mpa)		(Mpa)	(Mpa)	
Experiment					6.5 ¹ , 7.2	7.3		4.8, 5.8	6	
Analytical										
SSP74		.005R	65000	240	6.6	7.1	I(9)	5	5.4	I(9)
Memphis		.001R	65500	238	7.28	7.8	I(8)			
UK MOD						4.99			3.56	
Numerical										
ABAQUS	5x5	Imp1	70000	260	7.68	7.68	I	6.75	6.75	
ABAQUS	5x5	Imp2	70000	260	6.06	6.59	I	5.97	5.97	
ABAQUS	5x5	Imp1	68300	233	6.98	6.98	I	6.08	6.13	
ANSYS	4.8x4.8	.001R	70000	233	6.5	7.2				
ANSYS	4.8x4.8	.0015R	70000	233				5.2	6.5	
ALGOR	5x5.3	none	71300	245	7.05	7.6		6.15	6.4	
ALGOR	5x5.3	meas	71300	245	6.8	7.15		5.65	6.2	
ANSYS		none?				6.8			1.6	
ANSYS	.65x2.6	meas	c. stress-strain curves			7.51	O		6.07	O

1 Experiment First Yield - Shell, Frame

Imp1 use Fourier components from measurements

Imp2 similar to Imp1 but scaled to .005R

in Table 12. These results show anisotropy in the fabricated cylinder, with the axial yield stress approximately 10 % greater than, and the shear yield stress approximately 10 % less than, the circumferential yield stress.

6.2.3 Round-robin Results

Table 13 gives the results of the round-robin tests. Participants used a variety of analytical and finite element codes, as well as variations in OOC imperfection representation and material properties.

Figure 10 shows the collapsed experimental specimen and Figure 11 shows the collapse process for the undamaged cylinder. Figure 12 shows typical nonlinear finite element collapse analysis for the model with the corrosion patch.

As was the case for the plate study, it was not possible to do as full a range of parameter variation as originally planned. Also similar to the plate problem, collapse of a ring-stiffened cylinder from external pressure is a very complex analysis where this study can provide some guidance and a benchmark for others. The cylinder collapses through buckling instability in a regime of elasto-plastic material behaviour and geometric nonlinearity. The buckling collapse also occurs suddenly, requiring care in the load-stepping procedure near collapse.

Most of the results showed reasonably good agreement with the experimental values, with some being conservative and others being unconservative, for both the undamaged and damaged models.

In modelling buckling collapse with numerical finite element analysis, it is necessary to include out-of-circularity (OOC) imperfections. The nucleation and growth of elasto-plastic instability requires some initial imperfection to begin the process. The magni-

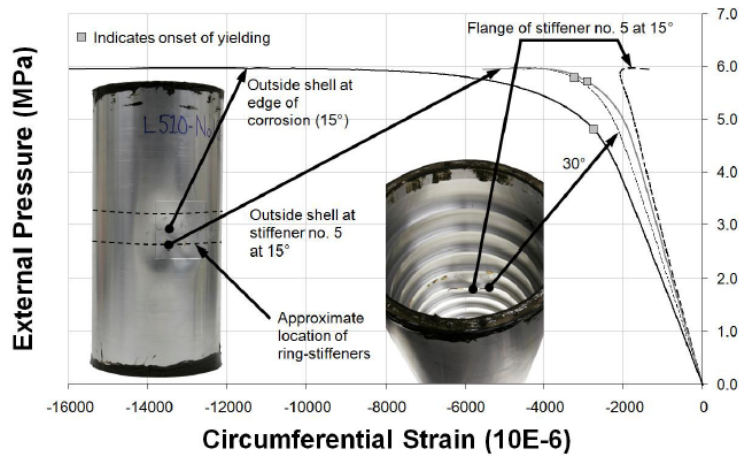


Figure 10: Experimental result of cylinder with corrosion patch

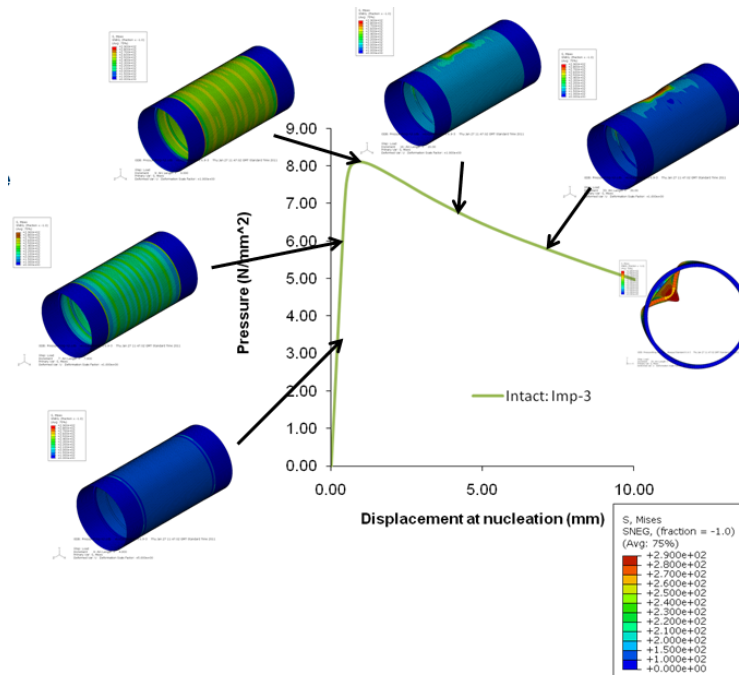


Figure 11: Elasto-Plastic collapse process for undamaged cylinder

tude and shape of the initial imperfection affect the final failure load. The larger the initial OOC, and the closer the OOC shape to the failure mode, the lower the failure pressure, in general. There are different approaches to defining OOC in analysis and design. The amplitude can either be measured from the structure if it exists, or a maximum build tolerance can be assumed (in this case $.005 \times \text{radius}$). The shape can also be measured if the structure exists, can be determined by first doing an elastic buckling analysis and using that shape to define the OOC for the subsequent elasto-plastic collapse analysis, or some statistical range of expected mode shapes can be

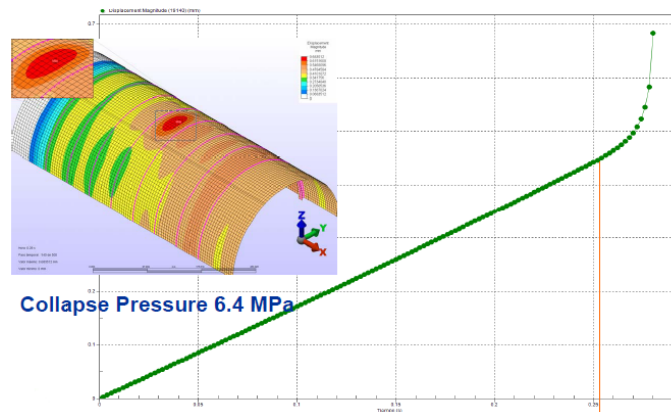


Figure 12: Results for model with corrosion patch

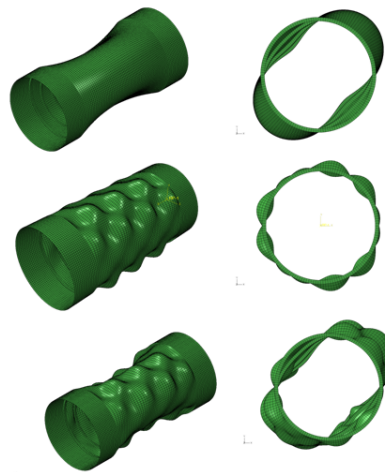


Figure 13: Overall, interframe and combined OOC shapes

used. Figure 13 shows the OOC shapes used in one of the analyses by combining measured overall and interframe modes.

The values of material properties, particularly the yield stress, also significantly affect the elasto-plastic collapse load. In general, the lower the yield stress, the lower the collapse load, unless failure is dominated by elastic buckling for very thin shells. It is difficult to distinguish the effects of material behaviour alone in Table 13 as the OOC values also vary.

The analytical methods predicted surprisingly good results, although are conservative for the corroded model case as it is necessary to assume thinning around the full circumference. There were no clear differences between finite element codes, although there are differences. This subject is mentioned in Chapter 3 where a discussion of the need to develop a protocol and safety factors for application of FEA to submarine collapse analysis is provided.

7 DISCUSSION AND CONCLUSIONS

From the discussion above it can be concluded that the larger part of the structural methods and calculations are common for naval and commercial ships, only with mi-

nor differences in characteristic values. This means that naval and commercial ship structural design can benefit from a common source of research and development of structural design methods. It also confirms the basis for using Classification Rules (so far, based on commercial ship experience) as a technical standard for naval ship structures.

The other conclusion that can be made is that the generic differences in structural design between naval and commercial ships are mainly related to the military load cases. For this area there is little common ground for exchange of methods and experience between naval and commercial structural design.

Seen in a broader perspective, the above conclusions raise some worrying questions for the naval community. The common knowledge basis for structural design through Classification Rules and Class Societies service experience is enormous. On the other hand, the knowledge basis for the military loads is small compared to this. As an example: a medium size Class Society like Det Norske Veritas is logging close to 6000 years of service experience per year for civilian ships. On the other hand, the corresponding service experience for naval ships is in the order of 100 years combined experience per year. In addition to this, the specific service experience on military loads is practically none. The question is then: How is the military loads taken care of in the future? How will the technical basis be maintained, and how will the personal knowledge and skills be maintained in the future?

Lightweight materials have great potential to save cost and improve performance for naval vessels. Some materials will come with restrictions that limit their application or have their weight savings reduced by additional concerns; however, optimization may be achieved in a logical and conservative manner. The cost savings demonstrated by the LASS project show a substantial benefit in fuel savings for a medium sized, high speed vessel that would be comparable to many naval ships. Furthermore, weight savings could be used to carry more fuel, cargo, or weaponry to enhance mission capability or used to reduce power (fuel) demand. Also, the inherent corrosion protection of aluminium, titanium, and FRP can help reduce maintenance costs and operational time lost to repair. Lastly, FRP construction is known to restrict thermal and acoustic radiation and offers very flat surfaces which makes the vessel less “visible” to sensors: thermal, acoustic, and RADAR; resulting in appreciable stealth benefits.

Submarine design methods have been discussed and it has been shown that much progress has been made with respect to standardizing numerical models for pressure hull collapse predictions, and furthermore, a significant amount of experimental-numerical data have been generated in support of quantifying the accuracy of the FE models. The most pressing needs, if FE methods are to be incorporated in hull design, are consensus regarding the best way to incorporate residual stresses in the analysis, further expansion of the experimental-numerical database in order to improve overall confidence in the FE results, and a set of rules defining the shape and magnitude of geometric imperfections for design.

As a final note, it is not expected that numerical methods will completely replace conventional pressure hull design curves and equations. The traditional analytical-empirical methods will likely be retained because of their simplicity and efficiency of use, as well as their value for use in iterative design procedures such as optimization routines and reliability analysis e.g. Radha and Rajagopalan (2006), Morandi *et al.* (1998). Numerical modeling is more likely to complement than to replace the conventional methods, as in a hierarchical design procedure, whereby analytical-empirical

methods are used to conduct parametric studies of design variables, and to determine the nominal dimensions of the structure. Nonlinear FEA is then used to determine the design strength, either in a deterministic or probabilistic (i.e. reliability) setting.

Two benchmark problems were analysed by the committee members, these were:

1. Plate subjected to air blast pressure loading
2. Collapse analysis of ring stiffened cylinder subjected to external pressure loading

For both problems the results from a variety of alternative theoretical/numerical solutions were compared with existing experimental data. The results of these two studies are discussed in some detail in Chapter 6 of the report.

The importance of Residual Strength of damaged ships is highlighted in Chapter 5 of this report. An overview is given.

8 RECOMENDATIONS

Although the Report of ISSC Committee V.6 in (2006) gave extensive coverage of military load effects it is recommended that the next ISSC naval committee focuses on the military loads, vulnerability especially the more sophisticated fluid/structure interaction theoretical methods for predicting the effects of Underwater Explosions (UNDEX), Shock and Blast which are currently being employed to replace experimental testing. The subject of the residual strength of both intact and damaged of naval ships should also be a major focus of the next committee. It is also recommended that benchmark studies should be carried out to investigate these topics.

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18th INTERNATIONAL SHIP AND
OFFSHORE STRUCTURES CONGRESS
09-13 SEPTEMBER 2012
ROSTOCK, GERMANY
VOLUME 2



COMMITTEE V.6
ARCTIC TECHNOLOGY

COMMITTEE MANDATE

Concern for development of technology of particular relevance for the safety of ships and offshore structures in Arctic regions and ice infested waters. This includes the assessment of methods for calculating loads from sea ice and icebergs, and mitigation of their effects. On this basis, principles and methods for the safety design of ships and fixed and floating structures shall be considered. Recommendations shall also be made regarding priorities for research programmes and efficient implementation of new knowledge and tools.

COMMITTEE MEMBERS

Chairman: George Wang
Alexie Bereznitski
Wim C. de Boom
Claude Daley
Frank W. DeBord
Shunyin Ji
Joong K. Kang
Per R. M. Lindstrom
Per O. Moslet
Jukka Tuhkuri
Akio Usami
In S. Yoo

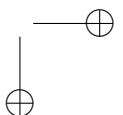
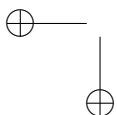
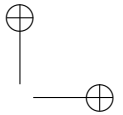
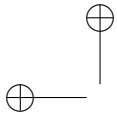
KEYWORDS

Arctic structures, ice loads, ice class rules, arctic environment.

18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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Proceedings to be purchased at <http://www.stg-online.org/publikationen.html>



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1 INTRODUCTION

During the late 1970's and early 1980's, the interest in Arctic research and development was very high due to anticipated resource development. In the late 1980's, the interest in Arctic development dropped and consequentially the volume of related R&D declined to a minimum. This trend changed in the late 1990's and early 2000's when global warming became a global topic of interest. Evidence reveals that the ice cap in the Arctic has been shrinking year by year. The Northern Sea Route (NSR), which was historically impassable, has been opened up for a small number of commercial ships during summer time. Recently, the USA government announced permitting further drilling in certain areas offshore Alaska. All these may imply the coming of another boom of Arctic development.

These recent demands resulted from interest in exploring for oil and gas in the Arctic and the potentials of commercial shipping using the Arctic routes. Figure 1 shows the Arctic ice cap that has been found to be retreating year by year. Accompanying this trend, research on ice-going ships and Arctic structures has also been revived.

Of particular importance to the R&D community are:

- Development of ice class rules and recommendation: Finnish-Swedish Ice Class Rules (FSICR), IACS Polar Class Rules, ISO 19906 Arctic Offshore Structures
- Application of risk assessment to supplement rules
- Ice loads measurement, prediction, and simulation
- Design and innovation of ice-going ships and Arctic structures
- Expanded scope of research to include winterization, escape, evacuation and recovery (EER), recovery of spilled oils, ice management

This Committee intends to cover recent R&D activities that are directly related to hull structural designs. Emphasis is therefore placed on:

- Design of ice-going ships and Arctic structures
- Rules, regulations and design guidance
- Ice loads and simulation of ice
- Application of structural reliability approaches (SRA)

This committee report concludes with recommendations for future research.

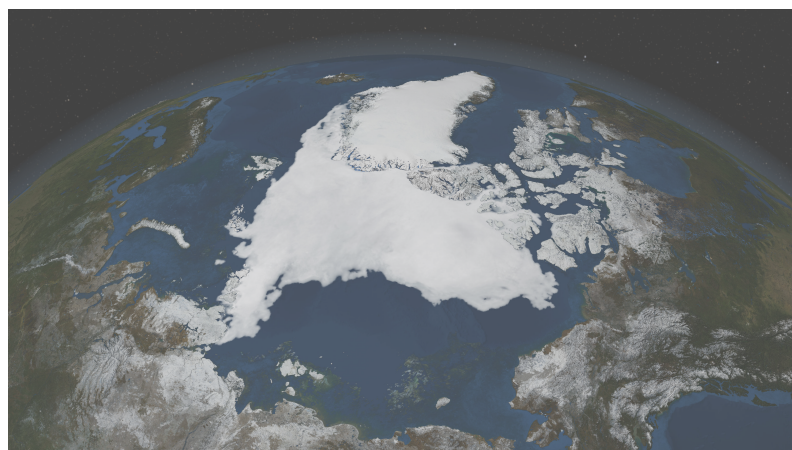


Figure 1: Arctic ice cap (http://mapas.owje.com/maps/10540_arctic-satellite-map.html)

2 ENVIRONMENT AND CLIMATE CHANGE

As far as structural safety of ice-going ships and Arctic structure is concerned, climate change (or global warming) would cast the following questions related to the current design practice:

- Are the existing rules, regulations and guidance adequate to address the structural design at time of changing climate?
- What changes will climate change bring to current design practice? Specially, will design ice loads increase or decrease?
- Are we prepared for the potential risks associated with the increased number and frequency of ships navigating in the Arctic region due to the extended navigational season and also the risk associated with cruise vessel visiting remote areas in the Arctic?
- What must be done to minimize and mitigate potential environmental impact of Arctic shipping and Arctic structures on the pristine environment in the Arctic?

2.1 Changing Sea Ice in the Arctic

According to the National Snow and Ice Data Center, Arctic sea ice extent is declining at a rate of 3.5 % per decade. The five lowest December extents in the satellite record have occurred in the past six years (Figure 2). Particularly, the Arctic ice cap in summer 2007 was $4.2 \cdot 10^6 \text{ km}^2$, which marked the lowest record (23 % less than the high record of September 2005). Some studies estimate that the Arctic could become ice-free during the summer months in a few decades (Wang *et al.*, 2009). Reports also suggest increasing variability in ice extent than before.

In-situ measurements have reported that ice of the Arctic has been thinning (Rothrock *et al.*, 1999). Substantial amounts of older perennial ice have been observed drifting out of the Arctic through the Fram Strait (Rigor and Wallace, 2004).

These environmental changes may result in a need for re-evaluation of ice loads that are the basis of structural design. So far, there is only very limited research on the potential changes in ice loads based on the long-term decreasing trend in measured peak ice loads (Matsuzawa *et al.*, 2010). Melting ice gives rise to the likelihood of iceberg collision (Hill, 2006), which has not been adequately addressed in the existing design codes or safety regulations.

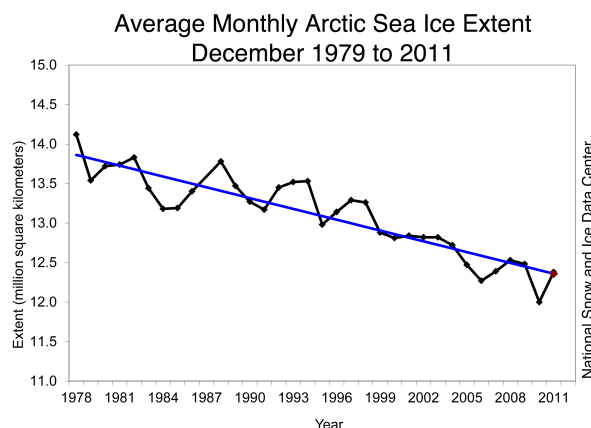


Figure 2: Decline of Arctic sea ice extent (<http://nsidc.org/arcticseaicenews/>)

2.2 *Environmental Concerns*

Commercial shipping and offshore rigs in the Arctic also raise significant concerns over oil spillage. Ice and cold temperature will make it very difficult to contain and recover spilled oil as most of current technologies will not be effective in cold water. The current MARPOL Convention Annex I does not designate the Arctic Sea as “*Special Area*” where un-conventional means of oil spill protection are required. This may become an issue for Arctic shipping and Arctic exploration.

3 ARCTIC SHIPS

3.1 *Overview*

The diverse range of activities in the Arctic and Antarctic, like increased shipping and oil and gas developments, requires (will require) operation of a wide range of vessel types and sizes. Operational experience to date has primarily been limited to escort and research icebreakers and relatively small cargo ships, coastal tankers and bulk carriers. Recently built icebreaking tankers have deadweight capacities less than 100,000 tonnes even though much larger sizes have been proposed for tankers, LNG carriers and bulk carriers since the early 1970s. Commercial resource developments will also require supply vessels, tugs, and dedicated icebreakers. Finally, governments intending to enforce laws and provide emergency response will need a year-round presence in all areas with commercial development and along proposed shipping routes. A variety of different vessels will be required to satisfy these needs.

Because these vessels are (will be) designed to operate in a wide range of ice conditions and climates, some operators will elect to operate year-round and others will choose seasonal operations. Depending on the specific geographic area of operation and season, design ice conditions could include:

- Open water with occasional small, thin ice floes
- First year ice with coverage from 5 to 100 % and thicknesses from several centimetres to two meters
- Compact first-year ice with large pressure ridges and rafting
- Thick multi-year ice with weathered consolidated pressure ridges

Other possible operating conditions would include open water with occasional large ice features such as icebergs, bergy-bits, growlers or ice floes.

3.2 *Research and Development of 1990's and 2000's*

The last time ISSC had a committee on Arctic technology was more than 20 years ago. Since then, IACS Polar Class Rules have been developing and the Finnish-Swedish Ice Class Rules have been refined. The development of these rules was supported by the results of the projects HELCOM, SAFEICE, BARENTS 2020 and others. In addition, commercial organizations have invested significant resources in research and development projects related to oil and gas exploitation in ice-infested seas in Russia and Alaska.

3.3 *Arctic Vessel Design*

This section uses an example to illustrate the design of Arctic vessel. The focus is placed on the design basis including selection of ice class and comparison between different ice classes.

Figure 3 shows the concept of a modern Arctic tanker design. A variety of issues must be addressed during design, including but not limited to ice-breaking bow design, ice-strengthening of hull structures, propulsion system designs, winterization of hull and

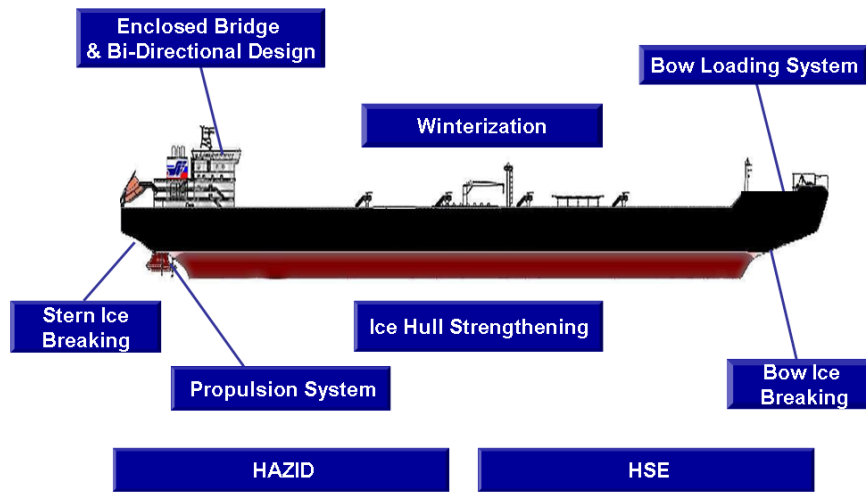


Figure 3: Concept of an Arctic tanker (modified from Kwak *et al.*, 2010)

machinery systems, bridge design, comfort of crew and passenger, and operation in cold environment (Kwak *et al.*, 2010; Dolny *et al.*, 2010). This section only addresses structural design.

3.3.1 Ice Class

The tanker shown in Fig. 3 was intended for year-round operations in the Barents Sea without ice breaker support. According to RMRS Ice Class Rules (see also Section 5.4), the ice class was selected to be ARC 6. The corresponding ice class in IACS PC (see also Section 5.3) is PC 4. Therefore, this tanker was also re-designed to satisfy IACS Polar Class rule PC 4.

3.3.2 Ice Loads

Table 1 shows the ice pressure that PC 4 and ARC 6 specify for design of plating/stiffeners (or local pressure as discussed in Section 7.1). The ice pressure of PC 4 is slightly higher than that of ARC 6, while the ice loads (patch load) of PC 4 are much smaller than ARC 6. This was not fully expected as normally it is believed that ice loads levels for ARC 6 and PC 4 are the same.

Table 1: Design ice loads of PC 4 and ARC 6 (Kwak *et al.*, 2010)

Ice strengthening structure areas by side depth		Ice strengthening structure area by hull length			
		B	BI	M	S
ice pressure [MPa]	Icebelt	6.70	5.36	3.58	3.91
	Lower	6.70	4.02	2.28	1.95
	Bottom	6.70	2.68	0.00	1.63
The vertical distribution of ice pressure b [m]		b_B	b_{BI}	b_M	b_S
		0.90	0.90	1.24	1.24
Horizontal distribution of ice pressure l_p [m]		w_B	w_{BI}	w_M	w_S
		4.24	4.24	4.45	4.45

(a) IACS Polar Rule

Ice strengthening structure areas by side depth		Ice strengthening structure area by hull length			
		p_A	p_A1	p_B	p_C
ice pressure [MPa]	I	6.55	5.77	3.27	3.27
	II	4.26	3.75	1.64	1.64
	III	4.26	3.75	1.47	1.14
	IV	3.28	2.89	0.00	0.49
The vertical distribution of ice pressure b [m]		b_A	b_{A1}	b_B	b_C
		1.10	1.20	1.65	1.32
Horizontal distribution of ice pressure l_p [m]		l_A	l_{A1}	l_B	l_C
		11.09	10.73	9.87	7.90

(b) RMRS Rule

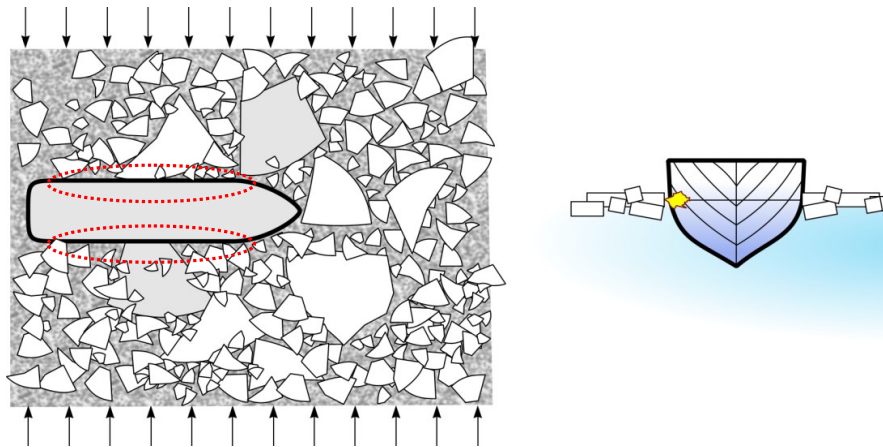


Figure 4: An example of ship-ice collision scenarios not addressed in ice class rules

3.3.3 Structural Design

The bow is designed to be capable of breaking ice (Figure 3). It is transversely framed because transverse framing systems are more efficient in resisting high ice loads. In order to investigate the impact on steel weight and labour costs, three structural designs were considered for the mid-body region. FEM was used to evaluate the structural responses for the rule-based ship-ice interaction model and also many other scenarios.

3.3.4 Scenario-based Evaluation

In addition to the basic rule check, the design team decided to evaluate the structural responses against ship-ice collision scenarios that are likely to take place but not covered in the ice class rules. Additional scenarios include head-on ramming collisions, thick ice flow oblique bow glancing collisions, ice compression in alternate patterns (Figure 4), and thick level ice oblique mid-body glancing collisions.

4 ARCTIC OFFSHORE STRUCTURES

4.1 Overview

The extensive offshore exploration activities in Canada and Alaska during the 1960's through the 1980's were mostly land based. In 1983, a specially designed drilling unit, Kulluk, was put into operation, drilling in limited level ice. In addition, oil and gas has been produced in approximately 50 m water depth using jacket wellhead platforms and jack-up based production.

Along the Canadian East Coast oil, Hibernia, Terra Nova and White Rose fields use production facilities that are either bottom-founded, "iceberg proof" or disconnectable FPSO's which can leave their locations when threatened by icebergs.

In the Russian Arctic region, the northern oil and gas activities are also mainly onshore. The Varanday field includes an offshore loading facility approximately 21 km from shore in 17.5 m water depth. Oil is loaded to shuttle tankers with icebreaking capacity.

The Prirazlomnoye Oil Field will adopt a square ice-resistant gravity platform (Velikhov *et al.*, 2010). This innovative platform will be built at SEVMASH of Severodvinsk, towed to the field and ballasted down to sit on the seabed. It combines all functions of drilling, production, storage and offloading.

The Sakhalin offshore field development in the Sea of Okhotsk uses concrete gravity-base platforms. Field development is progressing but no further offshore structures have been installed in the reporting term of this report.

Research and development work has been reported for the Shtokman development, which is awaiting a go-ahead decision. This project will use a floating production unit, moored by a turret (Marechal *et al.*, 2011). The design will be capable of resisting significant ice loads and will be disconnected in cases where a threat may exceed the design limit.

4.2 Recent Activities

Current research and development into arctic offshore structures focuses primarily on exploration drilling and floating production units.

Arctic floating structures normally remain at a certain operating site for months. Their operation window can be 3 to 9 months long per year. Production units will have to stay on location year round. This means that these offshore floating structures will have to be heavily reinforced against ice loads. This also means that the station keeping will have to be ensured by utilization of extremely high capacity mooring systems, possibly, still supported by the ice management when the ice conditions become too severe.

The direction of the ice drift is difficult to predict and the offshore arctic unit must be prepared to meet ice coming from any direction. One of the design solutions for ship shaped units is the application of a turret. Here a care should be taken that the ship will be always keeping the bow (or stern) against the drifting ice (Zhou *et al.*, 2011; Hidding *et al.*, 2011). Another solution is utilization of a circular shape unit. A good example is the existing drilling unit Kulluk (Gaida *et al.*, 1983; Loh *et al.*, 1984; Wright *et al.*, 1999; Wright *et al.*, 1998). Additionally, circular shape units (i.e. SEVAN concept) are being proposed (Dalane *et al.*, 2009; Loset *et al.*, 2009; Bezzubik *et al.*, 2004; Bereznitski *et al.*, 2011; Bereznitski 2011).

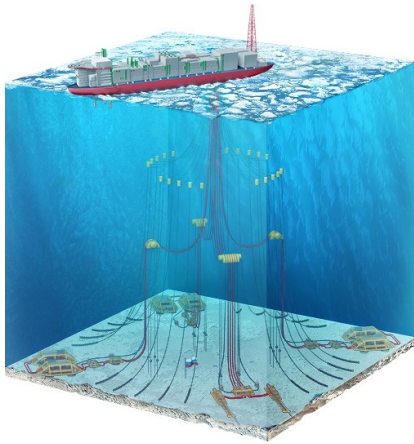
Doelling *et al.* (2010) presented the design of the Aurora Borealis, an icebreaking research vessel, developed under a grant from the European Commission. This vessel features interesting novel concepts to keep station in level ice based on dynamic positioning (DP). The drilling capabilities, however, are designed for scientific coring, not for oil/gas exploration drilling. The vessel is in the design stage.

A number of Arctic drillship designs have been introduced for year-round operation in ice-covered waters. Due to confidentiality restrictions, only a few publications about these developments are available in literature.

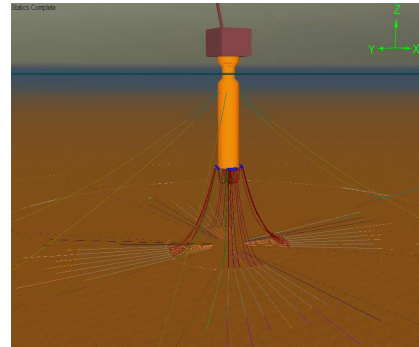
Concepts for floating production systems have been presented in recent literature. Figure 5 shows some of the proposed Arctic floating structures. The afore-mentioned vessel shaped FPU planned for Shtokman is probably most progressed. Sablok *et al.* (2011) presented an Arctic Spar. The unit has a disconnectable keel buoy (bottom part of the Spar body) which carries the risers when the Spar has to be moved out of location in case of an ice threat exceeding the ice design conditions. Srinivasan and Sreedhar (2011) proposed a circular FPSO for Arctic Deepwater. The unit has sidewalls designed to provide adequate ice-breaking capabilities.

4.3 Mooring and Structural Designs

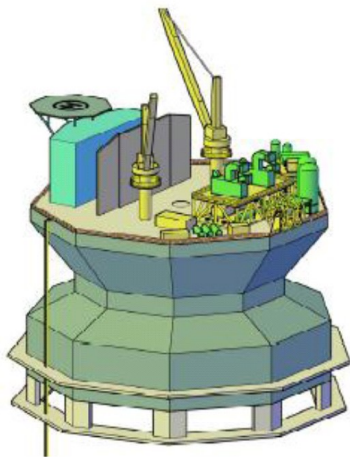
The ISO 19906 standard gives a general basis for design of Arctic offshore structures. The design has to be further developed by following the design standards from classi-



(a) Shtokman field (Marechal *et al.*, 2011)



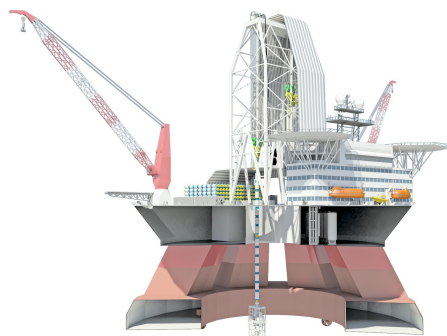
(b) Disconnectable spar (Sablok *et al.*, 2011)



(c) Circular FPSO (Srinivasan *et al.*, 2011)



(d) Alternative circular FPSO (SEVAN concept) (Dalane *et al.*, 2009)



(e) Circular MODU (Bereznitski, 2011)

Figure 5: Some proposed concepts for Arctic floating structures

fication societies. It is also necessary to strike a balance between requirements for ice sea environment during winter and open sea environment during the summer.

4.3.1 Mooring System in Ice

The holding capacity of the mooring systems designed for Arctic ice conditions will be typically much higher than the open water mooring. A disconnection procedure may be needed at the time of emergency, e.g., when a severe ice condition is forecast to exceed the capacity of the mooring system. The drilling units connected to the seabed with riser can have a very small positional offset, especially in shallow water. This small offset requirement in combination with high ice loads makes the design of mooring system extremely challenging.

A number of codes can be applied for the design of mooring system such as API-PR-2SK, DNV-OS-E301, ISO-19901. However, the safety factors are not clearly defined.

4.3.2 Ice Loads

The ice class rules for ships can be directly applied to ship shaped floating structures. API and ISO19906 can be referred to for floating structures. Challenges remain to define ice loads on non-ship shaped structures.

4.3.3 Ice Management

Ice management (IM) normally includes a system to detect large ice features in advance and employ standby ice-breakers to assist in diverting or destroying dangerous large ice features (e.g. by supply vessels towing icebergs). IM has been found to be effective in extending a rig's operating season, ensuring station-keeping and increasing the operability of floating structures (Wright, 2000, Coche *et al.*, 2011). IM should be considered during design of floating structures.

5 RULES AND REGULATIONS FOR ICE-GOING SHIPS

5.1 Ice Class Rules for Ships

Ice class rules play a central role in the design of ice-going ships. The most important ice class rules are:

- Finnish-Swedish Ice Class Rules (FSICR)
- IACS Polar Class Rules (IACS PC)
- Ice class rules of classification societies (ABS, BV, CCS, DNV, GL, LR, NK, RMRS)

Ice class rules specify requirements based on ice conditions and operation of vessels. Details of structural requirements appear to be based on a combination of experience, empirical data and structural analyses.

The FSICR have been adopted widely and have been incorporated by most classification societies as the basis of first-year ice conditions. The exception is RSMS Ice Rules for vessels navigating in the Russian Arctic waters. Other than FSICR and RSMS Ice Rules, few existing ice class rules have actually been used to design ships. The IACS PC Rules are becoming more and more accepted, especially for multi-year ice conditions. To supplement FSICR and IACS PC Rules, some classification societies have rules for icebreakers and guidance on winterization for operation in cold environment. See Table 2 for a summary of some existing ice class rules.

As far as structural requirements are concerned, the following are the key components of ice class rules:

Table 2: Ice class rules for ships (Most classification societies except RSMS have aligned their first-year ice class, ice-strengthening requirements with FSICR, and are implementing IACS PC)

Ice class rules	Multi-year ice	First-year ice	Ice-strengthening	Ice breaking	Notes
FSICR	-	×	×	-	<i>De-facto</i> standard for 1st year ice
IACS PC	×	×	×	×	PC 6, 7 aligned with FSICR 1A+, 1A
RSMS	×	×	×	×	Russian region
ABS	×	×	×	×	Supplemental guidance on winterization, ice load monitoring, enhanced PC class
DnV	×	×	×	×	Supplemental requirements on winterization
LR	×	×	×	×	Supplemental requirements on winterization, ice-induced fatigue
NK	×	×	×	×	

- Ice classes that correspond to ice conditions and vessel operations in ice-infested seas
- Areas of ice strengthening that are normally divided into bow, parallel body and aft regions in general
- Ice loads that are associated with various ice-ship interaction scenarios
- Scantling requirements that are dependent on elastic or plastic responses of structures
- Corrosion/abrasion allowance

One of the issues currently facing owners and designers is selection of the appropriate design standards (Daley *et al.*, 2007). A significant amount of experience has been developed for government and escort icebreakers, icebreaking oil field work vessels and small cargo vessels. However, very little information has been published related to the adequacy of design standards used for these vessels. Currently, experience with larger tank vessels is being accumulated and industry is developing designs to support oil and gas exploration in several Arctic regions. Large state-of-the-art icebreaking tankers have recently been constructed and are now providing year round service to the Varanday gravity based production platform offshore in the Russian arctic (Iyerusalimsky and Noble, 2008). This project includes much needed collection of full-scale ice loads data for application to the design of larger vessels anticipated for future commercial developments.

5.2 Finnish-Swedish Ice Class Rules (FSICR)

FSICR were primarily intended for merchant ships trading in the winter Baltic. The rules are based on the premise that icebreaker assistance is available when required. FSICR define four ice classes, which are IA Super, IA, IB and IC (Table 3). Requirements are specified for minimum propulsion power, hull and machinery scantlings.

Over the time, FSICR has become the de-facto global standard for designing ice-strengthened ships for first-year ice condition. The latest update in 2010 streamlined the hull rules (Riska and Kamarainen, 2011).

Table 3: Ice classes of FSICR (TRAFI, 2010)

Ice class	Ice condition and vessel operation
IA Super	ships with such structure, engine output and other properties that they are normally capable of navigating in difficult ice conditions without the assistance of icebreakers;
IA	ships with such structure, engine output and other properties that they are capable of navigating in difficult ice conditions, with the assistance of icebreakers when necessary;
IB	ships with such structure, engine output and other properties that they are capable of navigating in moderate ice conditions, with the assistance of icebreakers when necessary;
IC	ships with such structure, engine output and other properties that they are capable of navigating in light ice conditions, with the assistance of icebreakers when necessary;
II	ships that have a steel hull and that are structurally fit for navigation in the open sea and that, despite not being strengthened for navigation in ice, are capable of navigating in very light ice conditions with their own propulsion machinery;
III	ships that do not belong to the ice classes referred to in paragraphs 1-5;

The scenario considered in FSICR is that a ship collides with a level ice edge while sailing in the ice channel at a speed of about 4 knots. The ice channel is created by the escort icebreaker.

The ice load on hull is a patch that is narrow in height, which is often simplified as a line load. The design ice loads were defined and updated based on ice loads measurements and observed damages to ships.

Recent statistical studies of ice load measurements suggested (Figure 6) that the FSICR design ice loads have a return period of 3.5 to 14.6 days. Measurement data on real ships have revealed that the FSICR design ice loads have been repeatedly exceeded. In comparison, modern commercial ships (such as the IACS Common Structural Rules) are designed for environmental loads with a return period of about 25 years.

FSICR uses formulation of initial yielding for shell plates and formulation of elastic response for frames (i.e. shell stiffeners).

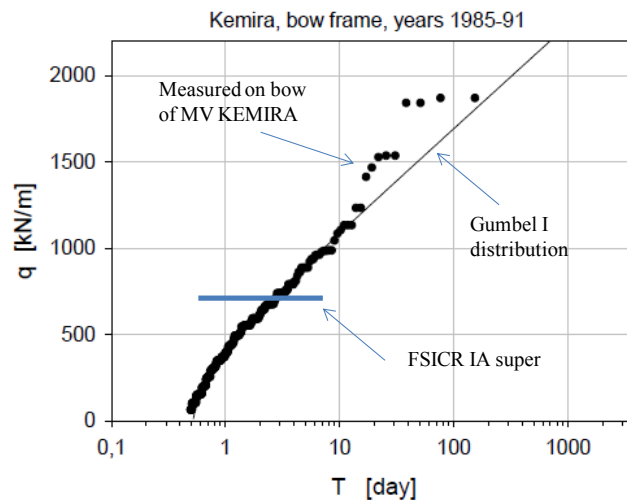


Figure 6: Measured ice pressure and design ice loads of FSICR (according to Riska and Kamarainen, 2011)

Table 4: Ice classes of IACS PC Rules

Polar class	Ice condition
PC 1	Year -round operation in all Polar water
PC 2	Year -round operation in moderate multi-year ice conditions
PC 3	Year -round operation in second-year ice which may include multi-year ice inclusions
PC 4	Year -round operation in thick first-year ice which may include old ice inclusions
PC 5	Year -round operation in medium first-year ice which may include old ice inclusions
PC 6	Summer/autumn operation in medium first-year ice which may include old ice inclusions
PC 7	Summer/autumn operation in thin first-year ice which may include old ice inclusions

5.3 IACS Polar Class Rules (IACS PC)

The IACS PC Rules define seven ice classes (Table 4), PC 1 to PC 7 with the lowest ice class PC 7 approximately aligned with FSICR IA class. IACS PC is intended to cover the full range of ships operating in multi-year and first-year ice conditions.

A notable feature of IACS PC is that a wider range of shell, including bottom shell, is required to be ice-strengthened. This might stem from the consideration that ice is pushed passing the bottom of a ship as the ship advances in more open water with swells.

The considered scenario is that a ship strikes an angular ice edge at the design speed (Figure 7). The ship penetrates the ice and rebounds. The assumption is that the ice loads are determined by the ice's crushing and flexural strength.

In addition, global hull-girder loading due to ramming operation is also specified.

The return period of IACS PC ice loads is not documented.

The IACS PC rules use plastic response formulation for plate and stiffeners (Daley, 2002a, 2002b). The interaction between bending and shear is considered in calculation of stiffener's load-carrying capacity.

5.4 Russian Maritime Register of Shipping Ice Rules (RMRS IR)

The RMRS IR is also important because of Russia's proximity to the Arctic.

The current RMRS IR has a total of 12 ice classes: three for non-Arctic ice conditions, six for Arctic operations, and three for ice-breakers (Table 5). The rules specify requirements for permissible operation condition that are based on permissible vessel speed and ice conditions (Table 6), which are contingent upon operation areas, seasons, navigation severity and availability of an escort ice-breaker.

The basis of ice load in RMRS IR is said to be a hydrodynamic model of solid body - ice interaction.

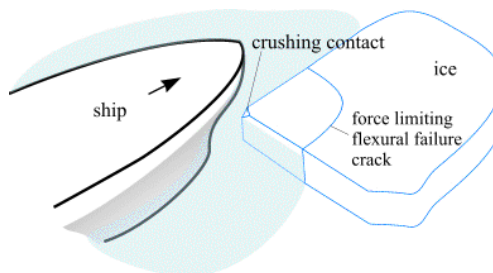


Figure 7: Ice-ship interaction scenario of IACS PC Rules (Daley, 2002a)

Table 5: Ice class category by RMRS IR

Ice class	Ice condition, operation
ICE 1, ICE 2, ICE 3	Non-Arctic ice condition
ARC 4, ARC 5, ARC 6, ARC 7, ARC 8, ARC 9	Arctic operation
Ice breaker 6, Ice breaker 7, Ice breaker 8, Ice breaker 9	Ice breaker

Table 6: Permissible service area for ships of Arctic classes by RMRS IR

Ice class	Ice operation tactics	Winter – spring navigation					Summer – fall navigation				
		1	2	3	4	5	1	2	3	4	5
		EHML	EHML	EHML	EHML	EHML	EHML	EHML	EHML	EHML	EHML
ARC 4	IO	---+	----	----	----	----	++++	--++	---+	---+	---+
	PO	*+++	---+	----	----	---	++++	*+++	--++	---+	*+++
ARC 5	IO	--++	---+	----	----	----	++++	++++	--++	---+	---+
	PO	*+++	--++	---+	---+	--*	++++	*+++	*+++	*+++	*+++
ARC 6	IO	*+++	---+	---+	---+	---+	++++	++++	++++	++++	++++
	PO	++++	**++	**+	**+	*++	++++	++++	++++	++++	++++
ARC 7	IO	++++	--++	---+	---+	---+	++++	++++	++++	++++	++++
	PO	++++	++++	*+++	*+++	*+++	++++	++++	++++	++++	++++
ARC 8	IO	++++	++++	*++	*++	*++	++++	++++	++++	++++	++++
	PO	++++	++++	++++	++++	++++	++++	++++	++++	++++	++++
ARC 9	IO	++++	++++	++++	++++	++++	++++	++++	++++	++++	++++
	PO	++++	++++	++++	++++	++++	++++	++++	++++	++++	++++

Legend:
 IO — independent operation
 PO — icebreaker pilotage operation;
 + — service is permissible;
 - — service is impermissible;
 * — service is connected with increase of risk to be damaged;
 E — extreme navigation (with average reoccurrence one time per 10 years);
 H M, L — heavy, medium, light navigation (with average reoccurrence one time per 3 years).

The RMRS IR adopts plastic capacity limit for the stress criteria.

5.5 IMO Guidelines for Ships Operating in Polar Waters

The International Maritime Organization (IMO) adopted Guidelines for Ships Operating in Polar Waters in December 2009. These guidelines augment the Safety of Life at Sea (SOLAS) and Prevention of Pollution from Ships (MARPOL) International Conventions. They include provisions related to vessel construction, equipment, operations, environmental protection and damage control. The current IMO guidelines are in the process of further revision and are due to become mandatory in the near future.

These guidelines refer to the IACS Polar Class Rules for the detailed hull and machinery requirements.

5.6 Canadian Arctic Shipping Pollution Prevention Regulations (CASPPR)

The CASPPR was established in 1972 as one of the sub-laws required under Arctic Water Pollution Prevention Act (AWPPA), which is a basic act put in force in 1970 to prevent marine pollution from offshore resources development in Canadian Arctic waters.

The CASPPR defines five “Canadian Arctic Class” for ice breakers, and four Types for ice-strengthened ships (Table 7). While not specifying structural requirements

Table 7: CASPPR ice classes

Ice class	Max. allowable ice type	Ice thickness (<i>cm</i>)
CAC1	No limit	No limit
CAC2	Multi year	No limit
CAC3	Second year	No limit
CAC4	Thick first year	> 120
Type A	Medium first year	70 – 120
Type B	Thin first year (Stage 2)	50 – 70
Type C	Thin first year (Stage 1)	30 – 50
Type D	Grey white	15 – 30
Type E	Open water / Grey	10 – 15

for Type A to D ships, the CASPPR accepted equivalency of ice classes by major classification societies, and consequentially established equivalency with FSICR.

A “Shipping Safety Control Zones” scheme has been long implemented under AWPPA, under which the whole area is split into 16 zones based on the sea ice statistics. AWPPA forces ships attempting to enter into these zones to comply with the requirements on ship construction, propulsion system, equipment and crew competence, etc. The details of these requirements are stipulated in CASPPR that translates the scheme into “Zone/Date system” (Z/DS) in which the operable period for each zone and ice class combination is specified for easy reference.

The Z/DS, however, is founded on statistics from the 1970s which do not necessarily reflect the present conditions. Therefore, conflicts have been reported between the data and the actual ice conditions.

In response to this situation, the “Arctic Ice Regime Shipping System” (AIRSS) has now been put in place to supplement the existing Z/DS. AIRSS is a regulatory standard currently in use only outside Z/DS and it emphasizes the responsibility of the ship owners and captains while providing a flexible framework for decision-making.

5.7 Supplemental Guidance

Supplemental requirements have also been developed to address issues generally not covered by ice class Rules. These include guidance on temperature and ice thickness of selected areas, vessel operation under low temperature, ice load measurement, ice-induced fatigue, propulsion system, additional machinery requirements, analysis of structures for ship-ice interaction scenarios that are not addressed in existing ice class rules (i.e., ABS, 2010, 2011, 2012; DNV, 2011; LR, 2008).

Low temperature environments present numerous challenges related to operation of equipment, systems, structure, vessel maintenance and safety equipment. Vessels designed and constructed without addressing the effects of low temperatures may experience increased structural and equipment failures and non-functioning systems.

The technical developments that led to the IACS PC also allow for extended structural evaluation for additional ice/ship interaction scenarios (see also Section 5.2 and 3.3). Guidance has been developed to describe supplementary loading scenarios and associated structural analysis (ABS, 2012). Procedures for grillage analysis have been developed for analyzing side structures of wider extent (ABS, 2012). Non-linear FEM analyses have also been accepted for evaluation of these additional cases.

6 GUIDANCE FOR ARCTIC STRUCTURES

6.1 ISO 19906 Arctic Offshore Structures

ISO 19906 Petroleum and Natural Gas Industries - Arctic Offshore Structures specifies requirements and provides guidance for the design, construction, transportation, installation, and decommissioning of offshore structures, related to the activities of the petroleum and natural gas industries, in arctic and cold regions environments. The objective is to ensure that arctic and cold regions offshore structures provide an appropriate level of reliability with respect to personal safety, environmental protection and asset value to the owner, to the industry and to society in general. ISO 19906 does not contain specific requirements for the operation, maintenance, service life inspection or repair of arctic offshore structures.

This ISO does not apply specifically to mobile offshore drilling units (see ISO 19905-1). The procedures relating to ice actions and ice management contained herein may be applicable to the assessment of such units. Mechanical, process and electrical equipment and any specialized process equipment associated with arctic or offshore operations are not covered except insofar as the structure needs to sustain safely the loads imposed by the installation, housing, and operation of such equipment.

6.2 API Recommended Practice for Planning, Designing, and Constructing Structures and Pipelines for Arctic Conditions (API RP)

This API RP contains recommended practice to those involved in the design of Arctic systems. The systems covered in this recommended practice for the Arctic environment include:

- Offshore concrete, steel, and hybrid structures, sand islands, and gravel islands used as platforms for exploration drilling or production;
- Offshore ice islands used as platforms for exploration drilling;
- Near shore causeways
- Offshore pipelines;
- Shore crossing for pipelines.

7 ICE LOADS

Ice loads may be conveniently categorized as local ice loads and global ice loads (ABS, 2011). Local ice loads are often defined as ice pressure acting on local areas (on shell plates and stiffeners). Global ice loads on ships are typically (vertical) bending moment on hull girder. With the recent progress of research, vibratory loads, iceberg impacts and cyclical ice loads are also being discussed.

7.1 Local Ice Loads

All ice class rules define local ice pressures. Design ice loads are determined based on field measurement and model tests. Simulations may eventually be used for deriving ice loads once the technology becomes matured.

In general, local ice pressures depend on ice type, ice thickness, ice-structure interaction, dominant ice failure modes. The load is on a small contact area, which forms where ice fails. Lab tests (Wells *et al.*, 2011) have shown that the ice most likely fails in either crushing mode or bending mode.

The average ice pressure is considered to be proportional to the contact area to the power of n . This constant n is found to be -0.52 from a study on data measured at ships (Figure 8). It is taken as -0.5 in DNV Rules and -0.3 in IACS PC.

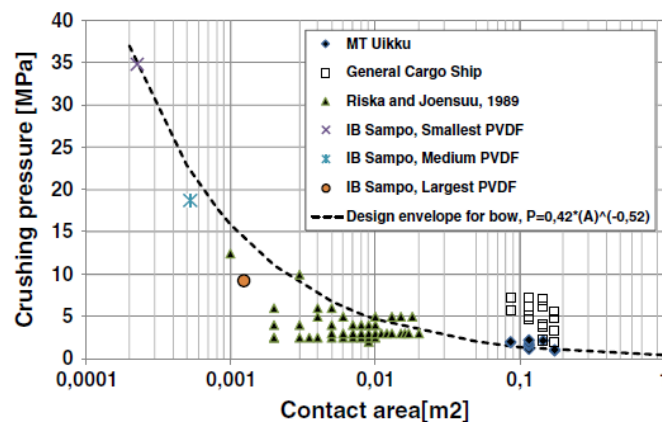


Figure 8: Ice pressure versus contact area (Kujala and Arughadhoss, 2011)

Probabilistic analysis of local ice loads has attracted some attention (Taylor *et al.*, 2009; Li *et al.*, 2010).

7.2 Global Ice Load on Ships

Some ice class rules (IACS PC) also specify global ice loads. The ice-induced vertical bending moments were derived from stresses measured at the deck of ships sailing in ice water (e.g. Chernov, 2009). The global bending moment is dependent on ship operation (ship speed and power), ice conditions (ice concentration, thickness and floe size), and ship-ice interaction.

Simulation approaches have also been applied to calculating global ice loads on ships and ship motion in ice-infested seas (Valanto, 2009; Su *et al.*, 2010a; Sayed and Kubat, 2011).

The peak ice-induced bending moments on MT Uikku were found to follow the Weibull distribution (Kujala *et al.*, 2009). The mean and standard deviations of the peak ice loads were said to be dependent on ice thickness.

7.3 Iceberg-Ship Collision

Simulation technique has been used to analyse iceberg-ship collision. Non-linear FEM tools are often applied (Kim *et al.*, 2011) to simulate such a collision. A major challenge is modelling of ice properties, which are highly variable depending on many parameters that are yet to be fully understood. Simplified analytical approaches were applied in some cases where the mechanisms of iceberg crushing are modelled in simplistic manners (Kierkegaard, 1993; Liu and Amdahl, 2010; Liu *et al.*, 2011c).

7.4 Ice Loads on Fixed Offshore Structures

Measurements taken in Bohai Sea (Yue *et al.*, 2009) revealed that ice may fail in ductile, ductile-brittle or brittle modes. These failure modes correspond to quasi-static loads, steady-state loads and vibratory loads, respectively.

Measurement data has been the basis of rule development. For example, data taken from Molikpaq has been instrumental in the development of ISO 19906.

To supplement the design codes and model tests, analytical and simulation tools are more and more used to assist in determination of ice loads. A major challenge is that different approaches result in different ice loads. A revisit of Molikpaq ice load

data suggested that “Historical Case” ice loads were about twice the level of the “Best Estimate Case” (Jordaan *et al.*, 2011; Frederking *et al.*, 2011). An analysis of Norströmsgrund lighthouse concluded that predicted dynamic ice loads on this lighthouse could be about 110 % higher than ISO/DIS 19906-2009 design code.

7.5 Ice Loads on Moored Floating Structures

Model tests have been relied on determination of ice loads on moored ships. The physical failure mechanisms of ice being pushed against a structure are quite complicated and include: crushing (or bending) failure of ice, ice accumulation, and ice movement around the structure. Attempts have been made to describe level ice sheets breaking against a structure (Croasdale *et al.*, 1994; Ralston, 1979; Nevel, 1992; Maattanen *et al.*, 1990), some of which have been incorporated into ISO 19906. The calculated global ice forces on conical structures are in some cases much lower than those measured in ice tank tests (Bereznitski, 2011). While the results of ice tank tests are well accepted, it is not recommended to base ice load predictions purely on ice tank tests.

Model basin tests have been reported for moored Spar (Evers and Jochmann, 2011; Bruun *et al.*, 2009, 2011), ice ridges (Dalane *et al.*, 2009), level ice (Wille *et al.*, 2011), moored FPSO (Chernetsov *et al.*, 2009), and interaction between ice and ship’s bow (Aksenes, 2011).

Analyses have been conducted to investigate mooring force in drifting ice (Aksnes and Bonnemaire, 2009; Aksnes, 2010, 2011b), pack ice loading and ice-hull friction coefficient (Woolgar and Colbourne, 2010), iceberg impact (Karlinsky and Chernetsov, 2010), time history of ice and mooring forces (Zhou *et al.*, 2011), and behaviour of a moored tanker (Karulin and Karulina, 2011).

8 STRUCTURAL RESPONSE

8.1 Elastic, Plastic Behavior of Plate and Stiffener

Ice damages to hull structures are in the form of dent, tripping, buckling, and rupture in some extreme cases (ice damage reports of e.g., Kujala, 2007). Limited plastic deformation to hull structures has been considered inevitable in ice-going ships.

Design of local structural members of shell, stiffeners and main support members is a key component in ice class rules. As shown in Table 8, the basis of scantling requirements in ice class rules varies to a great degree. A recent paper attempts to shed light on the various structural formulations using the concept of “design point” (Riska and Kamarainen, 2011), which includes a definition of the limit state of the structure and the frequency of the ice loads.

Extensive studies have been conducted to investigate the structural responses of shell plate and stiffeners subject to ice loads (Varsta *et al.*, 1978; Kendrick *et al.*, 2007;

Table 8: Basis of structural scantlings requirements of ice class rules

Ice rules	Ice loads	Limit state for plate failure	Limit state for stiffener failure
FSICR	Frequent ice load	Slight yielding	Initial yielding under bending
IACS PC	Extreme ice load	Plastic collapse	Collapse under both bending and shear
RSMS	?	Plastic collapse	Plastic collapse under bending

Daley, 2002a, 2002b; Wang *et al.*, 2004, 2005, 2006). Recent studies tend to apply the non-linear FEM (e.g., Liu, 2011). This is partially encouraged by a tentative acceptance of the Finnish Maritime Administration to use such advanced tool to evaluate structural scantlings.

8.2 Ice-Induced Vibration

Dynamic structural response has been observed in fixed structures of lighthouses, bridge piers, jackets, caissons or multi-leg structures (Peyton, 1968; Blenkarn, 1970). Reported damages to jacket structures in the Bohai Bay include global structural collapse and local damage like pipe failures due to fatigue damage and on channel markers (Ji and Yue, 2011).

Research on ice-induced vibration has also been a topic of interest. As an effective means of reducing ice-induced vibration, ice-breaking cones have been installed on offshore structures such as the Finnish Kemi-I lighthouse in the Gulf of Bothnia, the piers of Confederation Bridge in the Southern Gulf of Lawrence, offshore wind turbines foundations in Denmark, the conical narrow jacket platforms in the JZ20-2 field of the Bohai Bay, China, the Single Point Mooring system in the Sakhalin Field, and a large faceted cone at Varandey in north Russia. The advantage of cone-shaped structures is that the ice force on a conical structure is small, and that a well-designed cone can change the ice failure mode from crushing to bending.

8.3 Ice-Induced Fatigue

Ice loads are cyclic in nature. ISO 19906 specifies that fatigue limit state shall be considered in the design of Arctic offshore structures. How to assess fatigue during ice season remains un-determined.

On the basis of measurements of a chemical tanker sailing the Baltic, Bridges *et al.* (2006) concluded that fatigue may become an issue in severe winter season. On the contrary, a recent study on the measured data for large LNG carriers concluded that ice-induced fatigue damages would be negligibly smaller than that induced by wave.

Investigations into the fatigue behaviour of welded joints under low temperature (Bridges *et al.*, 2011) have been completed to develop guidance on predicting ice-induced fatigue (Zhang *et al.*, 2011).

9 NUMERICAL SIMULATION OF ICE

Numerical simulation is considered useful in studying the physical behaviour of ice failure process (Daley *et al.*, 1998). The increased computational capability has made it feasible to model larger volumes of ice using fine mesh, and thus to analyse the complicated failure mechanics of ice ridges.

This section reports recent numerical modelling efforts on constitutive modelling and failure of ice, failure of ice against offshore structures and ships, ice ridges, ridge strength and ridge loads. The focus is placed on sea ice related to design of ships and offshore structures. Modelling used in geophysical studies on large sea areas is therefore not reviewed.

9.1 Constitutive Modelling and Failure of Ice

As a material, ice creeps when loaded slowly, and fractures when loaded rapidly. The behaviour of ice depends on grain structure, loading direction, temperature, salinity and so on (Schulson and Duval, 2009; Weeks, 2010; Timco and Weeks, 2010). It is challenging to consider all of these properties in one single ice model.

A practical way is to apply different models for different ice behaviour. The following approaches have been studied. Some have been implemented in commercial FEM codes.

- A rheological model with springs and dashpots is often used to represent the visco-elastic ice behavior (Jordaan and Taylor, 2011).
- A model based on the continuum damage mechanics was developed for the brittle failure of isotropic ice (Kolari, 2007; Kolari *et al.*, 2009; Kuutti and Kolari, 2010).
- The ice is modelled as an elasto-plastic or foam material. The ice failure criterion is left to the user to define. Commercial FM codes support user-defined failure criteria.

Many papers have been published on modelling ice failure processes, with focus placed on: material non-linearities, friction and contact between ice and a structure (Sand, 2008), a multi-surface failure criterion (Wang and Derradji-Aouat, 2009), ice fracture and propagation (Liu *et al.*, 2011), and modelling of ice as a crushable foam (Gagnon, 2007, 2011).

9.2 Ice-Structure Interaction and Discrete Element Method (DEM)

Simulation of ice failure against offshore structures or ships also needs to be taken into account:

- Accumulation and clearing of broken ice
- Shape and stiffness of the structure

Often, it is not known in advance what ice failure modes will be dominant. Therefore, a range of ice models must be attempted before sensible conclusions can be drawn. Studies on ice-structure interactions include those by Gürtner (2009), Konuk *et al.* (2009), Kolari *et al.* (2009), Kuutti *et al.* (2010).

The discrete element method (DEM) has found extensive application in ice-structure interaction problems (Ji and Yue, 2011). The ice floes are modelled with spherical and cubic particles, and the ice cover can be modelled in one layer or two layers of these in regular or random packing. DEM has demonstrated its capability in qualitatively describing the mechanism of rotating and sliding of ice pieces, and seems to have high potential for estimating submerged components (Sawamura and Tachibana, 2011; Zhan *et al.*, 2010; Lau *et al.*, 2011; Kioka *et al.*, 2010; Paavilainen *et al.*, 2009, 2011). Simplified ice models are often favoured in studies on water-ice interaction during ice bending (Sawamura *et al.*, 2008), ship performance in level ice (Valanto, 2009), simulation of ship-ice interaction (Su *et al.*, 2010; Lubbard and Løset, 2011), level ice actions on moored ships (Aksnes, 2011).

9.3 Ice Ridges

The recent research on ice ridges is concentrated on:

- Ridge loads on structures
- Deformation, failure, and strength of a ridge

Various material models have been attempted, including a shear cap material model (Heinonen, 2009), Drucker-Prager model and the arbitrary Lagrangian-Eulerian (ALE) FEM for rubble failure against a conical structure (Ranta *et al.*, 2010).

The challenges are material parameters for ice. Punch-through tests have been used to measure the ridge and rubble strength both in full scale and in laboratories. Derivation of the material properties from the experimental data is not straightforward and usually requires assistance of numerical simulation (Serré, 2011a, 2011b; Polojärvi and Tuhkuri, 2009, 2010).

10 STRUCTURAL RELIABILITY ANALYSIS

One challenge to the Arctic development is the lack of experiences. Ship design has traditionally relied on operational experience for the development of design methods and design codes. In the absence of this experience, alternative methods are required. Structural reliability analysis (SRA) may have been a useful role to play in this regard.

SRA holds, in principal, the promise of more rationalized structural designs that achieve consistent safety levels. The reliability methods are attractive since they provide a framework to properly account for the uncertainty associated with the relevant design variables.

10.1 *Structural Reliability Approach (SRA)*

Several recent surveys of SRA literature provide good overview of the theoretical development and practical applications. ISSC had a Specialist Committee on “Reliability based structural design and code development” (ISSC, 2006). This ISSC committee work was performed at the time when the IACS was developing Common Structural Rules (CSR). A recent trend is to apply SRA to hull integrity management (Wang *et al.*, 2010). However, there is only limited coverage on SRA applications to ice-going ships and Arctic offshore structures.

A major challenge for practical application of the SRA is the proper selection of uncertainty models (Guedes Soares, 1988, 1997; Moan *et al.*, 2006; Wang *et al.*, 2010). The apparent disparities in SRA results presented by different research groups can be attributed to the differences in uncertainty modelling and the formulations of the limit state functions (Guedes Soares and Teixeira, 2000; Wang *et al.*, 2010; VanDerHorn and Wang, 2011).

10.2 *Probabilistic Ice Loads*

The ISO 19906 (2009) recommends a probabilistic approach that takes into account the high uncertainty of the ice geometric, kinetic, and mechanical characteristics, and various possible interaction scenarios in addition to those related to the ice, structure, soil, and mooring parameters.

As usual, the challenge is to determine the relative importance of these parameters and to concentrate on the significant interaction scenarios to increase the reliability of the calculated ice loads.

The ice loads are random in nature like other environmental loads. A large number of variables are needed to characterize ice failure phenomena and the resulting ice loads. This includes, among others, ice thickness, salinity, flexural strength, compressive strength. In addition, the ice loads on a ship also depend on the vessel's characteristics such as power, hull form of the entire vessel, and the location of interest. Virtually all ice load models are based on measurements in full- or model-scale tests. Short-term and long-term full-scale measurements have been made on ships travelling in the Polar regions, and these remain the most reliable sources of information.

Three limit mechanisms define the net imposed ice load on a structure (Wang *et al.*, 2011):

- **Limit strength:** An ice floe cannot sustain itself and crushes when the applied stress exceeds the material strength of ice. This strength corresponds to crushing and bending failures in the ice floe.

- Limit momentum: This is the load imposed by ice due to the floe moving with acceleration and impinging on a structure to impart its momentum as a load on the structure. The CSA code (CSA, 2004) indicates that the limit momentum can be neglected compared to the limit strength if the ice floe diameter is less than 5 km .
- Limit force: The ice load caused by the moving ice floe, where the movement is due to wind or current force, or due to movement of surrounding ice pack.

The ice load on the structure is limited by the force necessary to fail the ice feature and by the force driving the ice feature against the structure. In the absence of sufficient environmental driving force, the ice failure force cannot be generated. Therefore, the minimum value of the environmental driving force (limit force) and the ice failure force (limit strength) is taken as the critical ice load on the structure.

Figure 9 (Wang *et al.*, 2011) summarizes the methodology for calculating the annual maximum ice load on an Arctic offshore structure. For an arbitrary year, the number of ice floes that would interact with the offshore structure is first calculated. This number depends on parameters such as ice season length, ice concentration, floe velocity, floe size, and the structure geometry.

For each floe interacting with the structure, the two force components calculated include the limit strength and the maximum ridge force across all ridges in the floe. The limit strength is calculated based on ice floe size, wind velocity, ocean current velocity, and pack ice force. The maximum ridge force is calculated by finding the maximum of each individual ridge force on the floe. Each ridge force is calculated based

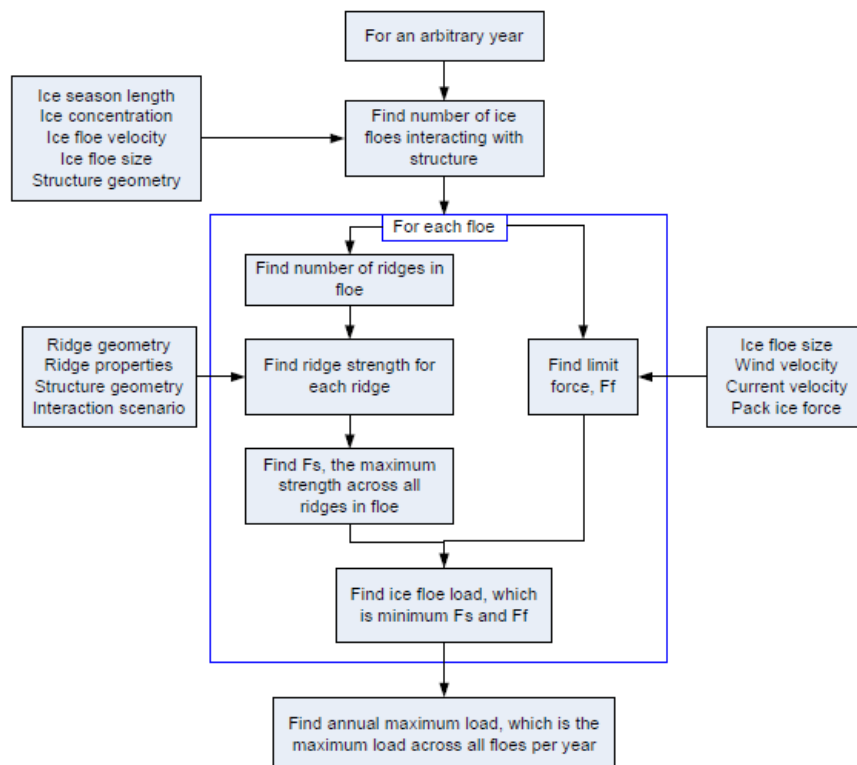


Figure 9: Derivation of probabilistic ice loads

Table 9: ISO 19906 maximum acceptable annual failure probability

Exposure Level	Maximum Acceptable Annual Failure Probability
L1 (high consequence/manned non-evacuated)	$1.0 \cdot 10^{-5}$
L2 (Medium consequence/manned evacuated or unmanned or Manned Evacuated with low consequence)	$1.0 \cdot 10^{-4}$
L3 (low consequence unmanned structures)	$1.0 \cdot 10^{-3}$

on ridge geometry and other ridge properties, structure geometry and the interaction scenario assumed between the ridge and the structure. The overall ice floe load on the structure is the smaller value of the limit strength and the maximum ridge force. Such floe loads are calculated for all floes during the year and from these the annual maximum floe load is calculated.

10.3 Implied Reliability Levels in Ice Class Rules

There are studies on the implied reliability level in the Finnish-Swedish Ice Class rules (Wang *et al.*, 2007). The plate thickness requirements of ice belt were investigated, and the influence of ship size and ice belt region was considered. The ice loads were assumed to follow type I extreme value distribution with a mean of 1.0 and a COV of 0.2, based on an existing statistical study on measurement data (Kujala, 1990). The calculated reliability indices were considerably lower than typical values for marine structures. The primary reason was the low level of applied ice loads in FSICR (see also section 5.2 of this Committee report). The acceptance criteria are correspondingly conservative when compared with the ultimate capacity of plate panels. As a result, the limit state for the FSICR thickness requirements has some features of a serviceability limit state.

If the FSICR is re-cast in an ultimate limit state, the ice loads need to be the extreme values and the resistance of the plate panels must represent the ultimate capacity. Assuming that the design ice loads have a 5 % probability of exceedance, the type I extreme value distribution would have a mean of 0.73 and a COV of 0.2. The resulting reliability level is significantly different.

The ISO DIS 19906 standard on Arctic Offshore Structures is a timely document that encompasses many aspects of Arctic and sub-Arctic Development. It employs the same principles of the other ISO standards such as ISO 19902 and ISO 19903 for fixed steel and concrete structures, respectively, and ISO 19904-1 for floaters. The ISO 19906 standard employs the Limit State design methodology that applies load and resistance factors to arrive at the target reliability levels as shown in Table 9.

11 SUMMARY AND RECOMMENDATIONS

The Committee strongly recommends that ISSC continues this committee. The revived demand for Arctic shipping and Arctic development will continue driving research and development of Arctic technologies.

11.1 Ice Class Rules

The ice class rules are the corner stone of ship design. The Committee attempted to survey literature that supports the development of ice class rules, and we realized that our coverage is rather limited.

The Committee noted that various differences exist in technical basis between ice class rules. This may offer opportunities of future research on, but not limited to:

- Concept of ice class rules (limit states, target failure probability)
- Definition of ice belt
- Ship-ice collision scenario
- Ice load (probabilistic feature, extent of ice patch, pressure versus area relationship)
- Structural analysis models for the response of plate and stiffener/frame – elastic versus plastic methods, including application of linear and non-linear FEM
- Materials for Arctic application
- Corrosion/abrasion

For Arctic structures, the following topics may need to be improved:

- Definition of operating parameters for each “class” of ice strengthening
- Evaluation of feasibility of applying ship design practice to Arctic structure

11.2 Tests, Analysis

Numerical and analytical tools will continue to be extensively applied in explaining ice behaviour, ice failure mechanisms, ice-structure interaction and the resulting ice loads. The Committee believes that there is room for developing and improving these tools, and comparisons with field measurement and model tests will be important.

Techniques of numerical simulation are advancing rapidly, but face a major difficulty in verification due to lack of data. Traditional ice models need information on compressive and bending strengths, but the more advanced models need more data about ice properties (i.e., shear strength, particle-particle bonding strength within a ridge). This will in turn lead to needs for additional tests and sharing of test results.

11.3 Structural Reliability

Structural reliability approaches deserve more research and development attention. SRA adds values to the understanding of the ice mechanics, and may potentially lead to refinement of design rules that are mostly based on limited experiences.

A very important task of SRA is probabilistic modelling of ice loads. Additional studies of this topic are needed.

11.4 Risks of Arctic Shipping and Arctic Development

Arctic shipping and Arctic development face a variety of risks (Tikka *et al.*, 2007). Existing ice class rules have focused on vessel performance and responses of hull and machinery. These rules only provide a minimum set of requirements that must be supplemented by more comprehensive considerations of a wider range of topics, including but not limited to:

- Propulsions
- Winterization of vessels and equipment
- Ice management
- Ergonomics
- Crew training
- Ice forecasting, ice management
- Oil spill behaviour and recovery

The Committee encourages increased applications of risk assessment in all these areas.

12 ABBREVIATION

CASPPR	Canadian Arctic Shipping Pollution Prevention Regulations
ISO	International Standard Organization
FPSO	Floating Production Storage Unit
FPU	Floating Production Unit
FSICR	Finnish-Swedish Ice Class Rules
IAHR	International Association of Hydraulic Research, Ice Symposium
IACS PC	IACS Polar Class Rules
NSR	Northern Sea Route
POAC	Port and Ocean Engineering under Arctic Condition
RSMS	Russian Society of Maritime Register of Shipping
SRA	Structural Reliability Approach

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18th INTERNATIONAL SHIP AND
OFFSHORE STRUCTURES CONGRESS
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VOLUME 2



COMMITTEE V.7
**IMPULSE PRESSURE LOADING AND
RESPONSE ASSESSMENT**

COMMITTEE MANDATE

Concern for direct calculation procedures for evaluating impulsive pressure loadings, namely slamming, sloshing, green water and underwater explosion, and their structural response. The procedures shall be assessed by a comparison of tests, service experience along with the requirements of the rules for relevant classification societies. Recommendations for structural design guidance against impulsive pressure loadings shall be given.

COMMITTEE MEMBERS

Chairman: Sang-Rai Cho
Muniyandy Elangovan
Allen Engle
Taebum Ha
Jørgen J. Jensen
Geert Kapsenberg
Sime Malenica
Jerome Matagne
Huiling Ren
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Svein Saevik
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Takao Yoshikawa

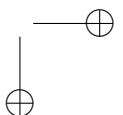
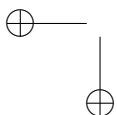
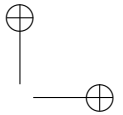
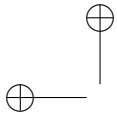
KEYWORDS

Slamming, sloshing, green water, underwater explosion, impulsive pressure, structural damage, classification society rules, natural period, peak pressure, impulse duration, equivalent static pressure

18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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1 INTRODUCTION

Recently assessments of impulsive pressure loading and response of ship and offshore structures have drawn more attentions of marine structural designers. In order to reflect the concerns regarding the pressure impact and structural response a new specialist committee was formed in the last ISSC. The committee report covered impulsive pressure loads and responses together, which were previously treated separate in various technical and specialist committees. The overall frame of this report is the same as that of the last ISSC report (Cho *et al.*, 2009).

The effect of impulsive pressure loading on ship structural response can be both global and local. Global impulsive loading makes whole ship structure vibrate while the local impulsive loading affects relatively small part of the structure at the fluid structure interface. The way in which the impulsive pressure loading will influence the structural response depends both on the amplitude of the pressure and on its evolution in space and time. In the analysis of the structural response, it is thus fundamental to consider both parameters, i.e. the pressure amplitude and space/time evolution, together with the structural response because the degree of interaction will depend both on the type of loading and on the structural characteristics. Indeed, the extreme pressure peaks (up to 50 or even 100 bars) do not necessarily means that the structure will encounter any damage because these pressure peaks are usually associated with very short duration in time and very small extent in space. That is why, when analysing impulsive pressure loading, one must always have structural response in mind. In addition to the importance of the spatial pressure distribution, one important parameter is the ratio in between the time scale of the loading relative to the structural natural periods contributing significantly to large structural stresses. When the loading occurs at the time scale of these periods, fluid-structure interactions must be considered which means that the fluid flow must be solved simultaneously with the dynamic structural response. All these comments are valid both for the global and local impulsive loading effects.

The committee reviewed many of recently published papers and reports related with impulsive pressure loadings and responses of marine structures. Comparisons of classification societies rules were also conducted and results were summarised in this report. Even though the progresses in predicting pressure impact loads and structural responses have advanced, the outcomes have not been reflected in the relevant classification societies rules. In hoping to improve those rules some recommendations are provided for structural design guidance.

2 LOCAL SLAMMING

2.1 General

With slamming is generally meant to be the impact between a structure and a body of fluid. Such impact creates rapid changes of fluid velocities and corresponding changes of hydrodynamic momentum. The related hydrodynamic loads increase with increasing rate of change of hydrodynamic momentum, which in turn increase with increasing relative velocity and decreasing relative angle between the body and the fluid. When the flat bottom in the fore part of a low speed ship is being lifted out of the water due to large relative ship motions, rather moderate relative velocities between the re-entering hull and the wave surface is needed for large slamming loads to develop. Higher ship speed implies larger relative motions and velocities, and here large slamming loads can develop despite the hull being flared or deadrised. Slamming might also occur when a

wave hits a stationary structure such as a platform deck or column. Slamming is characterized by large free-surface deformations together with spray jet formations in the intersection between the body and the water surface. The phenomenon is accompanied with related large pressure gradients, rapidly propagating peaked pressure distributions, complex flow separation and possibly air entrapment at small relative angles. For flexible structures the situation might be complicated further due to structure deformation related local changes of the relative velocity and geometry, and combined structural and hydrodynamic inertia effects. To this also adds the random nature of waves and ship motions. All in all this makes the prediction of slamming loads and related hull structural strength assessment a real challenge which still is far from fully mastered. Consequences of limitations in predictive capabilities in structural design might be structural damage or overly conservative and heavy structures. This chapter reviews research performed in the last three years in the area of local slamming, i.e. slamming loads and related fluid-structure interaction and responses for hull panels and other local structure. The chapter is divided into the four different problem areas: 1) *Fundamental hull-water impact*, involving studies of rigid simply shaped bodies impacting calm water and related hydrodynamic loads; 2) *Hydroelastic interaction*, involving studies of water impact of flexible structures and related hydrodynamic loads and structural responses; 3) *Wave impact*, involving waves impacting on stationary structures; and 4) *Concurrent modelling of waves, ship motions, slamming loads and structural responses*.

2.2 Fundamental Hull-Water Impact

Experiment is of course a very important source of knowledge, both for understanding the mechanisms involved and for gathering reference data for evaluation of theoretical models. However, due to the large complexities involved even in simplified fundamental hull-water impact situations, the experimental setup is far from trivial and only a few experimental series such as those by Aarsnes (1996) are available in the literature. Some new significant experimental work on fundamental hull-water impact has however been performed lately. Lewis *et al.* (2010) for example provide a comprehensive report on an experimental programme involving upright impact of a 25 degree dead-rise angle rigid wedge studying different wedge masses and drop heights. Pressures, vertical accelerations, position sensing and high speed camera images were obtained during the experiments. The comprehensive set of measurements, the detailed description of the experiments and equipment, and a detailed uncertainty analysis makes these data highly suitable for validation of predictions. Tveitnes *et al.* (2008) designed an experimental setup to enable near constant velocity impacts entries and exits with wedges with end plates. Forces, velocities, wetting factors, and derived added masses are presented and evaluated. The results are particularly valuable in simulation of planing in calm water based on the planing-immersing section analogy. Also Battley *et al.* (2009) carried out experimental investigations with near constant velocity panel-water impacts as further commented in the hydroelastic interaction section below. Huera-Huarte *et al.* (2011) designed a novel test rig in order to experimentally study high-speed panel-water impacts. Impact force and velocity are measured and high speed imaging is used. High velocity impacts up to 5 m/s were conducted at angles between 0.3 and 25 degrees with a practically rigid panel. Good correlation with the experiments by Tveitnes *et al.* (2008) and asymptotic theory are shown for impact angles larger than 5 degrees. Cushioning at smaller angles is demonstrated and discussed. De Backer *et al.* (2009) have performed experiments on water impact of different axisymmetric bodies. Wetting factors, pressures, impact velocities and ac-

celerations are presented. Measured pressures are compared with a three-dimensional asymptotic theory for axisymmetric rigid bodies which are found to significantly over-predicted the pressures. Possible reasons for the discrepancies between experiments and theory are discussed.

Not many studies considering analytical methods have been done lately. One of the few is by Yoon and Semenov (2009) presenting a semi-analytical method for modelling oblique wedge-water impact. The method is used to study the onset of flow separation from the wedge vertex as a function of wedge orientation and direction of impact velocity. Very good agreement with experimental data is demonstrated. Tassin *et al.* (2010) present a detailed review of several different analytical methods for the prediction of the hydrodynamic impact forces and pressure distributions acting on two-dimensional and axisymmetric bodies entering calm water. The studied methods reviewed include the original and a generalized Wagner method, the modified Logvinovitch method and the matched asymptotic expansion method. Results from the reviewed methods are compared with results from explicit finite elements arbitrary Lagrangian-Eulerian (ALE) simulations and experimental observations. The different methods are shown to agree well with small deadrise angles but differ significantly for larger angles.

Encouraged by the increasing computer power, fundamental hull-water impact has lately been studied using computationally intensive techniques including various RANS methods, Smooth Particle Hydrodynamics methods (SPH), Moving Particle Semi-implicit method (MPS), finite-element arbitrary Lagrangian-Eulerian methods (ALE), and Boundary Element Methods (BEM). Viviani *et al.* (2009) review a Smoothed Particle Hydrodynamics Method (SPH) that is under development and compare simulation results for drop tests with two-dimensional sections with corresponding results from a Reynolds-averaged Navier-Stokes solver (RANS) and experiments. Both numerical methods seem to be capable of capturing the physics of the slamming phenomenon, showing an overall satisfactory agreement with experimental results in terms of local pressure and total forces. Special considerations were made in the development of the SPH method to ensure its generality of application. However, it is concluded that further investigations are needed for example considering pressure instabilities and drift. Veen and Gourlay (2011) use a 2D Smoothed Particle Hydrodynamics method (SPH) to study slamming impacts of hull sections. Excellent agreement with experimental data is shown for a wedge shaped section regarding vertical velocity, force and pressure, and fairly good agreement for a flared hull section. The importance of proper modelling of the impact velocity profile is demonstrated and discussed. Khayyer and Gotoh (2010) highlight various challenges related to particle methods such as the Moving Particle Semi-implicit method (MPS) and the Smooth Particle Hydrodynamics method (SPH), for example regarding conservation of momentum and energy, interpolation completeness, non-physical pressure fluctuations, criteria for assessment of the free-surface, and provide references to a large number of efforts on improving these methods.

Fairlie-Clarke and Tveitnes (2008) use the finite volume of fluid method implemented in the CFD code Fluent 4 to study constant velocity wedge impacts on calm water. Pressures, forces, and free surface profiles are presented and possible modifications of the slamming momentum theory are discussed. Yang and Qiu (2010a, b) extend earlier developed Constrained Interpolation Profile methods (CIP) from 2D to 3D. In the CIP method the fluid-structure interaction is treated as a multiphase problem, which should make it suitable for modelling slamming problems with large free-surface deformations. The method is validated for a 3D wedge and a sphere showing good

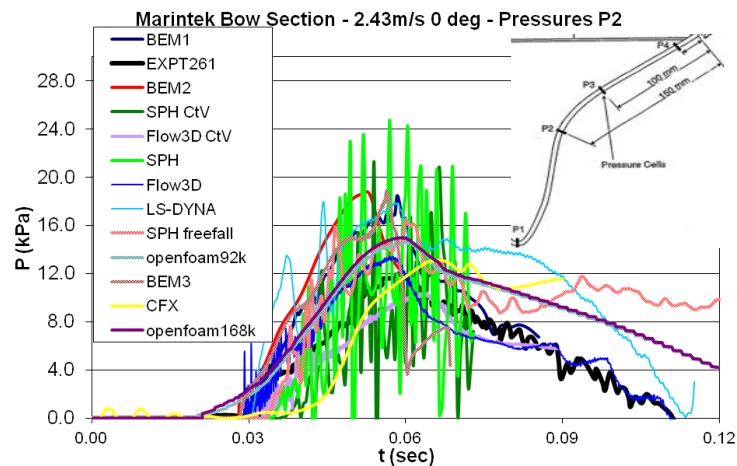


Figure 1: Predicted and measured pressures at point P2 of the bow section (Temarel, 2009)

agreement between simulations and experiments. Yang and Qiu (2010b) applied both 2D and 3D CIP methods on a 3D planing hull shape entering calm water with different roll and pitch angles. Slamming forces for the 2D method are generally larger than those by the 3D method. Experimental validation of the planing hull simulations are said to follow. Sun and Faltinsen (2009) studied two-dimensional water entry of a bow-flare ship section using an improved boundary element method where the fully nonlinear free surface conditions and exact body boundary conditions are satisfied and flow separation from knuckles is considered. Simulations are compared with previously published experimental results for upright as well as heeled sections. Fairly good agreement is found between simulations and experiments but the existence of experimental bias errors is obvious. Special effects related to the evolution of the free surfaces flow separation for heeled sections are demonstrated and discussed.

Brizzolara *et al.* (2008) provide extensive comparison between various numerical methods and experiments in the modelling of pressures and forces on a rigid bow section impacting a calm water surface. Both upright and heeled conditions are studied at various impact speeds. The numerical methods include three different Boundary Element Methods (simplified, i.e. not fully nonlinear), various commercial RANS software such as FLOW-3D, FLUENT (limited results), ANSYS-CFX (limited results), and LS-DYNA, OpenFOAM, and a Smoothed Particle Hydrodynamics (SPH) approach. There is reasonable overall agreement between predicted and measured pressures but the scatter is large as seen in Figure 1. The agreement is better for the lower impact speeds. All BEMs appear to overestimate the pressures. RANS type approaches FLOW-3D, as well as OpenFOAM and SPH appear to result in the best predictions, although they may suffer from large oscillation, especially the SPH. For the RANS methods, in general, the most stable results were obtained using free fall, rather than constant speed or simulating the velocity profile obtained in the experiments. The slamming forces show the same trends as the pressures. The complete set of calculations is reported by Temarel (2009).

2.3 Hydroelastic Interaction

The influence of hydroelasticity was investigated through analytical/numerical methods and a few experiments mainly, though not exclusively, focussing on V-type sections

prevalent in high speed craft. From these investigations it is apparent that the fluid-structure interactions involved are complex and that there is urgent need for more and systematic experiments for validating the numerical predictions.

Khabakhpasheva (2009) investigated the 2D coupled FSI problem of an elastic cylindrical shell penetrating at constant velocity in a thin layer of ideal fluid. Initially the shell is in contact with the liquid surface at a single point. The normal mode of approach is used for the coupled hydroelastic problem and the flow region comprises four subdomains with a solution obtained through matching. Results are presented for shells made of steel, aluminium and glass fibre plastic, for a range of thicknesses and impact velocities. Strain evolution with time, obtained at the bottom centre of the steel shell, show reasonably good agreement with numerical predictions in deep water and experimental results in shallow water. The study concludes that stress and deformation of the shell increases as the thickness of the liquid layer decreases. This study can be of practical interest, for example for bottom slamming of bulbous bows.

Mutsuda and Doi (2009) combined the Constrained Interpolation Profile (CIP) with the SPH method, the former for the fluid particles and the latter for the particles modelling the elastic structure. Examples are provided for a range of 2D impact problems, such as wedge with a range of deadrise angles, an elastic (aluminium) cylindrical shell, an elastic bow flared section impacting at 45 degrees and wave breaking on an elastic (steel) vertical wall. For the aluminium shell the strain variation with time at bottom centre is close to experimental measurements in deep water. The authors plan to improve their numerical model by accounting for air compressibility. Luo et al (2010) used the explicit FE code MS Dytran for the impact of stiffened panels. The code uses FE and finite volume methods to model structure and fluid, respectively, with an ALE algorithm for the fluid-structure coupling. Numerical results for one case of a stiffened steel panel fall within the scatter of experimental measurements of maximum pressure vs. maximum acceleration. Strain measurements were not available from these tests. The authors carried out a detailed numerical simulation for another steel stiffened panel, including various drop speeds and air cushion effects. These results show important differences between rigid and elastic impacts for pressure peak values and time histories. Their numerical results also show significant increase in peak pressures when the air cushion effect is neglected, but the predicted stress results do not increase as much. Oger *et al.* (2010) used the SPH method for fluid-structure coupling to simulate a range of impact problems, including 2D modelling of an elastic wedge (deadrise angle 10 degrees) impacting still water. Predicted pressures and deformations are compared with a semi-analytical solution showing good correlation, provided the tensile instability in the couple SPH method is removed using the artificial tensor procedure.

Maki *et al.* (2011) carried out 2D numerical investigations of an elastic wedge impact using one-way coupling, namely CFD (OpenFOAM) analysis of a rigid wedge and transfer of relevant information to a structural model which uses modal analysis for the wet wedge. The predicted deflections were compared to fully coupled theoretical and numerical models. The authors conclude that their method has poor time accuracy in the impact stage but shows good agreement for the maximum deflection, the latter indicating that the approximation used for the flexural added mass is acceptable. Stenius *et al.* (2011) analyse flexible panel-water impacts for a range of different panel properties, panel boundary conditions and impact scenarios using the explicit arbitrary Lagrange-Euler finite element method implemented in the code LS-DYNA and a simplified method combining beam and potential flow theories. Hydroelastic effects

are quantified by comparing with a rigid quasi-static solution where the hydrodynamic loading is modelled as unaffected by the structural deformation and the structural response is modelled as unaffected by structural inertia. The authors concluded that both hydroelastic inertia and kinematic effects can be important. Furthermore they emphasize the significance of impact scenario (or envelope) on the hydroelastic effects increasing or reducing pressures and panel response. Campbell *et al.* (2010) used a coupled FE-SPH approach to model the nonlinear FSI behaviour where the structure also experiences large nonlinear deformations. The explicit FE software DYNA3D was selected. The method is verified by simulating the dam break problem. It is subsequently applied to simulate the impact of a detailed helicopter sub-floor structure with water. The predictions are compared with drop test experiments. The acceleration presented shows reasonable qualitative agreement, due to differences between simulated and experimental conditions. However, the overall predicted deformation of the structure is claimed to be consistent with deformations observed in the experiment, such as joint failure and plastic deformation.

Battley *et al.* (2009) carried out experimental investigations on panels made of three different composite materials, one effectively rigid, for various impact speeds, deadrise angles and boundary conditions. The authors concluded that hydroelastic inertia effects have a significant influence on panel response. They also identify kinematic effects, such as large reduction in local deadrise angle at the chine, resulting in much higher peak pressures than for rigid panels. Kong *et al.* (2010) present experiments and numerical simulations of drop tests with a flexible section of a trimaran hull including side hulls, cross structure and internal structure. The numerical simulations are performed with the Autodyn FE solver using an Eulerian-Lagrangian method where the coupling between the fluid and the flexible structure is considered. Remarkably good correlation between measured and predicted peak pressures is presented. The validity of these observations is, however, somewhat difficult to judge, for example due to the limited information provided regarding pressure transducer diameters.

Qin and Batra (2009) use sandwich composite panel theory (including transverse shear and transverse normal deflection for the core and Kirchhoff plate theory for the skin) and 2D potential flow analysis, excluding separation effects. The majority of their calculations only partially account for the hydroelastic effects, excluding added inertia, having shown that such a model is adequate. Their study, however, only considers the initial phase of slamming until just before the water separates at the upper panel boundary. Das and Batra (2011) use explicit arbitrary Lagrange-Euler finite element method implemented in LS-DYNA to make detailed studies of slamming of sandwich panels including quantification of strain energy densities in core and laminates and effects of delamination. Another interesting application for sandwich panels is presented by den Besten and Huijsmans (2009). The structure is modelled using Euler beam theory for the skins and a linear orthotropic continuum for the core, with relevant compatibility conditions. The hydrodynamic impact force is modelled based on Wagner's method. The forced vibration response, subject to the impact load, is evaluated using modal summation. Damping is included using complex core moduli. The numerical results for a stiffened aluminium and a flat sandwich panel show that the bending stresses are larger for the former. There are no experimental measurements to compare the predictions.

2.4 Wave Impact

The occurrence of wave impacts is a critical feature in the design and re-assessment of many offshore structures. With evidence of increasing storm severity and with

subsidence an important characteristics of some mature fields, the quantification of impact loads arising on both the columns and the underside of the lower deck of large offshore platform remains a difficult but important issue.

Roos *et al.* (2009, 2010) presented the experimental data arising from a physical model study of Gravity Based Structure subject to a severe sea state. They showed that far from being a highly localized effect, involving a thin sheet of water, the run-up associated with a steep wave can involve significant volumes of water, travelling at very high velocities, leading to occurrence of large impact pressure acting over substantial areas. They also showed that the largest loads frequently did not correspond to the tallest or steepest incident waves. They showed the importance of wave-structure and wave-wave interaction effects and the need to undertake long random wave tests in offshore engineering design. Baarholm (2009) performed a small scale model test campaign of wave impact on an idealized platform deck and clarified that the three-dimensional effects significantly reduced the wave-in-deck loads, in particular, for the water exit phase, the vertical force is almost halved due to three-dimensional effects. They showed that the Wagner based method with three-dimensional correction yields good results for the water entry phase, but it overestimates the water exit force and underestimates the duration of the wave-in-deck events. Kendon *et al.* (2010) compared the measured vertical load on the deck against simple potential theory and the result from CFD code STAR-CCM+. They concluded that for isolated impact events the simple potential flow based model is adequate for predicting the vertical loading on the deck. However, if there is a strong likelihood of steep wave grouping resulting in closely following wave-in-deck impact events, the aforementioned simple method may be non-conservative, and a CFD analysis or model test may be advisable to predict loading for such case.

Clauss *et al.* (2010) analysed stochastically the data from model tests with the Sleipner A GBS for estimating the impact pressure due to braking waves corresponding to an annual probability of 10^{-4} . The procedure for calculating shock pressure due to breaking waves recommended by DNV was also applied. The two calculation approaches resulted in significantly different estimates for the characteristic 10^{-4} probability impact pressure and the procedure recommended by DNV seemed to be strongly underestimating the impact pressure. They suggested that the different force sensor sizes had an influence on the resulting characteristic 10^{-4} probability pressures. They concluded that the recommendation of DNV had to be altered to ensure a reliable prediction of the characteristic impact loads if the difference was still present by using the adequate force sensor.

Iwanowski *et al.* (2010) studied numerically a wave-in-deck load due to an extreme wave, acting on a jacket platform. Firstly, they calculated the fluid pressure acting on the platform using CFD code ComFLOW, in which VOF method was used to calculate the behaviour of fluid free-surface. Subsequently, the pressure was mapped automatically onto structural FEM shell elements and the structural response was calculated using LS-DYNA. Liang *et al.* (2010) calculated air gap response of a moored semi-submersible adopting a Navier-Stokes solver by VOF method. To confirm the accuracy of the numerical solver, the predetermined irregular wave train was simulated and verified against physical tank results. Xu *et al.* (2008a, b) studied the steep wave impact pressures and the structural dynamic response of floating production storage and FPSO bows using 1:80 scale segmented, instrumented models. They developed a time history simulation method, which makes use of a simple modification to linear random wave theory and a relatively simple slap force prediction based on

velocity times rate of change of added mass in order to calculate bow loading in random sea. Comparisons are made between experimental and calculated impacts and associated pressures. Simplified design rules for curved bows were proposed. Ten and Korobkin (2009b) used potential flow analysis for modelling steep wave impact on an elastic vertical wall. The fluid domain is compressible in the vicinity of the wall and incompressible elsewhere. The wall is modelled using Kirchhoff's plate theory. The equations of motion of the FSI system are in terms of the principal coordinates of the elastic plate and fluid loads. The method has been verified in terms of convergence analysis. It was also applied to two boxlike structures with different impacting wall thicknesses. A sensitivity analysis of modelling compressibility is also carried out.

2.5 Concurrent Modelling of Waves, Ship Motions, Slamming Loads and Structural Responses

Since the pioneering work by von Karman (1929) and Wagner (1932) until present days the major research effort on local slamming has been on the idealised situation of a two-dimensional body impacting a calm water surface at constant speed or free-falling. However, in order to assess the slamming pressure and the related consequences for the hull structure in a design situation the slamming calculations must be combined with modelling of the ship motions in waves. As discussed in Hermundstad and Moan (2009) two main approaches can be distinguished, the "k-factor methods" and the "direct methods. In the "k-factor methods" the slamming loads are determined by scaling slamming coefficients (so-called k-factors), which have been pre-determined based on calculations or experiments, with a statistical measure of the square of the impact velocity, which typically is determined based on linear strip theory and linear response analysis (e.g. Ochi and Motter, 1973). The "direct methods" involve more thorough modelling of the ship motions in waves including the non-linear slamming mechanisms. Direct methods obviously have the potential to be significantly more accurate than the k-factor methods. Due to the high complexity of the slamming problem and the randomness of the waves direct methods however require significantly more computational effort. Kaspenberg and Thornhill (2010) for example reports that a 60 second CFD simulation (ANSYS CFX 11.0) of a captive model at forward speed in irregular waves took 7 days using 40 1.6 GHz processors. The development of direct methods that are feasible for design purposes will hence require special concerns to limit the computational effort. A few attempts of developing such approaches are reviewed in the following.

Lin *et al.* (2009) developed a nonlinear hybrid numerical method for predicting wet deck slamming of high-speed catamarans. In this method, the fluid domain is divided into an inner domain that encloses the ship and its nearby flow field and an outer domain that extends from the near- to the far-field flow. The flow in the inner domain is modelled with viscous flow theory while the flow in the outer domain is described with potential flow theory. An overlapped matching zone is employed to couple the two flow solutions. Simulations with a wave maker, a sphere impact on a flat water surface, and the wet deck slamming of a high speed catamaran are demonstrated but quantitative evaluation is limited.

Hermundstad and Moan (2009) use a non-linear strip method to simulate ship motions and determine slamming loads based on the simplified 2D boundary element method in Zhao *et al.* (1996). To speed up the calculations a pre-calculation and scaling approach is used and slamming forces are only calculated for wave encounters for which slamming conditions have been detected with a simpler method. Simulated

pressure time series are compared with measurements from two different experiments with 2D sections impacting calm water and from two different model experiments with ships in waves. The agreement is reasonable but the pressure magnitudes differ considerably in several cases. Evaluation based on statistical measures would be a good complement. Kaspenberg and Thornhill (2010) determine slamming loads based on momentum theory with particular concern for slamming related pile-up and pile-up due to the static bow wave. Similarly as in approaches for simulation of planing craft in waves (e.g. Garne and Rosén, 2003) spacial derivatives of added masses are pre-calculated and then scaled with the momentary position and velocity of the ship in a seaway. Hereby the computational effort can be limited making long term simulation of slamming loads and derivation of statistical properties for design purposes feasible. The idea is that this simplified method should be tuned with more advanced CFD calculations for improved accuracy and then integrated in a ship motion simulation scheme. The method is compared with experiments and CFD calculations for a captive model in a seaway showing reasonable agreement.

3 GLOBAL SLAMMING

3.1 General

With the increasing demand of large-scale and speed in shipping industry, the wave impact of large container ship, cruiser and multi-hull boat is becoming more and more important. Study on wave impact is mainly to make right Class Rules which can guide structural design of ships according to reasonable and reliable impact loads, thus, we need a practical direct calculation method about impact response.

The first construction on wave impact was by von Karman (1929). Now, people have achieved fruitful results on the impact problem of two-dimensional structures, reliable results were given in numerical method (e.g. Zhao and Faltinsen, 1993, Zhao *et al.*, 1996) and laboratory experiments (e.g. Chuang, 1967, 1970). Reliable results have not been obtained for 3D hull impact in numerical method (Xu, 2010); Many researchers had conducted ship model test, but the scale effect must be considered, and the impact responses of hull in model test need to be validated by the result of in-service experiments.

Sailing ship's wave impacts are complex and dynamic physical phenomenon, it is a very difficult task to establish a comprehensive physical model and mathematical model. This model should include geometric nonlinear, wave-surface nonlinear, nonlinear motion and 3D effects. It is not practical that all of the nonlinear and 3D effects are included in the model. We can predict the impact response of ships by establishing a simple model which takes part of nonlinear and 3D-effects into considerations, then continue to carry out the improvement and revision.

A methodology for investigation of this challenging phenomenon is drawn up and a mathematical model is worked out. It includes the definition of ship geometry, mass parameters, structure stiffness, and combines ship hydrostatics, hydrodynamics, wave load, ship motion and vibrations. The modal superposition method is employed. Based on the presented theory, a computer program is developed and applied for hydroelastic analysis of a large container ship (Senjanovic *et al.*, 2009).

In general, the global slamming response needs to be combined with the simultaneously obtained global and local steady state load effects, in terms of extreme values for ultimate limit state checks and cyclic load histories for fatigue design checks. Vessel speed and possible heavy weather avoidance are also important factors and the

operational profile should be properly defined when determining design load effects. Moreover, it was noted that even if slamming loads initially induce large sagging loads, they would also imply large hogging loads due to the transient dynamic character of the response (Moan *et al.*, 2006). This is important since the hogging condition may be the governing design condition, e. g. for container vessels.

The need to augment existing design rules with a rigorous means to identify design wave conditions is discussed by Kim and Troesch (2010). Rather than using Monte Carlo methods for determining the effects of combined wave plus slam induced whipping loads, a design load generator analysis process is used. Basically with this approach, the phase distributions that lead to the m^{th} maxima at a prescribed time are determined. From there, an ensemble of short time series that will return target extreme events at a present time is created. The analyst can then use these time series ensembles to predict lifetime maximum loads at prescribed target extreme values.

3.2 Laboratory Experiments

For laboratory experiments, a scaled ship model is needed which directly brings the scale effect, methods to extrapolate the results of models to full scale are not yet developed (Hirdaris and Temarel, 2009), but in-service experiments are much more expensive. The experimental program consists of tests in both regular and irregular head waves, and the measured quantities included wave elevation, vertical motions and hull pressures (Tiao, 2011).

Slam events experienced by high-speed catamarans in irregular waves were characterized through experiments using a hydroelastic segmented model. The model was designed to represent the dynamic behavior of the full-scale 112 meter vessel and to allow the measurement of the slam load on the bow and wet deck (Thomas *et al.*, 2011). In order to measure vertical moment, the ship model must use segmented model in the experiments. This also accords with the actual condition of ships. The real ship hull will vibrate in waves.

Hydroelastic segmented model tests have been undertaken in head-seas to investigate the parameters affecting the whipping vibratory response of high-speed catamaran vessels subject to slamming. The first longitudinal modal frequency measured on full-scale INCAT catamaran vessels is used as a basis for predicting the flexural response frequency of the hydroelastic segmented model (Lavroff *et al.*, 2010).

3.3 Hydroelastic Analysis

The classical approach to determine ship motions and wave loads is based on the assumption that the ship hull acts as a rigid body. The wave load is then imposed to the elastic 3D FEM model of ship structure in order to analyse global longitudinal and transverse strengths, as well as local strength with stress concentrations related to fatigue analysis. Large ships are relatively more flexible and their structural natural frequencies can fall into the range of the encounter frequencies in an ordinary sea spectrum. So, a reliable approach to determine ship motions and wave loads requires analysis of wave load and ship vibrations (springing and whipping) as a coupled hydroelastic problem (Senjanovic *et al.*, 2009). Ship motions and wave loads can be analysed by 3D nonlinear hydroelasticity theory which takes wave impact into consideration.

The study of hydroelasticity of ships first gained momentum in the late 1970s with the work of Bishop and Price, who established the 2D hydroelasticity theory of ships (Wu and Cui, 2009). Then the theory of hydroelasticity was extended to 3D for ships

with a forward speed in the middle of 1980s (Wu, 1984; Price and Wu, 1985; Bishop *et al.*, 1986).

In recent years, the research on nonlinear wave load calculation method has made some progress. Many methods have been introduced, including first-order theory, second-order theory and the body nonlinear theory. The nonlinear factors include the speed square of pressure expression, wet surface and free surface. Through a large number of studies it is shown that the dynamic nonlinearity is mainly due to the body nonlinearity together with the free surface nonlinearity.

A 3D nonlinear time domain simulation hydroelasticity analysis method of ship motions and wave loads is presented, taking body nonlinearity and hull impact into consideration. The hull girder is simplified as a Timoshenko Beam. Combined with the wet surface generation method, the velocity potential of flow field is solved by the source-sink distribution method. After all the wave forces acting on the hull girder are obtained, the forced vibration equation of hull girder is established. Then, the principal coordinates of each order vibration and the section loads of ship are obtained (Li, 2009).

It is important to determine the impact force, the existing calculation methods include numerical method, laboratory experiment method, in-service experiment method and empirical formula method. The results of theoretical calculation can be checked with experiment values. The accuracy of numerical method for the 3D hull cannot be trusted.

The empirical formula method is based on the Wagner wedge impact theory, the Chuang cone impact theory, and experiments performed at the David Taylor Naval Ship Research and Development Center. Determination of the impact pressure is based on the hypothesis that the impact velocity is equal to the relative velocity normal to the impact surface of the moving body and the wave surface. The proposed method has been verified by several model tests in waves and by actual ship trials of the catamaran USNS Hayes (Stavovy and Chuang, 1976).

Calculation and prediction of ship slamming pressures in severe seas is difficult for the sea conditions and 3D characteristics of ship hull shape must be totally considered,

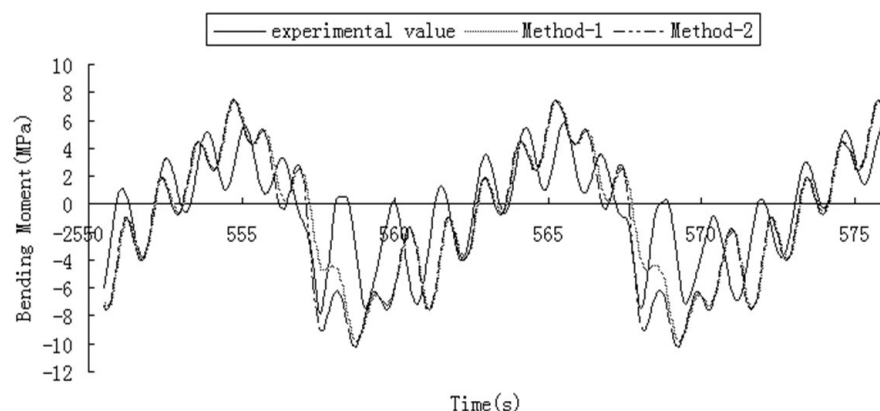


Figure 2: Experimental value and calculation values in different method of determine the impact force. Method-1 is empirical formula method. Method-2 is momentum method. Ship speed 9 knots ($\lambda/L = 0.9$) in head waves, wave amplitude 8.75 meter.

the influence of ship motions such as heaving, pitching and rolling should be included. The slamming pressure coefficient is then a key factor to be determined (Wang *et al.*, 2010). The direct calculation of ship slamming pressure based on empirical formula is reasonable and reliable. The 3D effect of hull shape and wave surface and the influence of ship motions are considered. Impact force can be obtained by integrating the slamming pressure along hull wet surface.

Another calculation method assumes that impact force is related to the change rate of fluid momentum and buoyancy. If the transient added mass and transient area of subsidence of profile sections are known, then the impact force can be obtained. In Figure 2, there is an impact response calculation case of one Container Ship. The profile bending moment in experiment and theoretical calculation were compared. The impact force was calculated by different methods.

4 SLOSHING

4.1 General

This chapter is devoted to the evaluation of the dynamic structural response of the cargo containment system (CCS) inside the membrane type LNG tanks of different floating units (ships, FPSO's ...). Sloshing loads represent dominant part of the design loads. These sloshing design loads are relevant both for the ship hull structure and for the cargo containment system. As far as the hull structure is concerned the situation is slightly simpler and normally only global loads matter. Concerning the cargo containment system the situation is significantly more complex because CCS is directly exposed to the violent sloshing impact loading. There exist today two main types of CCS and they are shown in Figure 3. Both systems are owned by Gaztransport and Technigaz (GTT), and both systems are structurally very complex and involve different types of materials (plywood, perlite, invar, stainless steel, foam, glue,...) which are connected together and attached to the hull structure.

Because there are no numerical methods that can fully describe the sloshing induced slamming pressures, one has still to rely on experiments which means in practice model tests. The challenges are how to scale the model test results to full scale and properly account for the structural elastic reactions due to the fact that a rigid model is used in model scale. There are many contributing factors to scaling which have to be considered and one has to do certain approximations. Generally speaking, Froude scaling is expected to be a dominant effect. Correct ratio between the density of the gas and the liquid, the Euler number due to possible gas pocket effects, boiling (cavitation number) as well as hydroelastic effects have to be considered. An implication is that the effects of viscosity (Reynolds number), surface tension (Bond number) as well as the change of the speed sound due to a mixture of gas and liquid are likely to be of secondary importance (Faltinsen *et al.*, 2009).

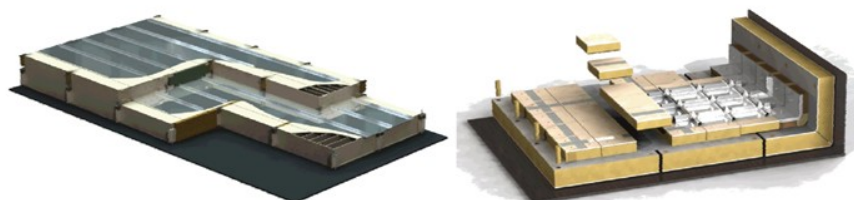


Figure 3: Two types of containment systems NO96 (left) and MarkIII (right).

The complex scaling issues are discussed, among others, by Yung *et al.* (2009) where attempt was made to propose a rational scaling procedure. The authors conclude that, despite the thermodynamic complexities along the NG/LNG phase boundary, dynamic similitude for sloshing is possible for geometrically similar models regardless of length scale provided that the Euler number, the Froude number and the Interaction index are the same. In particular, the Interaction index, which relates dynamic pressure communication between the ambient vapor and the sloshing liquid, provides a means to scale impact pressures for model tests with fluids readily available at convenient thermal conditions. The work of Yung *et al.* (2009) was a part of very extensive research done by Exxon Mobil in cooperation with GTT (Kuo *et al.*, 2009; He *et al.* 2009; Issa *et al.*, 2009; ...) with the final goal to produce a rational design methodology based on direct calculation approach. However, this very interesting methodology has not been applied in practice yet, which suggest that still many uncertainties exist.

Methodologies proposed by the Classification Societies for the practical design verification of the containment system are still essentially based on the so called comparative approach which relies on the use of the small scale model tests for reference and target ship. Within this comparative approach the small scale model tests on the reference ship, which doesn't experience any damage, are used to deduce the conservative pressure scaling factor and the same scaling factor is applied to the target ship. After that the resulting pressure loading at full scale is deduced and compared to the capacity of the containment system. The critical point in the analysis is obviously the scaling factor which does not have clear rational justification since it mixes all the different hydrodynamic phenomena into a single number.

Let us also mention that in addition to the evaluation of hydro-structure interactions during impacts, the direct calculation methodology for sloshing requires a very complex seakeeping analysis which has to be fully coupled with sloshing dynamics. This is obviously necessary in order to determine the representative design tank motions. Finally, a very complex statistical analysis is required both on seakeeping and sloshing impact sides in order to simulate the ship life.

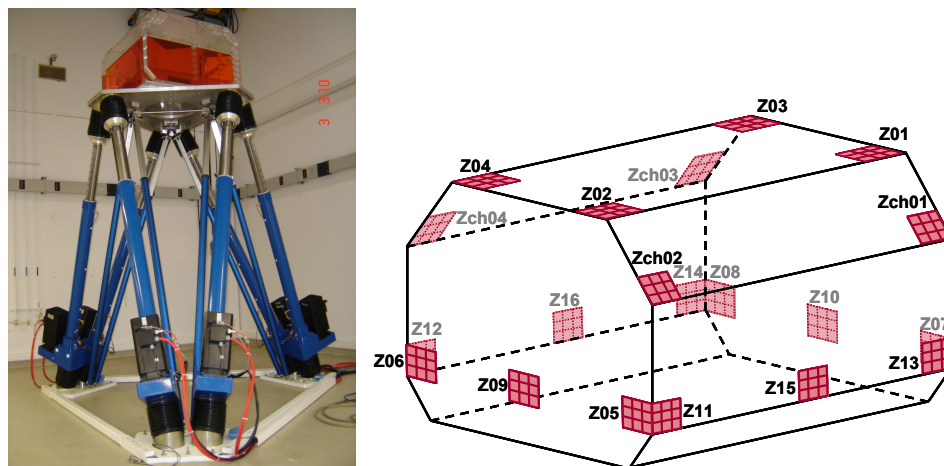


Figure 4: Hexapod system for sloshing model tests and typical pressure sensor locations

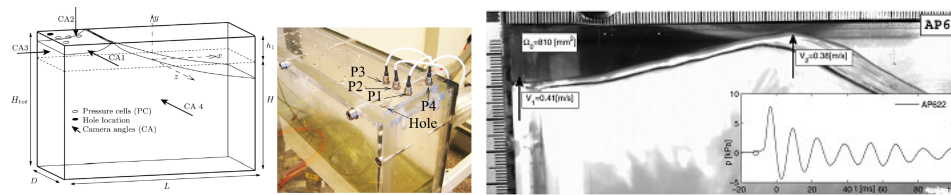


Figure 5: Model tests on impact involving the air pocket and typical pressure signal

4.2 Model Tests

Several different types of model tests at different scales and with different objectives were proposed in the last few years. In particular, small scale sloshing model tests became nowadays rather classical and many important facilities exist all around the world and allow for testing the tank models at scale up to 1/25. The most typical sloshing model testing facilities are based on the use of hexapod (Figure 4) which showed to be very efficient in generating arbitrary time history of the tank motions.

As far as the overall sloshing behaviour is concerned the small scale model tests are very useful and give good qualitative impression of the violent fluid flow. At the same time the overall forces on the tank show good repeatability regardless of the model scale (Diebold *et al.*, 2011). This is because the overall sloshing behavior is mainly driven by the Froude scaling. When it comes to the measurements of pressure the situation is much more complicated; both regarding the repeatability and accuracy of the pressure measurements and, as already indicated, regarding the scaling of the measured pressure to the full scale. Different works on small scale model tests were published in the last few years (Kim *et al.*, 2009; Maillard *et al.*, 2009; Repalle *et al.*, 2010; ...).

In Abrahamsen *et al.* (2011) a dedicated model test to investigate the specific impact type on the roof of the rectangular tank was performed (Figure 5). The impact type is the one with the entrapped air pocket. The goal was to investigate the decay of the oscillations in the air pocket and possible sources of damping. Authors concluded that the leakage is not the main cause of decay and that heat transfer in between air and water might be important. Similar investigations were done by Lugni *et al.* (2010) where the breaking wave impact involving the air pocket entrapment was studied under different ullage pressures. One of the conclusions is that the influence of the ratio in between ullage and vapor pressure plays an important role and the decay of oscillations is much stronger in the vapor pressure regime. This suggests that the phase transition in between liquid and vapor phases plays an important role for damping the pressure oscillations.

This fact was also confirmed by Braeunig *et al.* (2010) where this phenomenon was investigated both experimentally (water and steam) and numerically. In Figure 6 the difference between the pressure signals with and without phase transition are obvious. All this illustrates again the difficulties related to the scaling of the model test results.

Very extensive experimental database of drop tests at small or full scale were produced at PNU by the team of Prof. Kwon (Chung *et al.*, 2007; Kim *et al.*, 2008; Oh *et al.*, 2009; Kwak *et al.*, 2010; Oh *et al.*, 2010). Very useful pressure measurements and high speed video of different impact types on NO96 and MarkIII geometries were produced. These types of measurements are essential for better orientation of the numerical developments and for their subsequent validation.

Driven by the difficulties related to the scaling, a very ambitious experimental project Sloskel (Figure 7) reported by Brosset *et al.* (2009) was initiated by GTT, Bureau Veritas, MARIN and Shell, and has been joined later by American Bureau of Shipping, Ecole Centrale Marseille, Chevron, ClassNK, Det Norske Veritas and Lloyd's Register. The originality of the experiments performed within Sloskel project lies in the fact that the real CCS was impacted by realistic wave impact conditions at full scale. The only, however not negligible drawback is that water under atmospheric conditions was used instead of LNG. Very extensive database of both loading (pressures, forces,...) and the structural response of the CCS were collected both for NO96 and MarkIII CCS. Maximum measured pressures went up to 56 bars and still no significant damage of the CCS was observed. Thanks to the Sloskel experiments significant progress in understanding of the physics of the sloshing impacts was made.

The fundamental importance of the local flow characteristics prior to the impact was confirmed once again. This means that every detail flow aspect makes the direct assessment procedures very complex. This also means that the analysis of the small scale model tests without the corrugations (MarkIII) or raised edges (NO96) should be done with greatest care. Among other interesting results from the Sloskel full scale experiments, it is worthwhile to mention the detailed analysis of the fluid flow evolution during the different impact situations. One example of typical impact on MarkIII CCS is shown in Figure 8.

Following these investigations, Brosset *et al.* (2011) proposed the classification of the different impact phases into different elementary loading processes (ELP). In that respect 3 main ELP's were identified: (1) the actual impact (discontinuity of velocity), very localized and inducing acoustic pressure with the local velocity of sound of the aerated water; (2) the building of a jet along the wall from the impact area; (3) the compression of entrapped gas pockets or escaping gas jets. The idea behind this classification seems to be the decomposition of the arbitrary impact situations into different ELP's. Once each ELP properly assessed (still not clear how!) the final result will be the sum of the different ELP's in time. This work is still in progress and no final conclusions can be made yet.

Many other interesting issues (scaling - Bogaert *et al.*, 2010, deformation of the foam - Kaminski *et al.*, 2011, ...) were investigated within the Sloskel project and the analysis of the huge databases is still in progress.

At the same time, Sloskel project generated very important research activities which accompanied the full scale tests. Indeed, during the full scale experiments different difficulties were identified, one of the main being the lack of repeatability of the measurements for some important impact conditions. It was thus decided to investigate

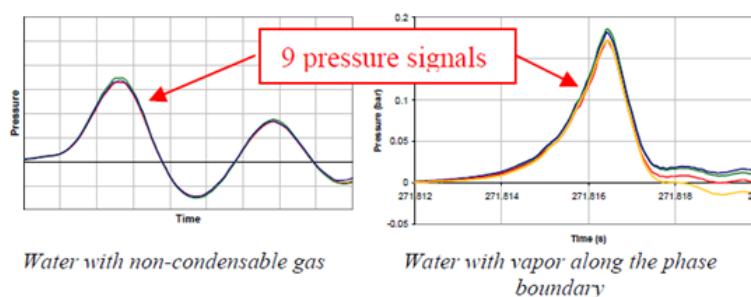


Figure 6: Air pocket pressure signature for different conditions



Figure 7: Quasi full scale impact experiments (Sloshel project)

this issue on a smaller scale and on a more simplified elastic structure. The MiniSlo project was organized and large scale model tests were performed in Ecole Centrale de Marseille. Measurements of the fluid flow (PIV) pressures and structural deflections were undertaken and very useful database for validation of the numerical codes was produced. Due to the well controlled laboratory conditions repeatability of the measurements was very good. One example of the measurements is shown in Figure 9. It is very likely that this kind of experiments will have larger importance in the future.

Parallel to the experimental work, important numerical activities were also performed within the Sloshel project (Oger *et al.*, 2009; Wang *et al.*, 2009; Braeunig *et al.*, 2009; Maguire *et al.*, 2009; Pillon *et al.*, 2009; Malenica *et al.*, 2009; Guilcher *et al.*, 2010; Dobashi *et al.*, 2010; Carden *et al.*, 2011; Wang *et al.*, 2011; Lee *et al.*, 2011; De Lauzon *et al.*, 2011). Different types of numerical methods were used (volume of fluids CFD, smooth particle hydrodynamics (SPH), semi analytical methods, ...) for both

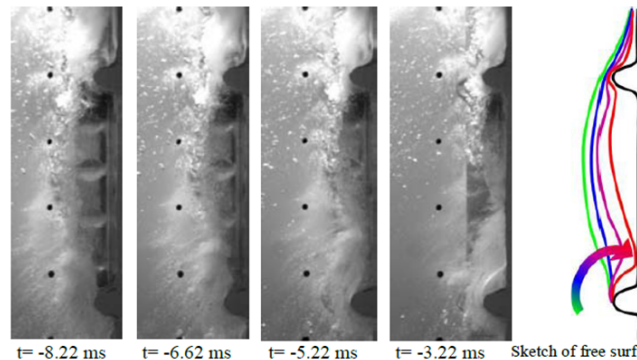


Figure 8: Different phases of the fluid flow during the impact on MarkIII CCS.

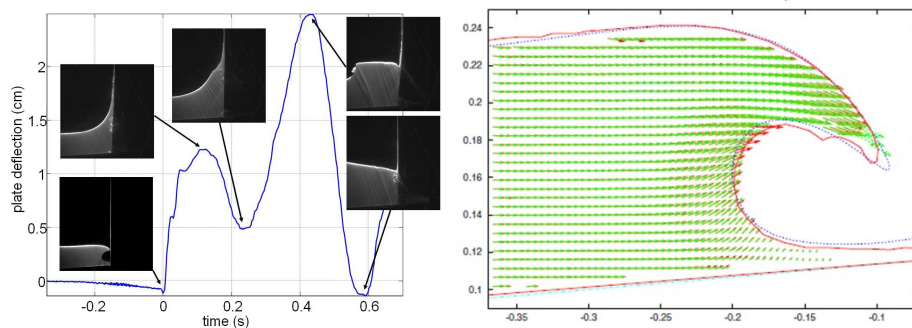


Figure 9: Large scale impact experiments (MiniSlo project)

rigid and hydroelastic types of hydro-structure interactions. In spite of all the efforts there still seems to be no fully efficient numerical method able to simulate this problem consistently.

Different other works on small/large scale model tests was done in the last few years (Kim *et al.*, 2009; Maillard *et al.*, 2009; Repalle *et al.*, 2010; Kim *et al.*, 2011) where different phenomena were investigated (pressure statistics, impact flow evolution, influence of density ratio, ...). One very important aspect of the model tests is the statistical properties of the pressure measurements. A large degree of uncertainties and scatter are usually observed (e.g. Fillon *et al.*, 2011). In this context, it is also important to mention that each pressure signal is not characterized by its maximum value only but the pressure should always be analyzed in combination with its time history (rise and decay time, oscillations, ...) and the surface which is affected. This introduces the additional non-trivial technical difficulties into this already complex problem.

4.3 Numerical Simulations of Sloshing

Different numerical methods for sloshing are proposed in the literature (e.g. Godderidge *et al.*, 2009; Chen *et al.*, 2009; Wemmenhove *et al.*, 2009; Rudman *et al.*, 2009; Ma *et al.*, 2009). These methods are mainly based either on potential flow, Euler or full Navier Stokes assumptions. Different numerical approaches which are usually employed are: BEM – Boundary Element Method, CIP – Constrained Interpolation Profile method, FDM – Finite Difference Method, FEM – Finite Element Method, FVM – Finite Volume Method, LS – Level-Set method, MAC – Marker-and-Cell method, MPS – Moving Particle Semi-implicit method, SPH – Smoothed Particle Hydrodynamics method, VOF – Volume-of-Fluid method and others.

Within the numerical methods for modelling of sloshing it is also worthwhile to mention the nonlinear analytically-based multimodal method proposed by Faltinsen *et al.* (2009a, b). The advantage of the method is its semi-analytical character which allows for fast calculations and detailed separation of different driving phenomena for sloshing. However, even if this method gives good insight into the overall sloshing motions it cannot be applied to the analysis of sloshing impacts.

With respect to all the numerical work which has been done, it is fair to say that there is still no fully efficient numerical method to deal with the overall sloshing hydro-structure interactions in a consistent way. Indeed, it appears that from computational point of view, it is impossible to take all the different physical effects at the same time. This is not only because of the prohibitive CPU time requirements but also because of the complexity of the physical phenomena which are involved (violent free surface deformations, hydroelasticity, phase transition, compressibility, 3D effects, low temperature, ...). That is why the actual research is more oriented to a kind of hybrid approach where the problem is subdivided into global and local parts. Indeed, the global fluid flow during sloshing can be reasonably described by the classical CFD tools but the complete treatment of the complex impact situations at the same time, appears to be impossible today. With respect to this, CFD can be used to determine the local conditions before impact (essentially the relative geometry and the relative impact velocity distribution) and the dedicated models for local impact simulations can be used for evaluation of the CCS structural response.

This idea was first introduced by Korobkin and Malenica (2006) and the most recent advances were presented in Ten *et al.* (2011). For different impact types (steep wave impact, impact with air-pocket, aerated impact and their combinations) which were

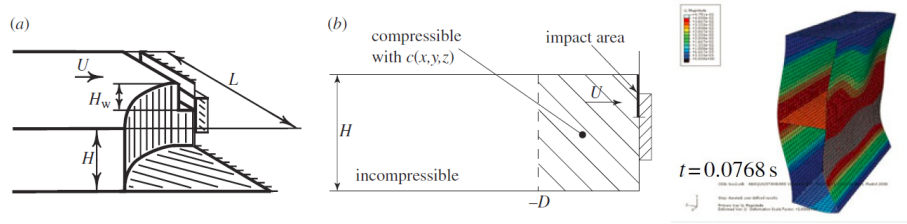


Figure 10: Example of simplified semi-analytical model for partially aerated impact type

identified for the low filling levels the semi-analytical (or semi-numerical) approach for fluid-structure interactions has been presented. Within this approach, the fluid flow is treated using the semi-analytical methods while the structural part is solved using the three-dimensional finite-element model. The choice of the simplified semi-analytical approach for the fluid flow was made in order to be able to have a full control of the flow characteristics, which allows for detailed investigations of the influence of different physical parameters. One example of the typical simplified impact situation is shown in Figure 10.

Different papers on the specific impact types were presented by the team of Prof. Korobkin (Khabakhpasheva *et al.*, 2009a,b; Malenica *et al.*, 2009; Ten *et al.*, 2009a,b; Khabakhpasheva, 2011). This work is ongoing and there is still lot of work to be done especially concerning the validation of different impact models.

The practical idea behind this global-local approach is to perform simplified parametric calculations for different impact configurations involving a small number of impact parameters (impact velocity, aeration, air-pocket volume, relative angle in between fluid and structure, ...) and check the structural resistance. Parallel to that the CFD (or alternatively small scale model tests) will give the most probable maximum value of the impact parameters. Both global and local results will then be combined in order to make the final check of the structural integrity of CCS. Similar ideas based on the exclusive use of CFD for both global and local flow is presented by Cho *et al.* (2008).

Finally let us also mention one important problem, which seems to not receive enough attention in the literature, and which concerns the numerical modelling of the CCS structure. As already indicated, CCS is a very complex structure composed of different materials connected together by special procedures and the representativeness of the classical finite element models should be considered more seriously. Even if some work on this issue has already been done (Isso *et al.*, 2009; Arswendy *et al.*, 2011a,b) this point requires more careful attention.

5 GREEN WATER

5.1 General

Ships and offshore structures are designed to withstand extreme sea states. This can be the extreme in the operational region for the offshore structure, or, for ships, usually the extreme in the North-Atlantic Ocean is chosen as the worst case scenario. Extreme sea states can lead to extreme events, one of these events is the exceedance of freeboard by the wave crest that results in a flow of water onto the deck and possibly an impact against a structure on the deck. Such an event is labelled “green water” and is the subject of research already for a good many years. Main reason for this research is the occurrence of damage on deck structures which are traditionally not designed for

these loads. For ships this damage is usually not critical, although a complete loss of the vessel can occur (Derbyshire accident), but for offshore structures like FPSO it can easily lead to oil leakage and loss of production time.

Interestingly, green water effects are also proposed for wave energy generation. Buchner and Jaouen (2009) describe model tests on a moored vessel where green water flows intendedly over the bow into a tank amidships. This tank drains via low water head turbines, thus generating an electrical current. The idea was presented in 2009, further work was published by Buchner, van der Schaaf and Hoefakkers (2010). The green water aspect of this device is only a secondary power source, the main power comes from the anchor system that, via the motions at the bow, drives an on-board Power Take-Off system (PTO).

The review has been organized in three major categories, i.e. analysis by numerical methods, approximate method and experimental analysis.

5.2 Numerical Methods

Numerical methods are being used more and more to predict loads on deck and deck structures. Although good results are being obtained by different researchers using different methods, in the most cases the conditions are very artificial. Even today, it is not realistic to expect a good statistical distribution of green water loads in a realistic sea state. The problem is the excessive CPU requirement, but more important one is a long term simulation of a sea state in a numerical domain with proper wave evolution. Furthermore, it is not yet possible to obtain realistic results without reflection from the boundaries of the domain.

The most traditional method for extreme free surface deformations is the Volume of Fluid (VoF) method by Hirth and Nichols (1981). This method is still being used today, and similar methods like the Marker-density method by Lee *et al.* (2009) are being developed. The Marker-density method has, similar to VoF, problems in capturing the actual free surface; essentially the volume fraction of a cell is calculated. Calculations were carried out by Lee *et al.* using a very simplified wedge-type ship and a step-like water surface to generate the incoming wave. These results were qualitatively compared to the experiments by Greco *et al.* (2004) on a tanker model that was fixed in the basin; the pressure on deck was compared to the experiments by Pham and Varyani (2004) with the S-175 container ship. Since this is a different geometry and different impact conditions, no conclusions can be drawn from the apparent agreement.

Brodtkorb (2008) used the VoF method (ComFlow) to develop design rules for deck structures of jacket-type platforms. Forces on the deck appeared to be very spiky, for a problem of the VoF method that was reported before by Kleefsman *et al.* (2004). Brodtkorb used a low-pass filter on these forces before further analysis. Calculations for different geometries were carried out and compared with the API rules, a simplified design rule. The maximum horizontal load varied as function of the impact height and the geometry of the deck like external deck girders. The external deck girders incurred significantly higher loads than those given by the API method (Figure 11). The external deck girders appeared also to increase the vertical force on the deck. Brodtkorb *et al.* (2008) extended this work by including three slender pillars to model the jacket platform. They extracted velocities from the VoF calculations and used Morrison force coefficients to calculate the loads. Under deck structures were accounted for by applying geometry weight factors. Again the objective of the study was to develop an improved design load for impact loads on jacket-type offshore structures.

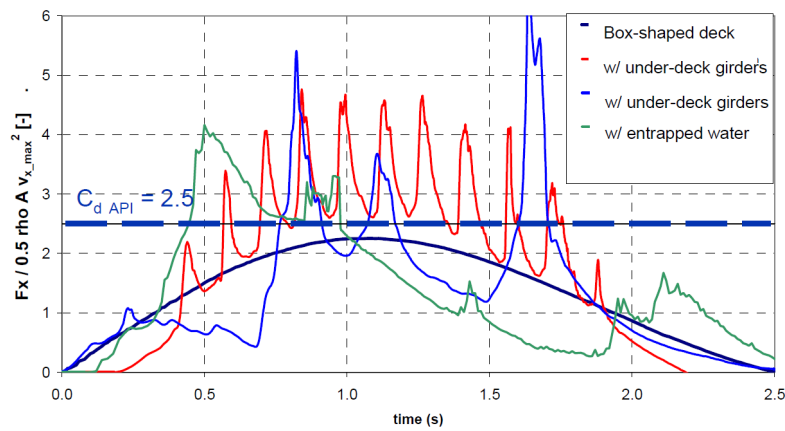


Figure 11: Horizontal force on a simplified deck structure with and without external girders. Results of CFD calculation compared to API method (Brodtkorb, 2008)

Schellin *et al.* (2009) studied the same problem using the combination of a VoF method and a FE analysis. The VoF method appeared to be quite advanced including a cavitation model for vapour bubbles that appear in low pressure regions and including an under-relaxation technique to avoid pressure peaks. Calculations were carried out on a coarse and fine mesh; the finer mesh showed a similar general flow behaviour, but the peaks of the horizontal forces were higher (Figure 12). These researchers

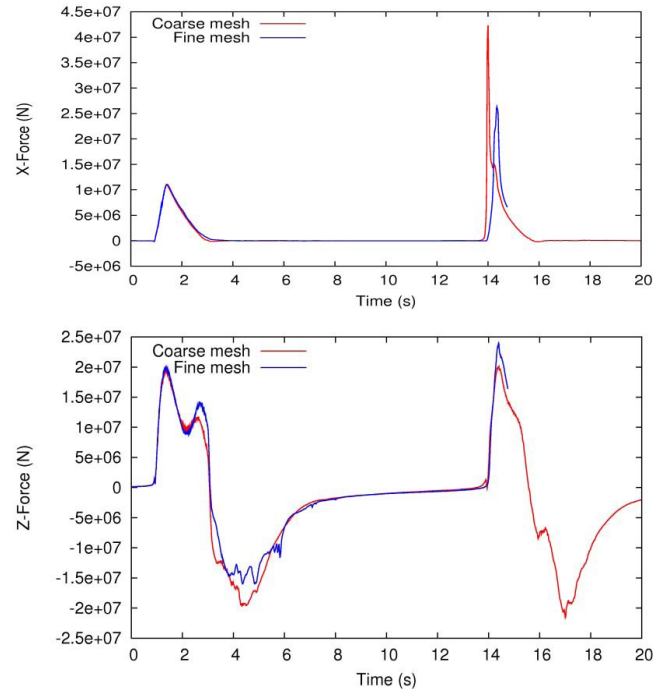


Figure 12: Effects of fineness of grid on horizontal (upper) and vertical (lower) forces on a platform for two successive wave impacts (Schellin *et al.*, 2009)

also recognized the importance of the local steepness of the wave crest, and that the velocity and consequently the loads are much higher if the wave overturns just before the impact. By using overlapping grids, one grid connected to the platform, the other lined up in the direction of the waves, they studied impacts in different wave directions. The results for the studied platform showed that impact forces were highest in head and following waves.

Liang *et al.* (2009) used the commercial solver FLUENT to model green water events and impact pressures on a moving 2D object in waves. For this 2D case they used 2.0×10^5 cells. The results were compared to experiments carried out by Hu *et al.* (2006). The initial body motions agreed very well to the experimental values, stronger deviations occur after a few oscillations, possibly due to the effect of the water on deck. A comparison of local pressures is not shown.

Also Zhu *et al.* (2009) used the commercial code Fluent to calculate green water events, pressures on deck and pressures on a vertical wall of a moving body. The results were presented for a 2D case and compared those to the experiments by Greco *et al.* (2001) with very good results. They also presented results for a 3D FPSO and compared against the experiments by Buchner (2002). A case in regular waves was simulated using the provided motions of FPSO calculated with potential theory and imposing those in the RANSE method. The results for the pressure on deck were very good, the pressure on the vertical bulkhead compared less favourably to the experiments.

Lu *et al.* (2010) developed a VoF method for incompressible flow problems. The method uses an unstructured grid and solves the Navier-Stokes equations using an arbitrary Lagrangian-Euler (ALE) frame of reference. The method is demonstrated using a 2D example of a wave flume with waves overtopping a horizontal deck. Results of the wave elevation on various locations in the flume, and at the edge of the deck the velocities were compared well against experimental results published by Cox and Ortega (2002) with focused waves. Calculations of the pressure on deck for a 3D block with very low freeboard was compared well to the experiments by Yamasaki *et al.* (2005), but when with a vertical wall installed on the model the impulsive pressures on deck were less well predicted. Finally the case of a moving model of an FPSO was used. The motions were imposed on the numerical model. The numerical grid consisted of about $5.0 \cdot 10^6$ elements; a simulation time of 45 s (model scale) took 29 hours on a 2-core processor. The results of pressures on deck and on the deckhouse showed a good correspondence to the experimental results by Liang *et al.* (2007).

Shibata and Koshizuka (2007) used a particle method to calculate the green water shipped on a fixed structure. Results were compared against the experiments by Tanizawa *et al.* (2004) on a FPSO. The numerical model was a simplified version of the physical model saving CPU time, and about 3.3×10^5 particles were used in the calculation. The wave elevation at the bow agreed quite well to the experiments, but not the pressure on deck. It was concluded that the particles need to be smaller for a more accurate local pressure. The method was further developed and new calculations were presented by Shibata *et al.* (2009). In the calculations the numerical domain was minimized to save CPU time. The wave flume was limited to a length of about a half wave length in front of the vessel. Only the bow part of the vessel was modelled and the cross section of the wave flume was made triangular. The vessel motions were imposed using the results of the experiments; the walls were forced to move with the orbital velocities. By using symmetry on the centreline (the experiments were in head waves) they needed $1.1 \sim 1.5 \cdot 10^6$ particles for the computational domain. The CPU time required was 2.5 ~ 4 days on a single core PC for a simulation time of 1.3 s. The

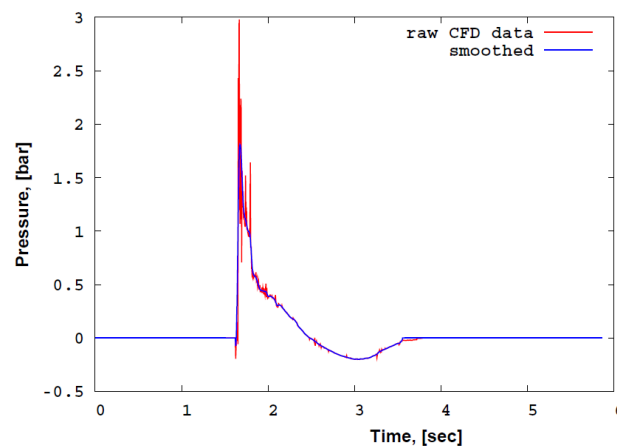


Figure 13: Spiky local pressure signal calculated by VoF method and smoothed curve, (Iwanowski, 2009)

general behaviour of the shipping phenomenon agreed very well to the experiments, but calculated pressures on deck showed very large numerical oscillations, although the values shown were averaged over 25 particles. The pressure impulse, $\int p dt$, showed a bit better agreement to the experiments although the differences were still 15 ~ 27 % in comparison to the experiments.

Iwanowski *et al.* (2009) coupled a FE analysis to the results of the VoF calculation for the same problem of a wave impact against the deck of a jacket platform. They also used the ComFlow method, similar as used by Bordtkorb (2008) and Brodtkorb *et al.* (2008). Also Iwanowski *et al.* removed the pressure peaks by a filtering and smoothing procedure, Figure 13. The structural analysis was done using the commercial package LS-DYNA; the FE method was used only as a postprocessor to the hydrodynamic code. The results of this study were not verified against other results.

5.3 Approximate Methods

A very simple prediction method for bow flare slamming and green water loads on deck and on the superstructure was developed by Stansberg *et al.* (2009). The method uses the relative motions at the bow as computed by linear theory but enhanced with a non-linear corrections. Green water on deck is calculated by a simple formula for dam-breaking by Stoker (1957). Impact pressures are calculated with a simple formula using a slamming coefficient and the dynamic pressure. The method is intended to predict the maximum impact pressures in the early design stage; it is acknowledged that simplicity was achieved at the cost of accuracy. For more accurate predictions Stansberg advises to use CFD methods like a VoF method.

5.4 Experimental Methods

Experiments for green water events are still being carried out today. The objective of such experiments can either be to determine extreme loads or it can be for the validation of computer programs. Although it is realized that also scaled experiments violate scaling laws with respect to the effect of air entrapment, it is still today the best – if not the only – method to arrive at a statistical distribution of extreme loads. However, most experiments in the open literature are done on simplified models or on

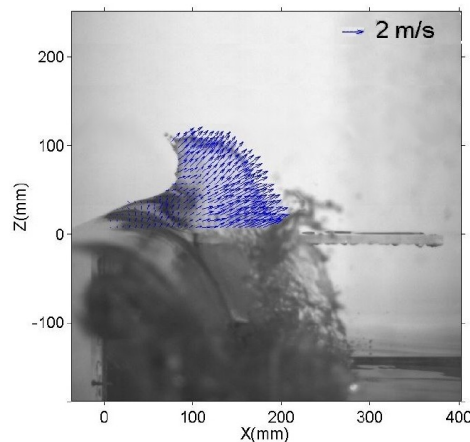


Figure 14: Flow velocities during an impact measured by Bubble Imaging Velocimetry (BIV) (Ariyaratne, 2009)

full blocked ships at zero speed (like a FPSO) using simplified wave conditions like regular waves or a focused wave to generate one single extreme event.

Rather fundamental experiments were carried out by Ryu *et al.* (2007). The experiment was carried out in a wave flume where a focused wave overtopped a fixed structure. The velocities in the fluid were measured using Bubble Imaging Velocimetry (BIV) as developed by Ryu *et al.* (2005). This technique is comparable to PIV, but instead of introducing particles in the flow the velocity of air bubbles is measured. The measurement plane is not illuminated by a laser sheet, this would lead to avoid scattering, but illuminated from the opposing side as the cameras. This method creates sufficient contrast to allow tracking of the bubbles. The results of the experiments appeared to correlate surprisingly well to the analytical dam break model by Ritter (1892) if or the initial water depth or the front velocity of the analytical model is properly tuned.

The objective of the experiments carried out by Ariyaratne *et al.* (2009) was to serve as validation material for numerical codes. They used a simplified wedge-shaped model of an FPSO, rigidly connected to the bottom of the basin. An extreme event was created using a wave focusing technique, flow velocities during the impact were measured in great detail using BIV, Figure 14. The instrumentation is described in detail, but, since the incoming wave is not presented in the paper, the results cannot be used by other researchers. Essentially the same results are again published in Chang *et al.* (2011); in this case also the measurement of the void fraction in the flow by a Fiber Optic Reflectometer (FOR) is mentioned; the void fraction is used to correct the pressure.

Experiments carried out by Tanizawa (2004) were used by Shibata, Koshizuka and Tanizawa (2009) for the validation of a particle method. The experiments were carried out with a towed model of a VLCC in regular waves; the pressure on deck was measured with 5 pressure gauges.

Also the work presented by Lee *et al.* (2010) is intended as validation material for numerical codes. They carried out systematic experiments on a barge type FPSO. They used a model, which was fixed in the basin, with a vertical blunt bow, an inclined blunt bow and a rounded bow to measure in detail the pressures on deck. The wave conditions used in the experiments were not realistic sea states, but regular

were. The intention was to have as simple as possible inflow conditions to facilitate CFD validation studies. A second advantage of using regular waves is, that – after the transient period – this constitutes essentially repeat tests. This allowed Lee *et al.* to carry out an uncertainty analysis.

Model tests on an FPSO in shallow water in wind, waves and current were mentioned by Guo *et al.* (2010). They used these data for validation of their CFD calculations, but since details of the experiments were not given, these results cannot be used by other researchers.

6 UNDERWATER EXPLOSIONS

6.1 General

Underwater explosions can cause significant damage to structures such as ship hulls. Considerable effort has been spent on understanding the physics behind these explosions so that precautions can be made to avoid critical damage. Many underwater explosion (UNDEX) studies were done with numerical simulation in addition to the experiments.

6.2 Numerical Procedure for Evaluating Deformation and Rupture under Explosion

There are three ways in which information may be exchanged between the fluid and structural solvers, as shown in Figure 15. The first is to transfer only pressure load at the structural interface from CFD solver to FE solver for structure. This one-way coupling is used when loading from the fluid domain is desired but the response of the structure has little influence on the load calculation. This procedure is useful for estimating the loads on a rigid body.

The second option transfers the fluid loading to the FEA solver and structural node velocities are transferred back to CFD solver. This option may be used for small deformation scenarios where the structural displacement has little influence on the fluid domain but the velocity of the structure changes the resulting pressure loading and by the response of the structure. This procedure may be used for estimating the cavitation of water. Cavitation effects play an important role in the UNDEX loading of

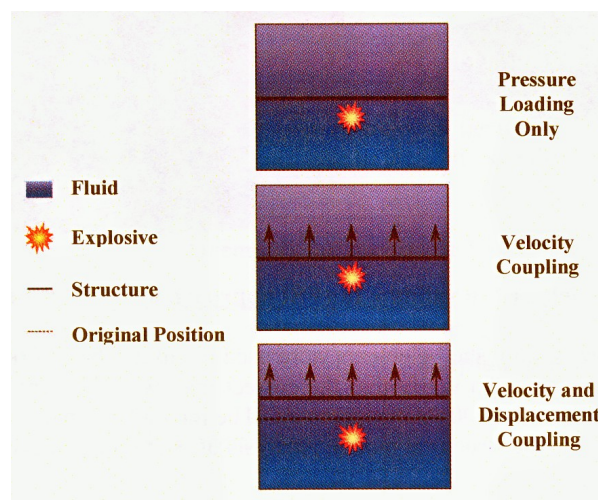


Figure 15: Three kinds of FSI coupling procedure

a structure. For far-field UNDEX, the structural loading is affected by the formation of local and bulk cavitation regions, and the pressure pulses resulting from the closure of the cavitation regions. A common approach to numerically modelling cavitation in far-field underwater explosions is Cavitating Acoustic Finite Elements (CAFE) and more recently Cavitating Acoustic Spectral Elements (CASE). Treatment of cavitation in this manner causes spurious pressure oscillations which must be treated by a numerical damping scheme. The third loading option is the same as the second option with the addition of the structure position being updated in the fluid domain. This option may be used for large deformation problems, where the structure moves over a large region of the fluid.

Using the DYSMAS code Wardlaw (2009) investigated computationally the detonation of a submerged charge beneath the plate suspended over a water surface. Simulations were conducted for a range of plate stand-offs and charge depths and validated against laboratory scale experiments. The results showed that the loading is dominated by water ejected upwards by the detonation for the case of a submerged charge and a plate suspended above the water. And, the shock produces pressures much larger than those associated with a plume strike for the case of either the plate or charge being on the surface. Aanhold *et al.* (2009) simulated a heavy underwater shock trial on the floating cylinder by three dimensional calculation. The calculations were done using the so-called Simplified Interaction Tool (SIT), an approximate interaction method developed by TNO as an add-on to LS-DYNA. The SIT is a very efficient tool for estimating the underwater shock response of complex 3D models of surface ships, because the water around the ship is not modelled by means of finite elements. Only added mass effects are considered in FEA. The results showed a good agreement of vertical motions with experimental results.

Lee *et al.* (2009) investigated the underwater explosive loading and failure of thin steel plates for close-proximity charges. The shock and bubble jet loading was measured, and a distinction was made between failure caused by shock alone, and failure caused by the cumulative loading from the shock and impinging bubble jet. They showed that the failure standoff limit increases as the plate thickness decreases. For 350WT steel the failure standoff limit increases as compared to A1008 mild steel because of smaller failure strain. These trends are correctly reproduced by a simple FEA model, but the standoff limit does not show good agreement.

Riley *et al.* (2009) conducted the experiments on rigidly-clamped circular and square air-backed steel plates, in which underwater explosive charges were placed at varying standoffs including contact. They performed the FE analyses with LS-DYNA using two different failure criteria based on a combination of normalized transverse shear stress and direct strain. They concluded that the LIC failure criterion predicts the onset of failure in a more realistic manner than the QIC.

$$LIC = |\varepsilon_e / \varepsilon_{rup}| + |\tau_e / \tau_{dult}| \quad (1)$$

where ε_e is the true element membrane strain, ε_{rup} is the true rupture strain, τ_e is the maximum of the through thickness shear stresses in the element, and τ_{dult} is the dynamic ultimate shear strength.

Riley *et al.* (2010, 2011) performed an extensive numerical modelling study using the Eulerian computational fluid dynamics (CFD) code Chinook, in standalone mode and coupled with the Lagrangian solver LS-DYNA, to investigate the prediction accuracy of the loading on rigid plates and displacement of flexible target plates subjected

to close proximity underwater explosion events. For the rigid targets, qualitatively Chinook was found to accurately reproduce the general trends in the experimental measurements. Quantitative gaps still remain in the load levels predicted. A major issue with Chinook is the lack of a material interface tracker which would allow for the distinction between the gas bubble and the surrounding water. For the flexible targets, the numerical simulation displacement time histories were compared to experimentally measured responses. They showed that Chinook impulse predictions were closer to the experimental results for the shock loading than the bubble collapses. They also showed that the three-dimensional models with a coarser fluid mesh give better agreement with experiments than more finely meshed two-dimensional models.

Dunbar *et al.* (2010) also investigated numerically to simulate a series of underwater explosions with the intent of estimating the critical standoff range at which the onset of rupture occurs by using CFD code Chinook and FE solver LS-DYNA. They showed that both the peak structural displacement and maximum bubble radius compare well with the experimental and empirical based solutions. And the peak shock pressures were improved by mapping from a detailed 2D model to a 3D model. However, this improvement did not significantly change the overall displacement, indicating that the response is possibly impulse dominated.

Stojko *et al.* (2010, 2011) examined the LS-DYNA/USA Fluid Structure Interaction (FSI) acoustic underwater shock methods, namely the Doubly Asymptotic Approximation (DAA), Cavitating Acoustic Finite Element (CAFE), and Cavitating Acoustic Spectral Element (CASE). A number of verification problems have been analysed and were compared with 'exact' solutions. The general strengths, limitations and suitability of the three methods were discussed. They also compared the calculation result of the bubble pressure, frequency, and radius with experimental results. For the bubble frequency, and radius and pressure trend versus depth the calculation results showed a good agreement with experiments, but did not show the quantitative matching of pressure.

Helte *et al.* (2011) performed small scale experiments to investigate the behavior of a flexible circular plate subjected to a close proximity underwater explosion. The most prominent effects are shock loading, target induced cavitation, loading from cavitation closure and bubble collapse. They also performed the calculation using the in-house 2D multi-material arbitrary Lagrangian-Eulerian hydro code GRALE2D. The performance and sensitivity of the parameters in the fluid-structure coupling, such as a penalty based method, and the cavitation modelling, a simple cut-off model, were of particular interest. Good agreement between experiments and simulations was obtained. But, the bubble collapse times in the simulations were too short in all cases. This could be an indication that the bubble energy in the simulation was wrong. However, the predicted response of the target from the bubble collapse was higher than the measured, contradicting the hypothesis of too low bubble energy.

Klenow *et al.* (2010) focused on investigating the severity of bubble oscillations on the structural response and a possible improvement to CAFE, based on the original Boris and Book Flux-Corrected Transport algorithm on structured meshes, to limit oscillations without the energy loss associated with the current damping schemes. By comparing CAFE, CASE, and the FE-FCT algorithm in the two-degree of freedom mass-spring oscillator problem, they showed the FE-FCT algorithm, which uses linear finite elements on structured meshes, used with residual diffusion and a one-sided flux limiter, is effective in reducing the larger oscillations associated with the CASE method while maintaining the increased accuracy.

Xie *et al.* (2008) developed further MGFM (Modified ghost fluid method) in order to increase the quality of results when simulating a close-in explosion in a deformable filled cylinder. Initial MGFM method was not very robust for this type of simulation because the FSI (Fluid-structure interaction) technique in this method didn't perform well under cavitation reload and solid tension wave i.e., the convergence was very slow and negative interface pressure appeared during the simulations. They proposed to solve the FSI explicitly rather than implicitly to avoid convergence problems and rewrote equations at the FSI zone without using tension stresses inside the solid in order to avoid negative pressures. Shin *et al.* (2011) investigated the applicability of numerical calculation using LS-DYNA ALE code for estimating shockwave motion of gas bubble generated by high explosive. They confirmed that the shockwave pressure was underestimated when the large number of element is not used. For the size of bubble and the time of expansion, the calculation results showed good correlation to the empirical formula. They also investigated the effect of ship speed on the dynamic response of high speed Mono-hull, catamaran, and trimaran in underwater explosion. They showed the possibility of calculation for hull rupture under explosion.

For the structural integrity assessment of pipelines subjected to underwater explosions, Monti *et al.* (2011) proposed an engineering approach, taking into account of loading due to the shock wave and gas bubble pulsation. Analytical and numerical approaches using ABAQUS/Explicit concerning the assessment of the structural response of the pipeline were presented, and criteria for Serviceability and Accidental Limit States were proposed.

6.3 Application of Composite Structure for Reducing Damage

Recently, there has been an increased interest in the application of composite structures in the marine industry to take advantage their high stiffness to weight and strength to weight ratios, and high impact/shock resistance characteristics.

Dunbar *et al.* (2009) investigated the polymer coating effect of plate on the deformation under explosive loading. To examine this numerically, they adopted the FSI approach using Chinook, Martec's CFD solver, with LS-DYNA, FE solver. Solid elements are desired for simulations where the through-thickness properties and resolution of layered materials, like the polymer coating plate, are required. But, this FSI approach was implemented for only structural shell elements. Then, for coupling to fluid domain, the shell elements with a null material are paved on the solid elements. Using this procedure for modelling, they investigated the effect of the polymer coating on the deformation under explosive loading and confirmed that the maximum plastic strain was reduced by as much as 45 % for the thin plate model investigated by adding the polymer coating to the plate.

Xie *et al.* (2009) showed a 2D numerical case study for the transient analysis of an air-backed three-layered sandwich beam with clamped ends subject to a close-in underwater explosion, and the results were compared with a similar case with a rigid structure. They showed that structural deformation and transfer of energy lead to a reduced pressure shock and the initial shock is mostly resisted by the bottom steel face, and later followed by compression (plastic yielding) of the soft foam core. The energy absorption and dissipation provided by the soft core layer helped to protect the rear steel face, which showed negligible deflections and stresses that are one to two orders of magnitude lower than the front steel face.

Liu *et al.* (2010) investigated the influence of interfacial bonding on the transient response of sandwich plates subject to underwater explosions. They found that un-

bonded sandwich plates receive lower impact energy, and are able to dissipate more energy through plastic deformation of the foam core than perfectly bonded plates. Consequently, interfacial de-bonding leads to lower net energy transfer from the explosion to the target structure although it also increases the structural deformation due to stiffness reduction. Parametric studies showed that the advantage (diminishing of net energy transfer) is more significant than the disadvantage (magnification of the interface deflection). Thus, interfacial de-bonding through active/passive mechanisms may be beneficial for blast-resistant designs.

6.4 Mounting of Equipments

In naval ships, some methods or devices are acquired both to cut off the transmission of vibration from shipboard machineries and to protect them from external shock loading. One of the approaches is to install the passive mountings between machinery and a flexible supporting structure. More advanced performance has become necessary recently so far as at high frequencies in order to retain the stealth function of certain types of naval vessels.

Czban *et al.* (2009) compared the shock test severity of the Mil-S-901 lightweight machine with the drop tests outlined by the newly proposed ANSI National Standard for equipment in a Rugged Shock Environment. The resilient mounting system in the drop test series (ANSI) bottomed out, while it did not during the Mil-S-901 tests. It is concluded that while the drop test shock environment may not be representative of underwater explosions.

Moon *et al.* (2010) developed a new hybrid mount for shipboard machinery installed on naval ships. The mount is combined with a rubber mount and piezo-stack actuators. The rubber mount is one of the most popular and effective passive mounts to have been applied to various vibration systems to date. The piezo-stack actuator is featured by a fast response time, small displacement and low power consumption. Through a series of experimental tests conducted in accordance with MIL-M-17185A (SHIPS), MIL-M-17508F(SH), and MIL-S-901D which are US military specifications related to the performance requirements of the mount, it has been confirmed that the hybrid mount shows more effective performance for use in naval ships.

7 DAMAGE TO STRUCTURES

7.1 General

Impulsive pressure loading induces different kind of damage on floating structures. Great pressure impulses may cause local plastic deformation on the loaded region but may also cause global damages in the midship section of a ship structure. Intermediate pressure pulses which occur several times may gradually cause large deformations or cause low cycle fatigue damages. Low pressure pulses with a high number of cycles may cause fatigue damages in structural details.

Generally the different types of loading may cause different kind of damages. In this chapter, the relevant literature regarding the permanent deflection as well as fatigue damages caused by impulsive pressure loadings has been reviewed. The chapter is divided in section corresponding to the loading types discussed in recent literature.

7.2 Slamming and Whipping

In January 2007 the Post-Panamax container vessel MSC Napoli was severely damaged in the English Channel. The vessel encountered a severe storm that overloaded the

structure resulting in the collapse of the hull girder just aft of the forward engine room bulkhead. It was concluded by DNV that the vessel did not have the necessary buckling strength margin. It also was stated that whipping could have contributed to the dynamic loading. Many Researchers concentrated on this item trying to find relationships between whipping and damages on ships. One result from the last ISSC-V7 committee work in 2009 was that the contribution from vibratory response doubles the fatigue damage induced by wave-frequency loads for bulk and container carriers.

Miao and Temarel (2009) analyzed the influence of whipping-induced loads on the structural strength of a container ship focusing on the investigation carried out on the failure of the MSC Napoli. Based on two-dimensional symmetric hydroelasticity analysis and relevant structural, hydrostatic and operational data, calculations were carried out in head regular and long-crested irregular waves. The investigation showed that whipping, due to bottom slamming, is only important for severe seas. The investigation also showed that the keel stresses, in way of the engine room, can be as large as the keel stresses at amidships. Storhaug (2009) published measurements from a 4400 *TEU* vessel similar to MSC Napoli in full scale and model tests. These measurements showed that whipping can increase the dynamic loading in similar sea states as MSC Napoli encountered. The measurements also illustrate that it is difficult to state exactly the amount of whipping in a specific sea state.

Experimental model investigations carried out by DNV, BV, HHI, CeSOS and Marintek for a 13000 *TEU* Container Vessel were presented by Storhaug *et al.* (2010a). In a similar project DNV, HHI, CeSOS and Marintek investigated the effect of springing and whipping on a 8600 *TEU* Container Vessel (Storhaug *et al.*, 2010b). The results show that wave induced vibrations can be of considerable magnitude relative to the conventional wave fatigue damage for the different trades. For the East Asia to Europe trade a fraction of 65 % of the total fatigue damage was related to wave induced vibrations. In this regard, whipping was considered far more important than springing. Dessi and Ciappi (2010) experimentally investigated the relation of slamming excitation and whipping response on a fast ferry sailing up to 40 knots at full scale and a cruise ship.

However, laboratory tests may not fully reproduce the critical conditions that may occur in reality (Gaidai *et al.*, 2010), e.g. model tests are mainly carried out in head seas. Measurements on real vessels in operation in harsh weather provide unique insight into the phenomena involved in the structural response to impulsive pressure loading. Mathisen *et al.* (2009) analyzed measurements from hull-monitoring systems on bulk carriers and container vessels to investigate the effects of wave-induced vibrations on fatigue damage. The results show a significant contribution to the total stress from the vibratory component under the harshest conditions that were available. The relative magnitude of the vibration stresses indicates that hull girder vibrations may need to be taken into account in the prediction of the extreme stresses in container ships.

Heggelund *et al.* (2010) presented the assessment of data measured on an LNG carrier during a period of about twelve months. It was found that the vessel has been in operation less than half the time during the actual period. The fatigue rate is found to be lower than predicted by component stochastic fatigue analysis and that the fatigue life is expected to be longer than the design life. Further, the contribution from vibration is found to be large (30 – 50 % of the total damage). The highest fatigue damage is obtained in rough seas and in the full load condition. It is found that most fatigue damage is accumulated in head or following seas.

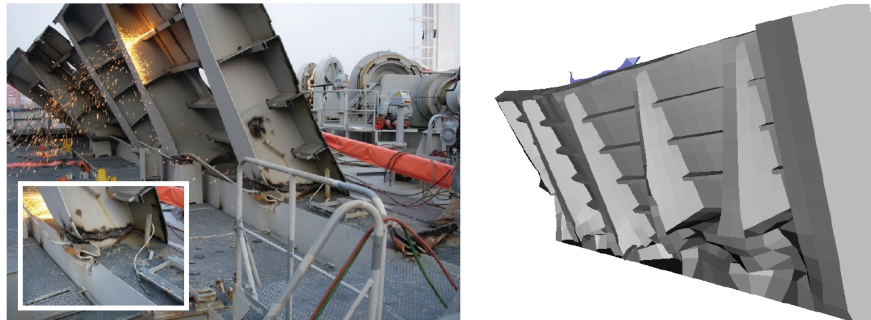


Figure 16: Comparison of actual collapsed breakwater (left) with numerically predicted one (right) (Heo *et al.*, 2010)

7.3 Green Water

The structural damages due to green water are mainly related to large deformations of local structures. Adegeest *et al.* (2009) studied the causes for severe breakwater damage of a Container Vessel crossing the North Atlantic in heavy weather. The particular interest was to identify possible measures to avoid failures in the future. The analysis involved linear and nonlinear sea keeping theory and a green water load calculation by adopting a 2D slamming theory developed for wedges. In general, good accordance with rules was found concerning calculated water pressures. Furthermore, the FEM calculations confirmed the experienced failure of the breakwater when subjected to the calculated pressures. Several studies of breakwater constructions and possible configurations were carried out.

Heo *et al.* (2010) proposed new design formulae that may be used to evaluate the structural performance of breakwaters installed on container vessels under green water impact loads. A series of numerical analyses for green water impact loads inducing post-buckling and breakwater collapse have been carried out. A verification study of the numerical results was performed using the actual collapse incidents of breakwaters on container carriers (Figure 16).

7.4 Underwater Explosion

Underwater explosion damage to steel panels has been studied extensively in the past. Recently interest on in-port vessels has prompted detailed research into structural damage from close-proximity underwater charges. Lee *et al.* (2009) investigated the detailed damage mechanisms caused by explosions of close-proximity underwater charges to thin target plates. The main loading on the plate was due the shock. However, because the standoff distances were less than twice the maximum bubble radius, a strong interaction between the detonation product bubble and the target plate caused a rapid water jet to impinge on the plate and cause additional loading and damage. As a result, four main regimes of loading and damage were identified: a) holing/petaling due to shock loading, b) edge tearing due to shock loading only, c) edge tearing due to the cumulative loading from shock and bubble collapse, and d) large deformation due to shock and bubble collapse loading.

In their study Lee *et al.* (2009) also performed finite-element analysis to investigate the detailed response and failure of the plates. Finite-element analysis showed good agreement with the experimental dynamic displacement due to shock loading. Plate slippage at the clamped boundary was found to influence the results significantly.

Riley *et al.* (2009) investigated the transition between the first failure mode (holing and petaling) and second of these failure modes (complete or partial edge tearing due to shock only). At less than 0.1 times the bubble radius holing, edge failure due to shock, and large plastic displacements without rupture were all observed in plate specimens. Explicit finite-element analysis with LS-DYNA was used to investigate the detailed response and failure of the plates. In the FE simulations the influence of different approaches including shell and solid elements for the plate, different failure criteria based on a normal strain, shear strain, or a combination of normalized transverse shear stress and direct strain. Finite-element analysis shows good agreement with the failure mode as well as with the post test deformations.

Yiannakopoulos *et al.* (2009) presented results from an exploratory study of the penetration of an aluminium target plate by a close-in underwater explosion. Two identical target plates were subjected to different levels of explosive load such that one was perforated whilst the other suffered only plastic deformation. High speed imaging and measurements of underwater pressure for these two events were used to probe the conditions leading to plate fracture. The data was compared with results from a LS-DYNA3D FE model to investigate the potential of current material models to capture this behaviour.

8 COMPARISON OF CLASSIFICATION SOCIETIES RULES

8.1 General

Traditionally, Classification Societies have made the safe requirement for the impulsive response based on a state of the art theory and many experiences. However, different procedures for the requirement have been developed according to the damage data due to the impulsive loads which each Classification Society has collected from its classed ships. Recently, IACS (International Association of Classification Societies) has implemented CSR (Common Structure Rules) for Tankers and Bulk carriers since the 1st April in 2006. Therefore, tankers and bulk carriers classed to an IACS member constructed with the same scantlings due to the impulsive loads.

In order to investigate the different requirements from Classification Societies Rules for impulsive response like slamming loads, comparative calculations have been performed for a container ship.

The principle particulars of the container vessel are given as follows;

- Ship Type: 4,600 TEU
- LBP: 240.5 m
- Breadth (Mld.): 37.50 m
- Draft (Scantling; Mld.) : 13.0 m
- Block Coefficient: 0.646
- Design Speed: 21.4 knots

8.2 Plate Thickness and Stiffener Section Modulus Required by Bottom Slamming Pressure

The required plate thickness and stiffener section modulus due to the bottom slamming at the following draft condition, which is assumed to be one of the worst loading conditions vulnerable to the bottom slamming for the container ship, have been calculated according to four different Classification Societies requirement.

- Draft at A.P.: 8.967 m
- Draft at F.P.: 4.517 m

The calculation results are presented in Figures 17 and 18.

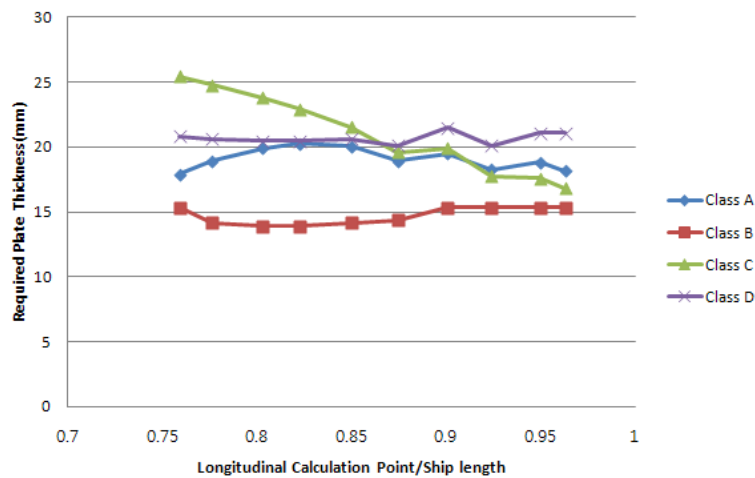


Figure 17: Required thickness of the bottom plates at centre line by the bottom slamming pressure

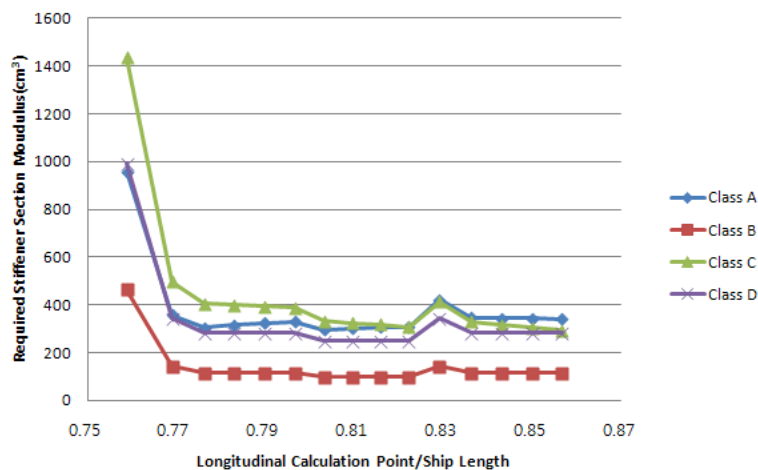


Figure 18: Required section modulus of the stiffeners at centre line by the bottom slamming pressure

8.3 Plate Thickness and Stiffener Section Modulus Required by Bow Flare Slamming Pressure

The required plate thickness and stiffener section modulus due to the bow slamming at a section of longitudinal position/ship length = 0.9 with the scantling draft (13 m) have been calculated according to three or four different Classification Societies requirements. The calculation results are plotted in Figures 19 and 20.

9 RECOMMENDATIONS FOR STRUCTURAL DESIGN GUIDANCE

9.1 General

The extreme difficulties related to the proper modelling of impulsive pressure loadings and the associated structural responses clearly appear throughout all the chapters in

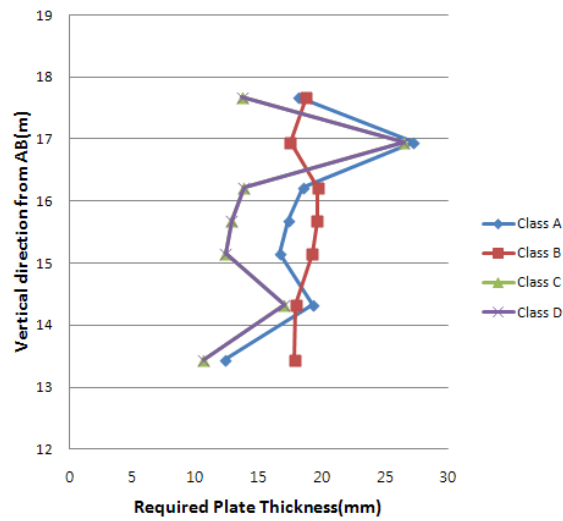


Figure 19: Required thickness of the shell plates at a section of longitudinal position/ship length = 0.9 by the bow flare slamming pressure

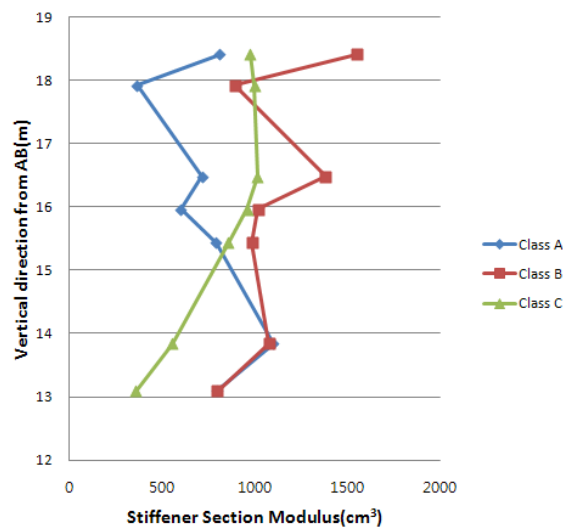


Figure 20: Required section modulus of the stiffeners at a section of longitudinal position/ship length = 0.9 by the bow flare slamming pressure

this report. This is probably one of the most difficult aspects of the hydro structure interactions in ship design. Indeed a very complex hydrodynamic flow needs to be coupled with the evaluation of the structural response. Whether this coupling should be weak or strong mainly depends on the ratio of the impulse duration relative to the natural period of the impacted structure.

In an attempt to improve the quality of the marine structural design against impulsive pressure loadings, some recommendations are provided herein based upon the reviews and investigations performed by the committee.

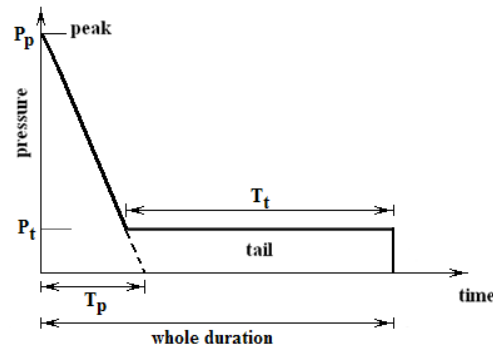


Figure 21: Idealised impulsive pressure history (Cho *et al.*, 2011)

9.2 Impulse Shape of Local Slamming Load for Design

For practical impact response and residual deformation analyses of local slamming loads it seems desirable to simplify the impulse shape. Some of the recently investigated results are summarised below.

Effects of tail part; When the peak duration, T_p is shorter than a half of the natural period (T_n) of a impacted plate, the effect of the tail part on the extents of damage can be 8.2 % at most. However, when the peak duration, T_p is the twice of T_n , the extents of damage can be a constant value regardless of the tail part duration. Therefore, the effects of tail part on the extent of damage can be negligible because it is known that the actual duration of the peak part of impulse due to slamming can be greater than the natural period of the impacted plate.

Effects of peak part shapes; For triangular impulses, if peak duration is shorter than a half of T_n , the damage extents are almost the same. However, when peak duration is longer than T_n , the monotonically decreasing impulse shows greater residual deformation than the other types of triangular impulse. When the impulse duration, T_p is shorter than T_n the isosceles type impulse gives the greatest deflections. However, when T_p is longer than T_n , the monotonically decreasing impulse yields the greatest deflections. According to published studies, it can be concluded that the rising time

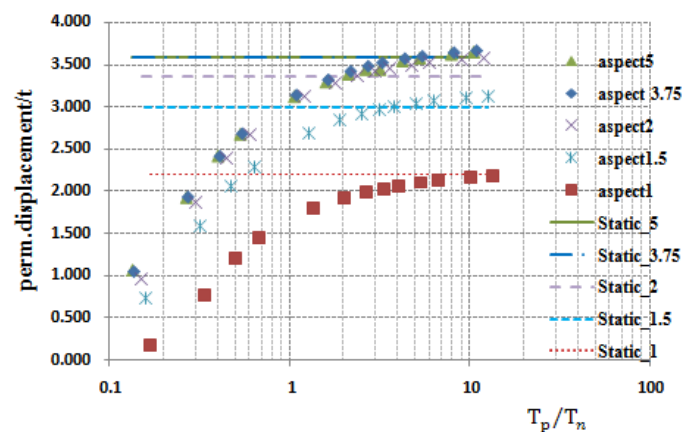


Figure 22: Effects of aspect ratio of unstiffened plates subjected to monotonically decreasing triangular impulse (Cho *et al.*, 2011)

of impulsive pressure loadings due to slamming can be neglected from the structural design view point.

Therefore, from a structural design view point, the impulsive pressure loadings induced by local slamming can be approximated by monotonically decreasing triangular impulses.

9.3 Effect of Aspect Ratio of Impacted Plates on the Damage Extent

As be seen in Figure 22, the extents of damage are increased according to the aspect ratio values. However, when the aspect ratios are greater than 2.0 the predicted damage extents are nearly identical regardless of aspect ratio values. In the figure, the extents of damage due to the static pressure are also depicted. When the aspect ratio is 1.0 the dynamic responses approach to that of static. However, when the aspect ratios are greater than 1.5 the converged extents of damage are greater than those of static.

9.4 Global Slamming

In the structural design of a slim high-speed ship, the local slamming loads and the springing effects of ship hull girder should be properly considered. Furthermore, slamming may induce whipping moment and the effects of which on the fatigue strength of the structure should be considered, especially of the bottom structures and the hatch corners. For the ships such as container ship and destroyer etc., the angle of the bow flare should be optimized to reduce the flare slamming. For ships with large openings, the torsion vibration effect induced by the slamming when the ship sailing in oblique wave should be studied.

9.5 Sloshing

The evaluation of the structural response of the CCS and the associated ship-structure in the LNG tanks is an extremely complex problem and the actual state of the art does not allow for rational direct calculation numerical approach. For that reason one has still to rely on experiments which means in practice model tests. The challenges are how to scale the model test results to full scale and properly account for the structural elastic reactions due to the fact that a rigid model is used in model scale. There are many contributing factors to scaling which have to be considered and one has to do important approximations. The small scale model tests are used in the context of the so called comparative approach where the scaling issues are hidden within the semi empirical considerations. Formally, the classification societies also accept validated direct numerical approaches. However, it still seems not to be a good candidate to perform these calculations, in spite of significant advances in numerical modelling which were achieved in the last few years.

9.6 Green water

Recent studies show that a considerable effort is being spent on applying CFD codes to the problem of green water and wave impacts on the superstructure. The main focus is on the zero-speed head seas case before sufficient confidence is built to approach the problem of a ship at speed in quartering waves. There is not yet a clear preference for any particular method, good results are also obtained with commercial codes. It seems very well possible to determine the general behaviour of the flow over the deck and also the pressure on the horizontal deck is well predicted by different researchers. This is not the case for the pressure on a vertical bulkhead. It is however not generally realized that such an impact is a rather chaotic event were small details of even air inclusions

can have a large effect. Consequently there are large spatial variations of the pressure and results can only be compared in a statistical manner. Another consequence is, that a twophase simulation modelling both water and air seems to be required for these types of impacts. The importance of turbulence of the green water problem is not fully investigated and studies using different turbulence models are welcomed.

9.7 Effects of Multiple Impacts

As mentioned in the last ISSC report (Cho, *et al.*, 2009) the plastic deformation of shell plates as a consequence of multiple impulsive pressure loadings may be significant. According to numerical and experimental investigation results the effects of the repetition of the impulsive pressure cannot be neglected in the structural design against impulsive pressure loading.

One of the open questions is which material model, especially the strain hardening model, is most suitable for modelling the deflections due to multiple impact loadings. While for a single impact it is quite irrelevant if kinematic or isotropic hardening is used as basis, for multiple loadings these hardening models follow different hysteretic curves resulting in a different permanent deflection.

9.8 Classification Societies Rules

Recently various investigations and research have been conducted to improve the accuracy of predictions and efficiency and stability of related calculations in the area of the impulsive loading including slamming. According to the state-of-the-art technology, innovations of the relevant classification society rules are progressing.

10 CONCLUSIONS

10.1 Local Slamming

Model or full-scale tests are still the most reliable approach in obtaining the pressure distribution and force on temporal and spatial scales, especially in disturbed water. Analytical approaches have been developed to a high degree of accuracy when predicting the impulsive slamming pressures in calm water for most dead-rise angles of the body with simple profiles.

Numerical simulations of slamming pressures have been, to a great extent, developed with acceptable accuracy and efficiency. Commercial or in-house software based on the CFD technique have attracted more attention in recent years and is an encouraging prospect. Due to the complexity of the slamming phenomenon, practical methods to obtain rational design pressures, forces and structural dynamic responses for a new design is still required.

10.2 Global Slamming

Due to the 3D characteristics of the bow flare, the direct adoption of any 2D methods will induce some error. This is of particular concern for ship sections with a relative roll angle during the impact. The 3D character of the bow and bulb of container ships is particularly challenging to model.

While FE methods provide excellent tools for modelling the structural behaviour, the main challenge in estimating global slamming response is the calculation of the slamming force, especially in oblique seas due to the effects of rolling motion. However, the full FE models of complex ship structures are still quite computationally demanding and combined use of simple models and refined models in a hierarchical approach is useful.

The high frequency fatigue damage due to whipping can be significantly reduced by including the steady wave for the relevant vessel, implying better correlation with the experimental results. Therefore, more work needs to be done to improve the high frequency stress modelling. This includes amongst others, identifying and quantifying the sources of damping of the vibration as well as verification of the excitation sources of high frequency response.

10.3 *Sloshing*

Correct numerical modelling of hydro-structure interactions during the sloshing impacts inside the LNG tanks is still beyond the state of the art and there is still no rational direct calculation procedure to be used for design verification of the CCS. It is however important to mention that the different research projects which were undertaken during the last years have brought much more light into the physics of the sloshing impacts and important progress was made in the modelling of sloshing impacts both experimentally and numerically.

The full scale measurements and monitoring of the real LNG ships would be extremely helpful for better understanding of the way how the CCS is “suffering” in reality. Indeed, with respect to all the difficulties discussed above, it appears clearly that more feedback from experience is necessary in order to get more confidence into the existing design procedures. How to perform these full scale measurements is another complex question.

In any case, the actual situation is that, for the design verification of CCS, we still rely on the so called comparative approach. It is however important to mention that, in spite of all the imperfections of the comparative approach, the overall safety record of LNG floating units is excellent and only few incidents were experienced (Gavory *et al.*, 2009).

10.4 *Green Water*

In general, a good CFD calculation includes a verification study. Such studies were different densities of the grid are being used, different time steps and convergence levels are tested are very labour and time intensive. Presently these studies are carried out for stationary problems; of course the effort required for the stationary problem is much larger, but the work is necessary to demonstrate that the numerical schematization of the problem has converged. Only one of the reviewed papers included results for two grids.

Also experimental studies suffer from the rather chaotic event of a wave impact against a vertical wall. In order to get information on the accuracy of the experiments, repeat tests are required. In the ideal world, a comparison of a numerical result to results from experiments should include confidence intervals for both data based on the numerical verification study and the repeat tests in the experiments.

There appears to be a surprising lack of interest for the statistical distribution of extreme pressures or loads in the period covered by this review. Using a design wave technique for the ultimate (impulsive) load presumes knowledge about the critical parameters for wave impacts; this is as yet unproven. It is however clear that a long-term distribution cannot be predicted by CFD methods, for the next years we have to rely on experimental techniques.

10.5 *Underwater Explosions*

Traditionally, underwater explosions have been treated separately as shock wave problems and gas bubble ones due to the big differences of their time scales. However, for

close proximity underwater explosions the shock wave and bubble effect can be coupled. Even with today's powerful computational techniques it is not possible to obtain meaningful numerical results. Practical design-oriented procedures, therefore, need to be developed for more advanced structural design especially of naval vessels.

10.6 Damage to Structures

Various structural damages due to impulsive pressure loadings have recently been reported and reviewed herein. Some of the damages can be categorised as the accidental limit state but some can be the serviceability limit state. Design loads specified in relevant rules and regulations need to be reevaluated and the adequacy of the design processes assuming equivalent static pressures have to be reassessed based on the fail safe design concept.

10.7 Comparison of Classification Societies Rules

In order to investigate the requirements of several Classification Societies rules for the slamming pressure acting on the hull structure, comparative calculations have been performed for a 4,600 teu class container carrier. The required plate thicknesses and stiffener section modulus due to the bottom slamming pressure and the bow flare slamming pressure acting on the hull structure of the container ship are compared in accordance with the several Classification Societies rules. A little difference among the classification societies can be found in the requirements by the slamming pressure. The difference seems to be due to different damage data by the slamming loads which each Classification Society has collected from its classed ships.

10.8 Recommendations for Structural Design Guidance

Based upon the recent progresses regarding the predictions of impulsive pressure loadings and structural responses reviewed in this report, recommendations for structural design guidance are provided. However, for the betterment of the marine structural design against impulsive pressure loads collaborations between related organisations including classification societies are required.

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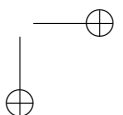
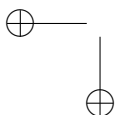
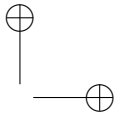
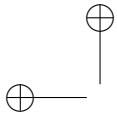
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18th INTERNATIONAL SHIP AND
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09-13 SEPTEMBER 2012
ROSTOCK, GERMANY
VOLUME 2



COMMITTEE V.8
YACHT DESIGN

COMMITTEE MANDATE

Concern for the structural design of sailing and motor yachts and similar craft. Consideration shall be given to the material selection, fabrication techniques and design procedures for yacht hull, rig and appendages. Attention should be given to structural issues associated with special fittings as large openings, inner harbours, pools etc and with security requirements. The role of standards, safety and reliability in the design and production processes should be addressed.

COMMITTEE MEMBERS

Chairman: Dario Boote
Robert Beck
James Blake
Richard Flay
André Hage
Han K. Jeong
Lex Keuning
Paul Miller
Leigh Sutherland
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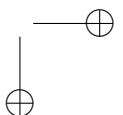
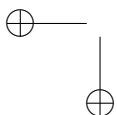
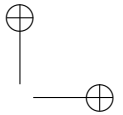
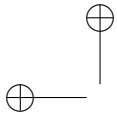
KEYWORDS

Motor yacht, design, structures, loads, materials, outfitting

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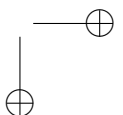
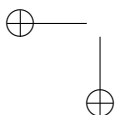
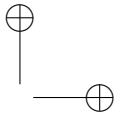
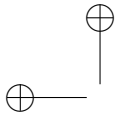
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18th International Ship and Offshore Structures Congress (ISSC 2012) - W. Fricke, R. Bronsart (Eds.)

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Proceedings to be purchased at <http://www.stg-online.org/publikationen.html>



1 INTRODUCTION

Even if the term ‘yacht’ was coined specifically for the sailing world, this word has been associated with the concept of going to sea for pleasure purposes, and extended to include ‘motor boats’ as well. Nowadays, when speaking about ‘yachts’, one can refer both to ‘sailing yachts’ or ‘motor yachts’ and it should be specified which of the two is intended. The previous ISSC 2009 Report of V.8 Committee was specifically dedicated to sailing yachts: in this second mandate of the V.8 Committee, owing to the large size of the worldwide motor yacht market, it was decided to focus on this very important and challenging sector of the marine industry.

The term ‘motor boat’ generally refers to a vessel whose main propulsion is provided by a mechanic propulsion system represented, in most cases, by internal combustion engines but can include steam engines or more modern gas turbines. The first motor boats were very simple, small and wooden, and were mostly work boats. The ease of handling and the higher performance of these motor boats with respect to sailing, yachts immediately attracted the attention of the boating public and pleasure motor boats powered by combustion engines soon became very popular. The high demand for bigger, faster and more comfortable vessels made motor boats ever larger and more technologically advanced, culminating in the huge range of pleasure vessels of today, from very simple and small motor boats to highly sophisticated and extremely large motor yachts.

Nevertheless, for a long time motor yachts were designed using an ‘experience-based’ approach by shipyard owners and craftsmen rather than naval architects and designers, and they were considered, in a certain sense, a ‘second class’ category with respect to ships. Nowadays a medium size motor yacht brings with it a huge series of problems to be solved, slightly different from those associated with ships, and these vessels contain a great deal of structural and high tech equipment packed into very concentrated spaces, all aimed at raising passenger comfort and safety to a high level. Whilst, up to very recently, most designers followed tried and tested paths in order to avoid possible mistakes, at present many use advanced design techniques and ‘high-tech’ to make their product stand out from those of the competition. Both attitudes are motivated by the high intrinsic value of the product and a large effort is spent in research and testing.

Thanks to these recent changes, the progress in yacht design and construction has increased significantly, leading to levels of technology equivalent to or exceeding those already existing for ships. Structures in particular have been an important subject of such a development, being heavily influenced by the introduction of new construction materials (such as composites), the increase in performance and size, the need to reduce noise and vibrations, and the continuous search for new shapes and lay outs to acquire new markets. In particular, the length of the yacht represents the main discriminating factor with regard to the technical and commercial typologies of the vessels, which have given rise to the categories ‘superyachts’, ‘mega yachts’, ‘giga yachts’ and ‘dream yachts’. However, the exact definition of these categories in terms of length are to a certain degree subjective and not clearly defined, and the only objective classification is that which divides the fleet into vessels below 24 m in length (‘small yachts’) or over 24 m (‘superyachts’). The worldwide pleasure yacht fleet in 2011 consisted of approximately 23,350,000 units in total, of which 5,980 are ‘superyachts’. The worldwide yearly production (2011) is approximately 550,000 small yachts units and 800 superyachts (values from The Superyacht Intelligence, 2012). The development of market

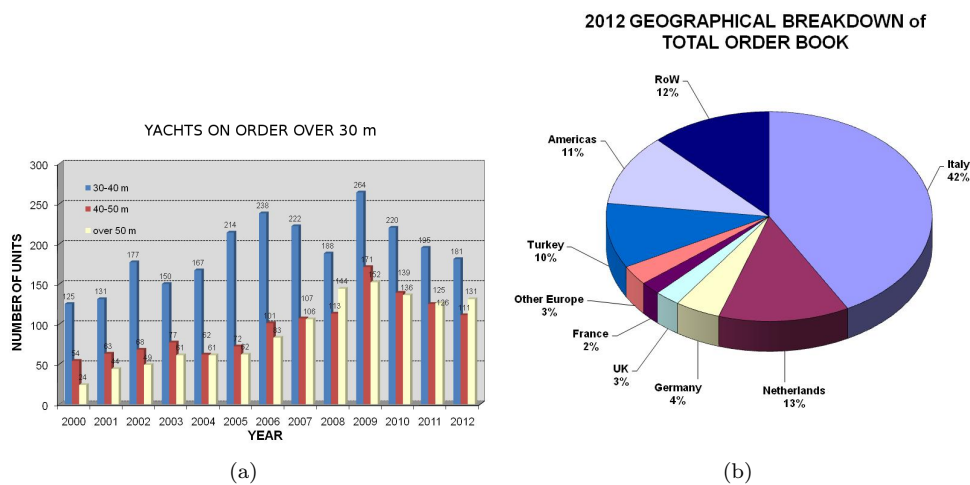


Figure 1: (a) Distribution of yachts on order over 30 metres from 2000 to 2012;
 (b) Geographical breakdown of the total order book of yachts over 30 metres in 2012 (The Superyacht Intelligence, 2012).

share with regard to yacht length is indicated in Figure 1a, which shows the continuous growth of the demand for yachts over 30 m from 2000 to 2009 and the slight decrease started in 2010, due to the economic global crisis. The production breakdown among the various producer countries is also reported in Figure 1b, which shows the market leadership of Italy and Netherlands and the interesting growth of Turkey.

Similar to sailing yachts, the history and development of motor yacht structures can be assessed and described according to various characteristics, such as size, performance, construction materials, interior and external design. The use of wood, steel, aluminium alloys and fibre reinforced plastic for motor yacht construction are discussed with respect to the various vessel typologies, and the relevant technological aspects in the following chapters.

Other concepts which are extremely important in the world of yachting such as ‘freedom’, ‘comfort’ and ‘luxury’ are determinant in attracting the interest of potential owners. Even if these concepts appear to be completely separate from the practical technical aspects, as stated by Nuvolari (2011), they must be translated by designers into real features of the yacht. ‘Freedom’ as an example is often associated with speed, which gives to the yacht commercial impact and visibility. From the technical point of view speed involves a wide range of technical subjects, such as more powerful and lighter propulsion engines, the developments of new propeller systems and water jets, the study of new and more efficient hull shapes, but also requires the development of light and strong hull structures.

‘Comfort’ is mainly related to the seakeeping behaviour of the ship at sea, together with low levels of vibrations and noise on board. These two latter aspects are also closely connected to hull structures, and detailed calculations to verify the dynamic behavior of hull structures and their responses to excitations must be carried out from the first stages of structural design. As far as ‘luxury’ is concerned, this is an additional way to distinguish between different vessels of the same length in order to specify a higher commercial classification, and to justify any associated cost increase. Even if this aspect may sound a little ephemeral to naval architects and marine engineers,

luxury is closely related to styling and fitting-out and, often, encompasses concealed technical challenges. Let's refer, as an example, to the external finish of the yacht, i.e. the hull fairing and painting; this is a very time consuming and difficult procedure, similar to that used in the car industry, and the final result depends on the relative stiffness of both shell panels and the hull girder, as well as the paint support.

After the consideration of sailing yachts by the previous ISSC 2009 V.8 Committee, it was decided to extend to motor yachts the mandate of the same Committee for 2012. The research of this Committee has shown that little published work exists specifically concerning large motor yacht structures, and that, depending on the topology of the vessel, this subject is often assessed in a same way as for 'ships', or using engineering techniques and technical knowledge that is not made public for commercial reasons. Thus, as was the approach used when considering sailing yachts, much of the information obtained here has been gathered via direct contacts with shipyards and engineering technical offices.

2 MOTOR YACHT BASIC DESIGN AND TYPOLOGIES

General guidelines for motor yacht design can be found in many books and manuals such as those by Phillips-Birt (1966) and Mudie (1977). The design philosophy for motor yachts in the sixties and seventies was succinctly summarised in one sentence by Phillips-Birt: "*The variety of power yachts found in the yachting waters of the world results from mixing the four basic ingredients of design in different proportions. The ingredients are: accommodation, endurance, seaworthiness and speed. ... The proportions of the ingredients determine the type of boat; their total amount fixes the size*". Even if still valid for small and medium size vessels, the 'ingredients' for modern motor yachts now also include the present day trends of ever increasing size and opulent comfort and luxury requirements.

2.1 Motor Yacht Basic Design

There are two ways to obtain a motor yacht: to choose it from the huge number of available models on the market (and this is normally the case for small vessels) or to build a new custom (or semicustom) one according to the owner specific requirements. The attention here is focused on the latter option, as the former falls outside the scope of this report.

Despite the fact that the phases of the design for yachts are the same as those for ships and workboats, one major and very important difference exists; the aesthetics (external and interior) together with comfort and luxury requirements drive the concept, design calculation and construction of motor yachts. These qualities appear to have a major impact on the 'dreams' of the potential owner, and they often become his strongest motivation to buy a yacht. Nevertheless the boat must also be safe, have high levels of performance, and yet be easily managed and handled by the crew. The basic design process must consolidate these conflicting requirements via a feasibility phase (concept design) and a preliminary design, right up to the final design. A synthetic analysis of the initial design procedure of a large, high performance motor yacht is presented by Mulder (1996).

The concept design is by far the most delicate phase of the entire procedure; the client generally contacts a specialised design office or the shipyard directly, with the support of his own staff, that is composed a minimum of an architect for interior/external design, a project manager and, often, a lawyer. The initial design parameters are often very few (yacht typology, length and performance), whereas the owner and his

staff are far more interested in addressing luxury items. Since the luxury items may well cause structural problems later in the design and/or construction process, the shipyard technical office must be sufficiently inventive to find solutions which make these ideas feasible. Sometimes the will to distinguish himself through his new vessel pushes the owner towards very audacious specifications which, on the one hand can give the technical team many headaches, while on the other can produce very innovative solutions. An example of fruitful synergy between a technologically advanced platform and a specific 'emotional' design framework is presented by McCartan *et al.* (2011) where the external and interior solutions are calibrated on the requirements of a specific superyacht owner.

Traditionally, the owner's desires are then transferred into paper sketches, and at this stage it is the ability of the designers to make very attractive hand drawings of external and interiors views of the yacht that are important. Even if many designers continue to prefer hand drawings, nowadays this task has been made more efficient by 3D modelling software such as *Rhinoceros*, *3DStudio*, *Think3DDesign*, *Solidthinking*, *Alias*, *Solidworks*, *Formz* etc.

If the owner decides to proceed, the preliminary design consists of initial calculations to obtain the main characteristics of the yacht, and to select the most appropriate construction materials. The subsequent process of the initial design can be started as soon as the contract has been negotiated and signed. Then the main aspects of hydrodynamics, stability and strength are assessed in more detail to be submitted to the Classification Society for approval.

During the design development great effort is spent in obtaining agreement between what the client (and stylist) wants, and what the yard can feasibly provide, but this generally results in too flexible specifications which are subject to changes through the design and building process. If the design would continue until the customer was completely satisfied, this would take far too long, and there is an economic need to mobilise the workforce in the yard before this happens. For this reason construction often starts before the design is finalised, and the risk (perhaps certainty) of the need for modifications during construction if required by the owner is accepted. This also occurs for merchant ships, but whilst in this case rework should not be necessary or at worst inexpensive, for motor yachts with luxury finishing any change implies very high cost. It is usual for disagreement on who must pay for these expensive changes to occur, and this is often only solved after recourse to a court of law.

As a matter of fact, customisation is the key point of yacht design, especially for larger vessels, and this often also reflects into structural design and construction costs. It is then necessary to have access to flexible, parametric tools for structural drawing and scantling which allow any modifications and evaluations of their consequences on the structural, outfitting and related items to be made quickly. This aspect has already been assessed in the field of cruise ships for which the solution has been found in the concurrent engineering concept (for more details see ISSC 2006, Committee Report IV.2 'Design Methods'). From this point of view designers are greatly helped by CAD software such as *Autocad*, *Microstation*, etc. and other more specialised integrated systems such as *Catia*, *SiemensNX*, *Proengineer*, etc.

An example of integrated CAD as applied to a steel superyacht is presented by Mathieu (2011); the application of modern Product Lifecycle Management (PLM) is described where all the activities and information of the early phase of the project are controlled and made available for design and engineering tasks, project management, purchasing,

manufacturing preparation, and for exchanging documentation and validation with classification societies.

A new trend in yacht design is represented by the concept of Design for Disassembly (DfD) which was transferred from the automotive to the yacht industries, as maintained by Schiffer (2011). In spite of the probable, initial problems in embracing this concept as a design and production philosophy, very large advantages could arise for the yacht industry as a result.

The owners' tendency towards repeatedly requiring new and exclusive vessels, together with the continuous search by industry for new forms and layouts to develop and/or acquire new markets, often drives the design towards quite astonishing radical and innovative solutions and new vessel typologies. For example, the transition from traditional stern shapes to the new 'swim platform' shape, the present vogue of reverse bows, or the *Wally Power* motor yacht, whose minimal lines and huge power made it a reference point for this new style.

2.2 Motor Yacht Typologies

The first known motor vessel was *Pyroscaphe*, a 148' wooden side-wheeler boat powered by a double-action steam machine. Built by Marquis de Jouffroy d'Abbans, this vessel made its first demonstration run on 15 July 1783 on the river Saone in France. The origins of motor yachting date back to 1830, when a rich Englishman commissioned the first known private motor yacht, the 130' steam-powered *Menai* designed by Robert Napier and built on the Clyde, Scotland. In 1857, on Como Lake in Italy, Barsanti and Matteucci experimented with a boat powered by an internal combustion engine. In 1883, the first horizontal internal combustion engine was created in Germany by Gottlieb Daimler. In 1886 a launch, called *Neckar*, with a twin cylinder combustion engine was tested on the Waldsee in Cannstatt in Germany.

The majority of present day motor yacht forms arose from the review and development of very old typologies. As an example, 'lobster boats', which were born at the very end of the 1800's in the U.S.A., originated from work boats and later became sophisticated pleasure yachts, only keeping the lines and lay out of the original vessels. Also, in the early 1900's a particular motor boat was designed to meet the requirements of business men living in Long Island who had to reach the New York Centre quickly: named the 'Fast Commuter' it can be considered the direct progenitor of present cruiser yachts.

At present the world-wide motor yacht fleet is composed of a huge quantity of vessels of many different typologies: in order to give a manageable overview of these typologies, the most important yacht categories are briefly outlined below as a function of relevant commercial and technical characteristics. From the commercial point of view the main subdivision is between sailing and motor yachts; this principle has been assumed by the V.8 Committee itself which assessed first sailing yachts in the 2009 Report and then motor yachts in the present 2012 Report.

Within motor yachts the most common subdivision is relative to the yacht overall length L_{oa} , since this parameter is a reference figure for technical, bureaucratic and commercial operations. At present, the worldwide accepted criterion is that of separating yachts with an overall length below or above 24 m, the latter vessels considered by classification societies (CS) as 'pleasure ships'. At lengths greater than this yachts are further subdivided into more subjective categories such as 'mega yachts', 'giga yachts', 'dream yachts' but without any clear objective correspondence to a length range. Below 24 m in length the subdivision often depends on the local classification societies and/or flag rules.

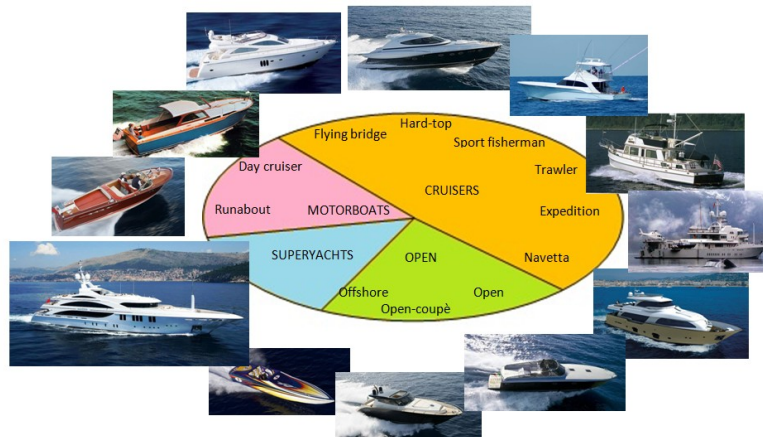


Figure 2: Motor yacht main typologies

A third subdivision refers to the hull typology: monohull or multihull. While catamaran and trimaran configurations are widely diffused in the sailing world, very few examples of multihulls exist as motor yachts, the great majority being represented by monohulls. Also related to the hull shape, motor yachts can be divided into displacement vessels, with a traditional round hull, and planing vessels with a hard chine hull and a flat bottom.

The last subdivision, probably the most important from the structural point of view, is relative to the construction material which heavily influences the design procedure and production technologies. In the following a brief description of the most important yacht typologies is presented, as summarised in Figure 2.

2.2.1 Motor Boats

In the 1930's, with the improvement of internal combustion engines with regards to power, weight and cost, the boat industry identified an attractive new business sector in the diffusion of boating at a popular level. Intensive production of small and fast motor boats took place and in few years many new shipyards were born on the American and European coasts. A typical product of this trend was the 'runabout'; with lengths below 8 or 9 m, planing hulls and equipped with petrol engines derived from the car industry they can reach speeds over 30 *knots*. Completely built in wood (cedar or mahogany planking over oak frames), glued and riveted by copper bolts, they had no deck, with all crew spaces completely open to the elements, or at most covered by a small tent or removable hardtop. They had the same layout of a car with seats, benches, sun beds and driving position with complex dashboards. Very famous names for this typology are Chris Craft, Gar Wood and Hacker of the USA and Riva of Italy.

With the introduction of fibre reinforced plastic (FRP) in the sixties the advantages of series production pushed this category towards great commercial success which still continues today. Even if still inspired by the original typology, commercial competitiveness made runabouts more luxurious, complex and high performance. With a small increase in length (up to 12 m) runabout became 'day cruisers' with a closed accommodation space below the foredeck fitted with a double bed, kitchen area and toilet to allow for short day cruises or coastal passages.

2.2.2 Cruisers

Cruisers are medium to large size boats with a continuous deck and large covered areas below or upon that deck which allow for full living quarters. The exterior aspect is characterized by the presence of an extended length superstructure along a significant portion of the boat. The most valuable spaces are those on the main deck, owner and guest night cabins (with bathrooms), normally lying below the main deck. The external space is fitted with a cockpit astern, for outdoor living, and a sunbathing area towards the bow.

Superstructures can be extended across the whole width of the boat, with a solution called 'wide body superstructure' which allows for larger spaces inside, or they leave space for a gangway of about 0.8 to 1 m along each side to allow easy access from stern to bow (walk around superstructure). On larger yachts a mixed solution is often assumed with a walk around solution astern and a wide body at the bow to maximise the internal spaces as far as possible. If the deck above the superstructure is accessible, it can be fitted with seats, sofas and a second set of driving controls. In this case the yacht is said to have a flying-bridge configuration; this is the most common and appreciated configuration and often gives the name to this category. If the roof of the superstructure is not accessible and it functions as a simple shelter of the internal spaces, the boat is said to have a hard top configuration. Depending on the yacht performance, hulls can be either displacement, semi-planing or planing and are mostly equipped with two engines either with an in-line or V configuration.

A large number of different versions with particular characteristics fall within the cruiser category, which form separate subcategories such as 'sport fisherman' and 'trawlers', which are derived from the evolution of sports fishing boats developed in the United States in the early twentieth century. 'Expedition' (or adventure) yachts have been recently introduced into the market and aimed at owners interested in visiting extreme sea areas characterised, mainly, by the presence of very cold water and floating ice. The first expedition yachts derived from the refitting of old tugs or supply vessels with luxury interiors. Now they are usually new builds from specific new designs (Bray, 2008), and both the demand for and the dimensions of these vessels are increasing every year. 'Navetta' is an Italian term (in English: 'small ship') coined to indicate a motor yacht specifically designed to give excellent levels of onboard comfort during navigation, without demanding excessively high speeds. The relatively short length (not more than 30 m), the necessity to provide large interior spaces and consequently voluminous superstructures, gives such vessels squat lines, making them in some aspects similar to a short ship.

2.2.3 Open

The term *open* indicates a relatively large motor yacht without superstructures and with a wide open area astern protected only by a simple wind screen. This arrangement gives the vessel very 'narrow' and sporty lines combined with spacious and comfortable interiors and very high performances thanks to planing hulls and powerful engines. Open yachts have a single deck extending approximately along the fore half of the boat length and a large cockpit astern, protected by a windshield that extends to the sides to form a kind of bulwark protection. The space below deck is devoted to accommodation and living areas with one or more cabins, depending on the size of the yacht.

Some slightly different versions of the same typology are available; 'Offshore' yachts are a sport version with smaller dimensions but with speeds similar to those of offshore

racing powerboats. Living spaces on board are limited and the layout is very basic to underline the sporty character of these vessels and to minimize the weight. Engines may best be described as exuberant and, together with the fuel, occupy a good portion of the available interior volume. *Open Coupé* is a modern compromise between an open and a hard top yacht: it has a lay out similar to that of a hard top yacht, but has a sliding roof which allows the transformation of the protected space under the superstructure into an open area.

2.2.4 Superyachts

Superyachts represent the development of cruisers in terms of increasing length, resulting from the requirements of very exigent owners looking for an absolutely exclusive and unrepeatable product. This was once attainable only by royal families and very important industrial or business men. Even if actually closer to ships than to yachts, some excellent historical examples should be mentioned as the first ‘mega yacht’: the *Savoia Royal Ship*, 133 m, built in 1883 in Castellamare di Stabia (Italy), the German imperial yacht *Hoenzollern II*, 120 m, built in 1893 in Stettin and the *Victoria and Albert III* Royal Yacht, 116 m long, built in 1901 by Pembroke Dock shipyards in Scotland. In the USA Herreshoff shipyards built more than 200 motor boats between 1878 and 1945, the most famous of which are the steam commuter 81 ft *Mirage* (1910) built for C. Vanderbilt, and the 114 ft *Navette* (1917) built for Jack Morgan. In Europe, the Ailsa Shipyard in Scotland built the first steel superyacht *Triton* in 1902. At 55.4 m long, this vessel operated in the British Royal Navy as a Royal Patrol Yacht during World War II.

The size of superyachts changed over the years (Figure 3), with a continuous enlargement until the Second World War, peaking with the construction of *Savarona*, a 136 m yacht built by Blohm & Voss in 1936 and destined to be the biggest yacht afloat for nearly 50 years. After the end of World War II there was a sensible reduction in average yacht dimensions, with the only exception being the 125 m Royal Yacht *Britannia*, launched in 1953 by John Brown’s Shipyard in Clydebank. Only in the eighties the size of the largest yachts start to increase once more; in 1980 Benetti Shipyards launched *Nabila*, 86 m in length and, few years later, in 1984 the 144 m yacht *Abdul Aziz* built by Helsingør Værft in Denmark, became the largest yacht in the world.

Nowadays a huge number of superyachts and a relatively high number of vessels of over 50 m are built every year and the demand for these vessels and for ever increasing dimensions and opulence seems not to slow down, although the current global economic

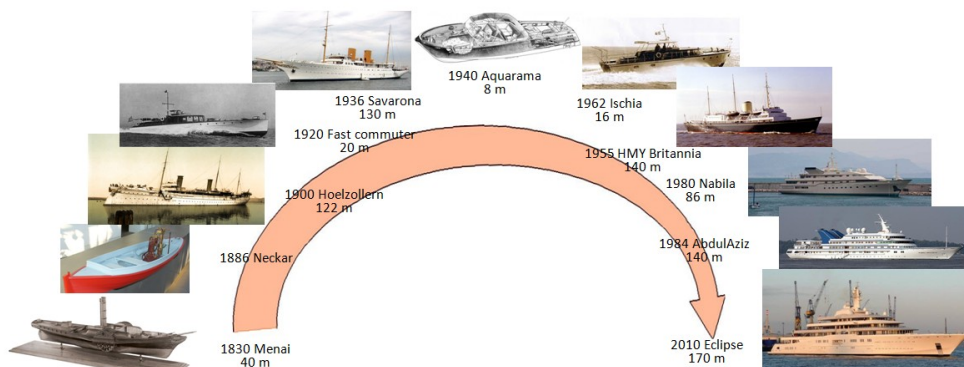


Figure 3: Development of yacht dimensions from the beginning till present days.

climate has cooled this previously rampant industry in recent years. An emblematic characteristic of superyachts is the large number of decks, giving the superstructures an imposing appearance, and very large internal spaces. The length of a superyacht is the defining characteristic, so the vessel's typology can be any of those previously described; hence, there are very large flying bridge, open and expedition yachts. At present the largest yacht in the world is M/Y *Eclipse*, at 164 m in length, delivered in 2010 by Lürssen Shipyards in Germany, and there are over 25 yachts of L_{OA} greater than 100 m.

3 RULES AND REGULATIONS

There is a wide variety of national and international rules and regulations for which motor yachts must adhere. In addition to the rules from CS, the International Maritime Organisation (IMO), National Regulations, and Port State Regulations, large motor yachts must meet the following International Conventions:

- Safety of Life at Sea (SOLAS);
- International Load Line Convention (ILLC);
- MARPOL, devoted to the control of the marine pollution;
- International Regulations for Preventing Collisions at Sea (COLREG), which provides requirements for steering and sailing, navigation lights and sound signals;
- Standards of Training, Certification and Watchkeeping (STCW).

The rule's applicability depends on yacht characteristics such as dimensions (represented mainly by load line length and gross tonnage), the type of service and the number of passengers. Yachts are subdivided into two main categories: superyachts with a freeboard length over 24 m and yachts below 24 m. While superyachts are subject to international rules, yachts below 24 m are considered differently by the various flag administrations. The reference length itself is not defined everywhere in the same way. For example in the European community, all the pleasure yachts built and commercialised in the EU with a hull length, L_H , between 2.5 and 24 m should be 'CE Marked' and comply with ISO Standard Rules. A further category of yachts below 12 m is also defined for which less stringent rules apply.

The type of service is important and can be designed and managed for a private use or a commercial use:

- private yachts are designed and managed for the personal use of the owner and should not be engaged in any kind of trade;
- commercial yachts are designed and managed in order to allow charter activity (trade). However, at times they might also be registered and managed as private yachts.

Private yachts are required to comply with MARPOL Rules, the International Tonnage Convention and COLREG. Private yachts need not to comply with the requirements of ILLC and SOLAS.

Large commercial yachts are equivalent to ships and must comply with International Conventions. Because the International Conventions have been written and issued mainly for cargo and passenger ships, in 1997 the UK Maritime and Coastguard Agency (MCA) developed the 'Code of Practice for the Safety of Large Commercial Sailing and Motor Vessels', known as the MCA Large Yacht Code (LY1), which adapted the International Conventions for yachts, allowing them to maintain their particular identity. In 2004 it was updated to Large Yacht Code 2 (LY2). Even

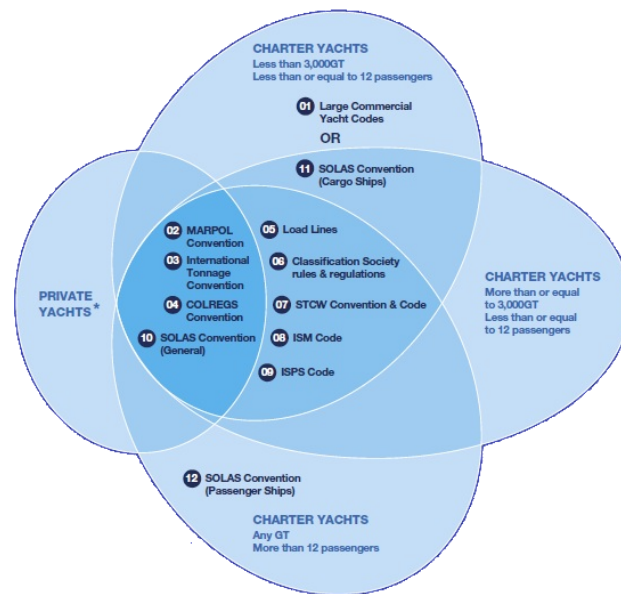


Figure 4: Schematic representation of rule requirements for private and commercial yachts (Manta Maritime, 2008).

though the LY2 is a statutory regulation for only the UK and Red Ensign flag charter yachts, LY2 has been the most frequently used Code by the industry all over the world. Fairbrother (2006) presents the main aspects of this code and highlights the most important topics. The MCA-LY2 (as LY1) recognizes American Bureau of Shipping (ABS), Bureau Veritas (BV), Det Norske Veritas (DNV), Germanischer Lloyd (GL), Lloyd's Register (LR) and Registro Italiano Navale (RINA) as the CS that have rules prescribing the required standards for construction and strength of large motor yachts. In addition, CS are authorised to carry out plan approval, surveys and issue certificates of compliance with certain parts of the MCA Large Commercial Yacht Code on behalf of the MCA, the CISR (Cayman Islands Shipping Registry), and other Red Ensign Administrations.

The Regulation environment is efficiently presented in Figure 4 (Manta Maritime, 2008) as a function of private and commercial use. Commercial yachts can be further subdivided into three main categories depending on the gross tonnage and the passenger number:

1. Commercial yachts with a freeboard length over 24 m, equal to or below 500 GT and carrying a maximum of 12 passengers should comply with MCA-LY2 or equivalent. The limit of 500 GT corresponds to a length of approximately 45 m for a normal motor yacht with standard superstructures and up to 55 m for a large sailing yacht (with small superstructures). Specific less stringent rules are considered by LY2 for 'Short Range Yachts' with less than 300 GT and a navigation limit of 60 nautical miles. Classification with one of the major CS is mandatory. Commercial yachts with a freeboard length below 24 m should comply with different codes, i.e. MCA MGN 280 the 'Code for Small Vessels in Commercial Use for Sport or Pleasure' (1997); classification is not compulsory.
2. Commercial yachts as above, but with a gross tonnage up to 3000 GT and with less than or equal to 12 passengers should comply with the MCA-LY2 as well;

in this case LY2 contains more stringent requirements about safety and arrangements in general because, from the SOLAS point of view, differences between these yachts and merchant ships are reduced. The limit of 3000 *GT* corresponds to an overall length of about 90 *m*. For this category of yachts classification is mandatory.

3. Commercial yachts above 3000 *GT* or carrying more than 12 passengers and up to 36 passengers should comply with the MCA 'Passenger Yacht Code' (2010). It applies to yachts with a maximum number of persons equal to 99, crew components included. Above this limit commercial yachts should fully comply with SOLAS Rules, without the Large Yacht Code 'smoothing interpretation'. Classification is necessary.

Further comments on the role of CS for charter yachts are contained in Cooper *et al.* (2009) and in Strachan and Lagoumidou (2009). The structural design and scantlings of any kind of yacht are regulated by CS' rules; as a matter of fact, very limited structural aspects are contained in MCA-LY2. Of the 30 sections in MCA-LY2 only one (Section 4) is relative to structures. It initially states that the purpose of this section is *to ensure that all vessels are constructed to a consistent standard in respect of strength and watertight integrity*. Concerning structural strength, LY2 reports that *all vessels must be classed*. It follows a brief discussion about watertight bulkheads and sailing yacht rigging. The remainder of MCA-LY2 contains mainly rules concerning watertight integrity, machinery, electrical installations, steering gear, bilge pumping, stability, freeboard, life saving appliances, fire safety, navigation equipments, anchoring and other issues related to protection and safety.

In the following, synopses of structural issues contained in the rules and regulations of the most important CS are presented. Details of the design loads used by the major CS are presented in Chapter 4 of this report.

The International Standards Organisation in 2005 completed the standard number 12215 'Small Craft - Hull Construction and Scantlings', mandatory for all commercial motor and sailing boats with an hull length between 2.5 and 24 *m* in the European Union. Although ISO Standards were built on the ABS 'Guide for Building and Classing of Pleasure Motor Yachts', there are a number of differences between ISO 12215 and the ABS Rules. ISO 12215 is, to some extent, a design standard more than a set of rules. Curry (2005) presents a comprehensive assessment of ISO 12215 in which all main parts are discussed, verified and compared with the principal CS' rules.

ISO 12215 is divided into 8 Parts. The first three parts are devoted to materials and they provide minimum required mechanical properties for composite single skin laminates, sandwich cores (foam and balsa), steel, aluminium and wood. Part 4 deals with workshop and manufacturing. Part 5 (2004) involves design pressure, design stresses, and scantling determination. Part 6 (2005) presents structural arrangements and details. Part 7 (at present under development) is dedicated to the scantling determination of multihulls and Part 8 is dedicated to rudder design. The sections of specific interest for structure design are ISO 12215-5 'Small Crafts, Hull construction and Scantlings' Parts 5 and 6.

In ISO standards 12215-5 (2004) motor yachts are divided into four design categories depending on the service range, wave heights and wind speed. As the standard doesn't take into account hull girder strength, the scantlings are assumed to be governed by local loads defined as sea pressures. Equations for shell thickness and reinforcement

modulus are provided for both motor and sailing craft. Shell thickness calculations depend on the construction material considered by ISO 12215 (wood, steel, aluminium, FRP single skin and sandwich). The section modulus is calculated with a unique procedure independent of the material. In all cases, equations that contain the design pressure and the material design stress σ_d , are provided by the Standards. Simplified scantling methods are provided as well in appendages for boats with hull lengths less than 12 m and sailing boats less than 9 m, design categories C and D (limited navigation) respectively. Other appendages conclude this part with very detailed specifications for material characteristics.

ISO 12215-6 (2005) deals with general structural arrangements, transverse and longitudinal structures, and structural details. Particular attention is devoted to deck and shell openings, FRP local reinforcements, hull-deck joints, steel and wood details and, finally, to rudder and keel structural arrangements and connections. A number of appendages concern glued and riveted joints with calculation procedures and application examples.

American Bureau of Shipping in 2000 published the 'Guide for Building and Classing of Motor Pleasure Yachts', which is applicable to *motor pleasure craft 24 m (79 ft) or greater in overall length up to 61 m (200 ft) in length, that are not required to be assigned a load line*. The rules are composed of 24 sections, 10 of which, from Section 3 to Section 12, are concerned with structural scantlings. Section 3 contains general definitions, such as effective width of plating and bracket standard proportions. In Section 4, mechanical properties of materials are defined in detail; steel, aluminium alloys, FRP and wood are all considered. In Sections 6 and 7, structural arrangements, details and fastenings are presented for all the materials considered by the rules. As for design loads, considered in Section 8, hull scantlings are considered separately for high speed craft and displacement craft. Section 9 deals with high speed craft, defined as craft having a maximum speed in knots not less than $2.36L^{0.5}$ where L is the scantling length in metres. Minimum thicknesses for plating and minimum section modulus for internals are defined for steel, aluminium, FRP and wood. Formulas for minimum thickness of shells and minimum section modulus of reinforcements are provided as a function of design pressure, material design stress, stiffener span and spacing. The same formulation is assumed in Section 10 for hull scantlings of displacement craft. The special structure of stem and stern frames, keels, shaft and rudders are considered as well by the ABS Rules in Sections 12 and 13.

Bureau Veritas 'Rules for the Classification and Certification of Yachts' (2012) applies to ships intended for pleasure or commercial cruising and with a length not exceeding 100 m. A lower limit on length is not mentioned, but it is stated that European flagged craft less than 24 m must meet the EC directive. BV Rules are applicable to sailing and motor vessels of monohull and catamaran type, built in steel, aluminium, wood and composite materials. Rules are organized in three parts: Part A is related to Classification and Surveys, Part B to Hull and Stability, and Part C deals with Machinery, Electricity, Automation and Fire Protection.

Structure scantlings are considered in Part B, from Chapter 4 to Chapter 8. Scantling requirements are influenced by the navigation notation ('n' coefficient). Design loads are provided in terms of overall global loads and local loads, both static and dynamic, in Chapters 4 to 7.

Plating and stiffener scantlings are assessed in Chapter 6 for steel and aluminium and in Chapter 7 for composite and plywood vessels. Minimum thicknesses for plating

are defined as a function of geometric characteristics of the panel (aspect ratio and smaller side), design pressure and material admissible stress. In the case of stiffeners, scantlings are sized with regards to a minimum section modulus and a minimum shear area given by formulas containing design pressure, material admissible stress, span and spacing of the stiffeners. For both plating and stiffeners, a procedure for a buckling check is presented. The BV rules also provide general considerations for structural layout and construction details of bottom, side, deck and superstructures areas.

Det Norske Veritas Regulations for motor yachts are included in the 'High Speed, Light Craft and Naval Surface Craft' Rules (2011) which consist of 8 Parts; Parts 0 and 1 contain general regulations, Part 2 metallic materials, welding and composites, Part 3 structure and equipment requirements, Part 4 machinery and systems/equipment and operation, Part 5 special service and type, Part 6 special equipments and systems, Part 7 HSLC in operation.

Yachts are then considered as a 'type' among other special service vessels and the rules apply to yachts over 24 m in length, *not intended for operation on a commercial basis, i.e. that the operation of the craft is being financed by others than those on board*. The following classes are defined:

- ✱ 1 A1 LC Yacht: when the displacement fully loaded is not more than $(0,20 \cdot L \cdot B)^{1,5}$. Vessels with a larger displacement can be assigned the notation based on special considerations;
- ✱ 1 A1 HSLC Yacht: when the displacement fully loaded is not more than $(0,16 \cdot L \cdot B)^{1,5}$ and its maximum speed exceeds $3L^{0,5}$.

No mention is made of length, except for the note that recreational boats less than 24 m may have to comply with the European Union Directive for CE marking. There are also no specific stipulations concerning loadings in this part, and in this respect yachts are considered with other craft in Part 3, Chapter 1.

Structural scantlings are dealt with mainly in Parts 2 and 3 which are dedicated to the hull structural design of steel and aluminium yachts; the general outlines for each chapter are very similar. Design loads, in Chapter 1 of Part 3, are subdivided into local and global loads. After a detailed description of bottom, side, deck, bulkhead and superstructure layouts, common design rules for most important details are presented. Material and welding characteristics, provided in the following sections, should be integrated according to Part 2 specifically dedicated to materials. Hull structure scantlings starts with the verification of a minimum hull section modulus. Plating minimum thicknesses are given by simple formulas containing, as usual, design pressures and stiffener spacing. Reinforcement scantlings are considered by DNV separately for secondary stiffeners and primary web frames and girders. In both cases the minimum section modulus should be calculated by formulas as a function of span, spacing, design pressure and allowable material stress. The rules also give a procedure for buckling control of plating, stiffeners, stiffened panels and girders.

Chapter 4 deals with composite hull structures; requirements about material manufacturing procedures and main characteristics are presented. This part should be integrated by other requirements contained in Chapter 4 of Part 2. The scantlings of FRP single skin construction is based on a minimum glass weight per square metre given by a table as a function of structural member and hull position. As the minimum content of fibres by volume is fixed by DNV at 25 %, the corresponding thickness comes accordingly as a function of the utilised glass fabrics. A specific section is dedicated to sandwich panels. FRP reinforcement scantlings are based on a direct approach

starting from the definition of a maximum bending moment (calculated as a function of span, spacing and design pressure) and a subsequent verification of the cross section modulus as a function of the design stress of the material.

Germanischer Lloyd classifies motor and sailing yachts in Part 3 of their rules for 'Special Craft'. Chapter 2 (2003) applies to motor and sailing yachts with a scantling length greater than 24 m for private, recreational use. Chapter 3 (2003) is related to motor and sailing yachts with a length between 6 and 24 m for private use. GL specifies that Rules for Special Craft were developed considering that yachts, with respect to merchant ships, are usually subjected to:

- less severe operating conditions than for ships in regular trade;
- limited yearly sea hours in relation to harbour hours;
- special care by the owner and usually good maintenance.

Two categories of yachts are considered: yachts with scantling lengths between 24 and 48 m and yachts over 48 m. In the first part of Chapter 2 the first category is assessed. Normal and high strength steel, aluminium and wood are covered by this section. For FRP and core materials reference should be made to a specific part of the GL Rules II – Materials and Welding, Part 2 – Non-metallic Materials, Chapter 1 – Fibre Reinforced Plastics and Adhesive Joints.

In Section 2.C a list of general criteria are provided in detail regarding, as an example, curved panel and girder correction factors, reinforcement span definition, effective width of plating, buckling evaluation criteria and others. Section 2.D is devoted to steel and aluminium structures. Design loads are defined as a function of a vessel's speed. Minimum plating thickness of hull, decks, superstructures, bulkheads and tanks is calculated by a unique formula containing design pressure, permissible stress of the material, dimension parameters and a correction factor for curved panels; an additional corrosion allowance is considered as well. In the same way the minimum section modulus of stiffening members is provided by a unique formula for stiffeners, frames, floors, beams and girders. The formula contains the usual parameters such as span, spacing, design pressure and material permissible stress. Pillar scantling and buckling verification is considered separately in a specific section.

Composite material hulls are considered in Section 2.E. For composite hull design loads, the same criteria as steel vessels are assumed. Plating and stiffener scantlings follow a different approach being based on classic beam/plate and laminate theory. Wooden yachts are briefly discussed in Section 2.F; the structural scantlings should comply with GL 'Rules for Classification and Construction of Wooden Seagoing Ships'.

Section 2.G deals with motor and sailing yachts with lengths exceeding 48 m and with steel and aluminium structures. Again, for high speed vessels reference should be made to GL HSC code (Part 3 - Special Craft, Chapter 1 - High Speed Craft, 2012). In the case of moderate speeds, scantlings should comply with the GL Rules Part 1 – Seagoing Ships, Chapter 1 – Hull Structure (2012).

As already pointed out, GL Rules have a specific section (GL, Special Craft – Yachts and Boats up to 24 m) for pleasure craft with length between 6 and 24 m. Also commercial vessels can be considered by these rules with certain add-on-factors taken into account. The chapter contains its own general rules and definitions mainly addressed to FRP construction and a detailed description of the material mechanical properties by means of empirical formulas and tables based on the laminate glass content by weight. The scantlings of plating are given in terms of glass weight (in g/m^2) of shells

(keel, bottom and side) by formulas as a function of stiffener spacing, design pressure and speed correction factors. Minimum section moduli are provided for transverse and longitudinal reinforcements by formulas containing reinforcement span, spacing, design pressures and speed correction factors.

Lloyd's Register of Shipping Rules for motor yachts are contained in the 'Rules and Regulations for the Classification of Special Service Craft' (2011). According to LR definition *a yacht is a recreational craft used for sport or pleasure and may be propelled mechanically, by sail or by a combination of both*. The rules are applicable to high speed craft, light displacement craft, multi-hull craft (both motor or sailing) constructed from steel, aluminium alloy, composite materials with an overall length between 24 and 150 m. The rules are composed of 17 Parts and the hull scantlings are assessed in Parts from 3 to 8. Part 3 introduces structural definitions and nomenclature, building tolerances and limits for geometrical defects due to welding for steel and aluminium. Some basic principles about structural continuity, fore and aft arrangements, bulkhead distribution and structure, and properties of beam sections are covered as well. At the end a comprehensive assessment of rudders, shaft brackets and other outfit components are presented. Part 4 reports additional information for yachts regarding water-sport platforms and shell openings, deck safety equipment, portlights and windows, protection of openings, corrosion protection, intact and damaged stability together with some special rules for sailing yachts.

Part 5 opens the structural scantling section with the definition of design load criteria. Parts 6, 7 and 8 contain scantling procedures for steel, aluminium and FRP vessel respectively. Parts 6 and 7 have the same lay out; in particular minimum plating thickness and stiffener modulus are governed by same equations as a function of design pressure, minimum yield strength of the material and usual geometrical parameters (stiffener spacing and span, panel aspect ratio etc.). In both parts, a table with minimum thickness requirements for different hull locations and vessel typologies is provided as a function of the material coefficient and of the yacht length. FRP is discussed in Part 8 because different approaches are necessary due to the very different nature of the material. The section provides mechanical properties of laminates as a function of glass content and reinforcement type (mat, woven roving, cross lied and unidirectional) together with nominal thickness of a single ply. The minimum plate thickness is defined by formulas as a function of service factor depending on service type notation, while the minimum thickness of laminate for both stiffener and laminated components are based on an assumed fibre content $f_c = 0.5$. The final chapters of all three Parts 6, 7 and 8 are dedicated to hull girder strength for mono and multi-hull and failure mode control. This last section provides criteria to evaluate deflection, stresses, buckling and vibrations.

Registro Italiano Navale Rules for yachts are published in two versions: 'Rules for the Classifications of Pleasure Yachts' (2011), which applies to yachts *engaged in private use of a length of 16 m and over* and 'Rules for the Classification of Yachts Designed for Commercial Use' (2011a) addressed to commercial vessels with length of 24 m and over. The two versions have an identical formulation, divided into five parts: Part A 'Classification and Surveys', Part B 'Hull and Stability', Part C 'Machinery, Electrical Installation, Fire Protection', Part D 'Materials and Welding', Part E 'Additional Class Notations'. Part B, 'Hull and Stability', contains design loads and scantling criteria for yachts made of steel, aluminium, FRP and wood.

Chapter 1 deals with general definitions, outfitting, equipment, tanks, loads and rudders. The loads are subdivided into overall global loads and local loads, both static

and dynamic. For each construction material RINA Rules give a specific chapter. The rules are valid for steel vessels up to 120 *m* in length and aluminium vessels up to 90 *m* in length. For yachts with greater lengths reference is to be made to RINA Rules for the Classification of Ships. For steel and aluminium vessels (Chapter 2 and 3 respectively) a comprehensive treatment of mechanical characteristics of materials and welding procedures is presented. Design stresses and buckling criteria are defined together with many joint and construction details and reinforcements. Plating and internal scantlings are provided for bottom, sides, decks, bulkheads and superstructures. Minimum plating thicknesses are calculated by a couple of formulas as a function of design pressure, stiffener spacing and material coefficients. For reinforcements, the minimum section modulus is calculated by formulas depending on the usual parameters such as design pressure, reinforcement span and spacing and material coefficients. Different sets of formulas are available for transverse and longitudinal hull structure.

Chapter 4 is devoted to the mechanical characteristics of composites with different types of reinforcements, resins and core materials for sandwich technology. A table with formulas for determining mechanical characteristics of FRP as a function of glass content in weight is provided. Plating and reinforcement scantling procedures, valid for monohull vessels up to 40 *m* and catamarans up to 35 *m* in length, are similar to those already presented about steel with, in addition, a specific section about sandwich structure scantling. Structural adhesives are considered as well at the end of this chapter.

Besides the six classification societies accepted by MCA LY-2 there are other CS taking into consideration motor yachts in their rules. *Korean Register (KR)* Rules and guidance for yachts in general can be found in 'Guidance for Marine Leisure Ships' (2011) which replaces 'Rules for the Classification of FRP Yachts' (2010). This guidance is applicable to leisure boats and yachts of lengths between 2.5 *m* and 24 *m* in mono-hull, catamaran and trimaran hull types. Steel, aluminium-alloy, wood and FRP are considered as construction materials and design pressures are detailed for ships with/without sails respectively in this guidance. Other issues about yacht structures are contained in 'Rules for the Classification of FRP Ships' (2011) and 'Rules for the Classification of Steel Ships, Part 10: Hull Structure and Equipment of Small Steel Ships' (2011).

Hellenic Register of Shipping (HRS) Regulations for motor yachts are contained in 'Rules and Regulations for the Classification and Construction of Small Craft' (2004) applicable to wooden boats up to 36 *m* in length, steel and aluminium vessels up to 60 *m*.

Nippon Kaiji Kyokai (NKK) Rules do not specifically take into consideration motor yachts. Guidance for yacht structure scantlings can be found in the 'Rules for the Survey and Construction of Ships of Fibreglass Reinforced Plastics' (2011), 'Rules for High Speed Craft' (2011) and 'Rules for the Survey and Construction of Steel Ships, Part CS: Hull Construction and Equipment of Small Ships' (2011).

4 DESIGN LOADS AND ASSESSMENT METHODOLOGIES

As described in the Chapter 2, the term 'motor yacht' covers practically the entire range of possible vessel types, from 10 *m* to over 160 *m* megayachts. Hence, there is a correspondingly large diversity in the relevant important structural loads and how they are estimated, depending on the size, type, speed, displacement or planing regime etc of vessel considered.

There is little literature concerning the loads on motor yachts specifically, since in terms of loads the fact that the vessel is a yacht often does not significantly change the loads to which it will be subjected to, with respect to conventional ships. Also, most of the research carried out concerning yachts is of a commercially sensitive nature and hence is not published. However, there are a few helpful references directly concerning yachts, and these will first be briefly described below. Following this, other work concerning the loads on high speed craft or ships, and which are also applicable to motor yachts will be outlined.

In the final section a brief description of the current rules directly applicable to motor yachts in terms of loadings will be made. However, it must be remembered that, especially for larger yachts, reference to the relevant 'Ship' or 'High Speed Craft' rules may also be required, but these rules fall outside the scope of this report, and hence are not described here (except where specific reference are made to 'yachts').

4.1 Loads on Motor Yacht

An effective subdivision of the overall loads on a motor yacht is reported by Verbaas and van der Werff (2002). They consider primary loads acting on the hull girder as a whole, secondary loads acting on large components such as decks and bulkheads and tertiary loads which affect local areas only. Primary loads consist of still water and wave induced bending moments, and torsion moments together with related shear forces. Rigging loads, in the case of sailing yachts and the loads derived from haulage operations can also be considered as primary loads. Secondary and tertiary loads, most important for the local strength evaluation, are represented by bottom and bow flare slamming loads, green sea loading and cross deck slamming for multi hull vessels. Impact loads against floating objects, or grounding loads belong to the same class of loads. As far as thermal loads are concerned, their classification depends on the extension of the area over which they apply. In the same paper the authors caution that other loads such as cargo loads and sloshing loads should not be neglected.

As stated by Marchant (1994), for smaller yachts with a length of less than 35 m, the structure is dominated by secondary and tertiary loads, particularly bottom and bow flare slamming, caused by the planing regime in which this type of vessels often operates. In the case of larger vessels primary loads, although combined with local loads, become predominant.

In fact, the length at which global loads become important for displacement, steel motor yachts is estimated at between 50 and 90 m dependent on vessel type and usage (Roy *et al.*, 2008). A very practical and comprehensive guide of how to identify the point (in terms of vessel size) at which global buckling loads should be considered for FRP motor yachts is given by Loscombe (2001).

The global loads which become significant for larger yachts are not significantly different from those acting on ships from a structural point of view, explaining why the literature concerning global loads on large motor yachts specifically doesn't exist. In fact, as highlighted by Roy (2006), owing to the continuous increase in average yacht size the trend in this regard is to employ design and construction technologies already developed in the commercial shipping industry. Hence, the literature found concerning loads on motor yachts specifically almost exclusively concerns tertiary loads, of which most are hydrodynamic loads.

The basis of planing theory and local pressure estimation for high-speed craft has been very well documented elsewhere. The classical works of Von Karman (1929)

and Wagner (1932) on water impact problems in the early twentieth century provided the background for later studies, such as that of Du Cane (1956), Heller and Jasper (1961), Savitsky (1964), Savitsky and Brown (1976), Allen and Jones (1978) leading to practical prediction methods that could be used by designers of high-speed craft to determine impact loads. In these works, mainly addressed to fast, small size motor boats, design loads are provided as slamming pressures derived by vertical acceleration measured on real scale tests. Assuming the boat dynamic behaviour like that of a rigid body, the longitudinal and transverse distribution of the vertical acceleration is calculated with respect to the centre of gravity maximum acceleration. It is then possible to determine the local pressure to be applied to the structural elements of bottom and sides in whatever position with respect to the centre of gravity.

Kaplan (1992) presented a comprehensive review of the state of the art of load calculation methodologies relevant to small and fast boats. Koelbel (1995) in his paper describes the materials used for fast boat construction and presents a complete history of structural design where all the reference theories for load calculation are listed and an alternative method for calculating design acceleration is suggested. A more practical approach to structural design of fast motor craft is given in Koelbel (2001). An assessment of planing theory for smaller craft in general is comprehensively described in many books such as those by Du Cane (1974) and Payne (1988).

The tension-compression, bending and shear loads on a 5.70m motor boat were obtained through full-size drop tests by Baur *et al.* (2004) in order to evaluate the response of the adhesively bonded construction used. The obtained data was to be used to improve laboratory simulation of service loadings of boat structures.

Rees *et al.* (2001) describe the development of a finite element code (HydroDYNA) which couples hydrodynamic and structural models in order to predict motion histories and wave slam loadings, and its application to FEM modelling of fast motor boats, and specifically to an RNLI Trent Class Lifeboat.

Santini *et al.* (2007) describe a method for optimizing hull structural design based on desired performance characteristics and expected operator manoeuvring profiles. They analyse the dynamic and transient nature of the hydrodynamic slamming of a small planing boat during drop simulations using an FSI (Fluid Structure Interaction) methodology. Slamming loads are then converted into static equivalent linear loads and input into the topology optimization software OPTISTRUCT[®], developed by Altair Engineering Inc. The software, based on the finite element method, generates the best structure lay out given a package space, loads, boundary conditions and a target weight.

The problem of slamming specifically for composite ships and yachts is considered by Meijer (1996), where it is stated that whilst the approach of using extrapolated experience more than first principles for steel ships may be satisfactory, '*composite hulls at high speeds are a completely different matter*'. The impact event may produce dynamic global loads in the hull girder - bending and torsional moments and shear forces, both transient (*whipping*) and continuous (*springing*) - which are normally only considered for larger ships. However, Meijer notes that these effects may become significant for smaller craft of relatively flexible FRP. The paper itself considers only local effects induced by slamming. The importance of resistance to solid object impacts is again noted, and special caution is advised if considering carbon composites.

Lalangas and Yannoulis (1983), noting some uncertainties in existing methods for predicting the design bottom pressure, pressure reduction factors and safety factors,

proposed a procedure to calculate the bottom design pressure for a 20 *m* high-speed aluminium motor yacht. He concludes that the bottom structural design methods used were satisfactory as no failures occurred after two summer seasons of use for all four yachts.

A simplified model as a practical design tool for the time dependent calculation of slamming pressures for composite yacht panels has been developed at SP-systems (Manganelli and Hobbs 2006, Loarn and Manganelli 2010). Hull curvature effects are included, and the model was found to be in good general agreement with experimental results. However, since dynamic and hydro-elastic effects were neglected, limitations to the 2D quasi-static approach were noted, and it was thought that the range of applicability will decrease for higher impact velocities.

Most of the CS' rules are based on the centre of gravity acceleration; the determination of this parameter by direct methods such as towing tank or full-scale tests is not so simple, especially when the yacht is large and operates at high speed. For such cases Hueber and Caponnetto (2009) present applications of CFD to superyacht design with particular reference to seakeeping computation for high speed vessels. The numerical approach is able to simulate the bow impact on waves in a heavy sea (despite the statement that the method needs to be refined). Two methods are considered, the first using a rigid motion of the hull mesh, and the other taking into account a smooth deformation of the mesh at impact. A useful time history of the vertical acceleration for two hulls with different dead rise angles is presented.

A comparison of midship plating design pressure calculated by several different methodologies is presented by Schleicher *et al.* (2003) as part of a feasibility study of an hypothetical 100 *knot*, 46 *m* superyacht. Pressures are calculated by ABS, Lloyd's and DNV Rules and by direct methods such as those by Koelbel (2001), Silvia (1978), Allen and Jones (1978), Heller and Jasper (1961), and Henrickson and Spencer (1982). The more conservative values are obtained using Koelbel, the lower ones from Henrickson and Spencer; with a difference of 400 % between the two. The values using the rules fall between these two extremes, at slightly higher than the average of the two.

A comprehensive series of motion and load measurements on an 18 *m* FRP motor yacht is presented by Carrera and Rizzo (2005). The trials were particularly aimed at studying the structural behaviour of the fore part of the hull structure, which is subject to impact phenomena. The authors describe the equipments and instrumentation utilised for the tests and the attained results. The signals from pressure sensors installed on the fore part of the bottom were recorded simultaneously with signals from accelerometers and strain gauges. As well as conventional accelerometers and rate-gyro sensors, a GPS-RTK system was installed for real time monitoring of the craft motions in six-degree of freedom. Tests were carried out for different sea conditions and headings. It is worth noting that the vertical acceleration of 1g, suggested by most CS' rules as a reference value for structure scantling, was exceeded more than once.

A systematic approach to the evaluation of design loads on rudders for high-performance boats and yachts is described by Blount and Dawson (2002), where practical methods for the evaluation of side-force, drag and torque loads are detailed.

A common structural issue (which arises from the fact that the use of a yacht is for 'pleasure') is that of surface finish/plate flatness due to temperature differentials between air-conditioned interiors and hot exteriors. That was often a fairly severe

problem and it became evident through discussions with shipyards and CS, but no work on this aspect has been found in the literature.

Glass structures are increasingly becoming en vogue, leading to larger glazed areas that are susceptible to wave impact or green water loads. Design loads on yacht glazing have been traditionally regulated by standards and conventions which are essentially a mix of a lot of empiricism and tradition with little science and hence a new standard, ISO/DIS 11336-1 (Verbaas and van der Werff, 2002) is under development in order to try to rectify this. Since glass is a brittle material, strength tests traditionally give a wide range of scatter. The existing approach therefore is to use ample safety factors that are incorporated into the design pressures, and for traditional glazing consisting of only small areas this was a prudent and workable approach. However, for larger glass structures the problem becomes more critical and the new standard aims to develop tests and test procedures to better define and control the variability in the glass properties in order to be able to reduce the ample 'hidden' safety factors included in the design pressures. Further to be considered, the properties of mounting methods which are of paramount importance for brittle materials since there is no local 'give' in the material.

An important issue for superyachts which is impacting increasingly on structural design, is the rising demand by owners for facilities to allow helicopter landings. This implies the space availability to install a platform of proper dimensions and structural strength to support the dynamic landing load and, as a consequence, strictly depends on the yacht size. In fact, just because of space restrictions, installing a heli-deck on yachts under 70 m overall length is not practical. At present LY2 references SOLAS II-2 and ICAO Annex 14 of the Convention on International Civil Aviation for requirements for helicopter operations. In recognition of the increase in demand for providing helicopter facilities on board yachts, the MCA has established an advisory group to investigate and formulate requirements for these arrangement. These requirements, when accepted, will presumably be incorporated into LY2. On this subject some CS, such as Lloyd's Register and ABS are developing their own rules.

4.2 General Loads Publications also Applicable to Motor Yachts

Since it already provides the most comprehensive source of information on ship structural issues, one of the best sources of information for the many different types of loads that may be applicable to yachts is that of previous ISSC reports. For larger, displacement motor yachts, the various 'Loads' reports will be most informative, whilst for faster, usually smaller, semi-planing and planing motor yachts the 'Weight Critical Structures' and 'Design of High Speed Vessel' reports are extremely relevant. Table 1 summarises where the various relevant different info can be found.

It is not possible to comprehensively cover all recent work relevant to high-speed craft here, but some relevant publications are now mentioned. Books for a comprehensive overview of the subject have been written by Faltinsen (2005), and Lewandowski (2005).

The ABS 'Guidance notes on structural Direct Analysis for High-Speed Craft' (ABS, 2011) provides instructions for the 'first principles' evaluation of loading conditions and load cases, wave loads, external pressures, slamming loads, internal tank pressures, and acceleration and motion-induced loads. Guidance is also given with respect to the loadings used for finite element modelling. Kim *et al.* (2008), discuss recent developments at ABS to revise the requirements for slamming loads on high speed naval craft.

Table 1: Committee reports detailing loads in previous ISSC reports

Type of loads	1991	1994	1997	2000	2003	2006	2009
Global Loads:							
Hydrostatic loads	L	L8					
Wave Loads		L2	L2	L2	L2	L2.1, L2.2	L2, L3.4
Wind Loads		L6	L5	L2	L2	L2.7	
Ice Loads		L7				L2.6	L3.3
Fatigue							L7, IPL3
Local loads:							
Slamming		L4	L4	L 4.1	L4.1	L4.1	IPL2
Green water		L4		L4.2	L4.2	L4.3	IPL5
Sloshing		L5	L4	L4.3	L4.3	L4.2	L3.5, IPL4
Object Impact		WCS3.3		HSV6	HSV5.5		
Collision & Grounding			CG3		CG3, CG4	CG2	
Model & Full Scale Tests:					L6	L2.3	L3, L5, IPL2, IPL4
Probabilistic/ Uncertainty modelling			L6		L5, L6	L5, L6, CG3, CG4	L5, L6
High Speed Craft		WCS2.1		HSV4.3, HSV5, HSV9	HSV2.2, HSV3	L2.4	

L: Loads, IPL: Impulsive Pressure Loading, WCS: Weight Critical Structures, HSV: High Speed Vessels, CG: Collision and Grounding. Number refers to report Section.

As part of the ‘Comparative Structural Requirements for High Speed Craft’ the SSC-439 Ship Structure Committee (2005) compare the calculations of design loads (vertical acceleration and design pressures) made by the relevant societies (IMO, ABS, DNV, UNITAS, LR and NK).

The ‘Hydrodynamic Pressures and Impact Loads for High Speed Catamaran / SES Hull Forms’ is the subject of the report of another Ship Structure Committee (Vorus, 2007), and illustrative sample unsteady hull pressure distributions on a 10 m bi-hull SES are given in Vorus and Sedat (2007).

Slamming loads on large yachts must be considered with care, especially for high speed vessels; the paper by Dessi and Ciappi (2010) presents a comparative analysis of slamming events and induced whipping vertical bending moment carried out on data collected with towing tank tests using segmented flexible models to allow a correlation between slamming and whipping response. Despite the fact that the work is relative to a passenger ship and a fast ferry, the results relevant to the latter case study can be utilised for large yachts, where the speed, dimensions, and the hard-chine hull form are very similar.

Much recent work concerning loads on planing craft, especially with respect to wave loads and slamming has been carried out at KTH Royal Institute of Technology Naval Architecture (Rosén 2004, 2010; Burman *et al.*, 2010; Garne *et al.*, 2010).

Finally, the goal of the ongoing Ship Structure Committee (SR-1470, not yet available) concerning the ‘Structural Load Prediction for High Speed Planing Craft’ is *to develop*

and verify a practical method to use time domain simulation to drive structural design of high speed planing craft.

4.3 Motor Yacht Loads in Rules

In terms of rules, the most important demarcation is that between ‘small’ and ‘large’ yachts and generally (although not exclusively) a length of 24 m is the value taken as the limit between the two definitions of size.

4.3.1 Superyachts

As described in Chapter 3 the ‘industry standard’ for large yachts is the MCA LY2 and in terms of the load assessment, the relevant information here is simply that, for unlimited operation, all vessels must be classed by any of the six CS listed in Chapter 4. Classification may be requested as a ‘Yacht’, a ‘High Speed Craft’ or a ‘Ship’, but in the present treatment only the CS’ rules where specific reference to a ‘yacht’ is made will be considered. In the following subsections the relevant rules of each CS are briefly outlined in terms of how they define, assess and allow for the definition, calculation and application of the various design loads. As a matter of fact the equations used are generally semi-empirical in nature, with bottom pressure calculations for fast craft usually based on the approach of Heller and Jasper and Allen and Jones (Marchant, 1994).

American Bureau of Shipping (2000) refers to design loads in Section 8 in terms of design pressures, where they are considered separately for semi-planing and planing crafts and for displacement vessels. For fast vessels hydrodynamic and static pressures are defined for the bottom, side, decks and bulkheads. Hydrodynamic pressure on the bottom structure is provided by a formula containing (besides displacement, length and breadth) vessel speed, deadrise angle and a service dynamic factor representative of the acceleration at the centre of gravity. The vessel location is accounted for by a vertical acceleration distribution factor. Static pressure depends on moulded depth only.

For displacement craft with a maximum speed in knots of less than $2.36 \cdot L^{0.5}$ (L in metres) the design heads for bottom, sides, decks, deep tanks, watertight bulkheads, superstructures and deckhouses are given in a table in Section 8.3.

Hydrofoils, air cushion vehicles, surface effect craft, and multihull vessels are considered in Section 8.5. The design pressures for shell, bulkheads and decks are to be not less than those for semi-planing and planing craft. This section also states that, *Design calculations for the external design pressures due to sea loading for the various operational modes and for structures peculiar to the vessel type such a hydrofoil struts and foils etc, are to be submitted to ABS offices for review.*

The ‘Design Pressures’ are then used to give the hull scantlings for ‘High Speed Craft’ (max speed, in knots not less than $2.36 \cdot L^{0.5}$, L in metres), and the ‘Design Heads’ used to give displacement craft hull scantlings.

In Section 11 ABS Rules take into consideration a minimum hull girder section modulus at amidships varying with length, breadth and block coefficient. This formula applies to yachts for which the beam of the vessel is not to be greater than twice the depth. If the yacht speed exceeds 25 knots (‘High Speed Yachts’) an additional formula has to be applied in which the displacement and vertical acceleration at centre of gravity and at the forward end are considered.

For yachts aiming at sailing in arctic waters, a specific ‘Ice Class Yachts’ has been introduced by ABS. This Class takes into account different ice characteristics, such as

ice cover, age and expected thickness, and type of navigation (independent or escorted by ice breaker). From the structural point of view ice navigation requires an increase in thickness for plates straddling the waterline, reduced frame distances and special material grades. The structural component must be dimensioned by a design ice pressure calculated as a function of vessel displacement, installed power, geographical position and hull shape.

Bureau Veritas (2012) design loads are provided in Part B, and in Chapter 4 an helpful table synthesises the assessed kinds of loads and where each of these may be found in the rules. Such loads are not to be amplified by any safety factor, this being already considered in admissible stress levels given in detail for each material in the relevant section. Rules also states that the wave induced and dynamic loads defined correspond to an operating life of the vessel of 20 years.

Vertical accelerations resulting in slamming phenomenon on the bottom area are dealt with in Part B, Chapter 4, Section 3 for high speed motor yachts ($V[kn] \geq 7.16 \cdot \Delta^{1/6} [t]$). These should be defined using the designer's model or full-scale tests, or lacking this via an apparently semi-empirical generalised equation. In the case that the designer does not provide the vertical acceleration, a simple formula dependant on length, the type of motor yacht (Cruise, Sport or with specific equipments) and the navigation zone is stated. Maximum admissible accelerations are also stipulated. For slow speed motor yachts no acceleration calculations are required.

As far as global loads are concerned, in Chapter 5 for steel and aluminium and in Chapter 7 for composite vessels still water and wave bending moment and shear forces are calculated as a function of hull dimensions, block coefficient and wave length and height, but only when one of the following situations occurs:

- length greater than 40 m;
- sailing yachts with significant mast compression or rigging loads;
- large deck openings or significant geometrical discontinuities at bottom or decks;
- transverse framing;
- decks with thin plating and widely spaced secondary stiffeners.

For multihull vessels a formula to determine the wave torque moment in a quartering sea is also provided. The manner in which the global loads should be combined is described in Chapter 5, Section 2 depending on whether the yacht is motor or sail, and mono- or multi-hull.

Local loads are defined in Chapter 4 Section 3 as hydrodynamic loads and bottom slamming loads. Hydrodynamic loads are represented by a sea pressure which is a combination of hydrostatic pressure and the pressure induced by waves. Sea pressure on the bottom and side shell is provided as a function of the navigation coefficient 'n', the full load draught, the wave height and a wave load coefficient X_i depending on the longitudinal location and on the type of yacht. The hull is longitudinally subdivided into 4 areas, for which the X_i coefficient has an increasing value from aft to stern. Impact pressure (wave impact load, distributed as a water column of 0.6 m diameter) on the side shell is also calculated both for monohulls and catamarans. Sea pressure on decks is provided by tables for exposed decks, accomodation decks and superstructures decks.

In the same chapter the bottom slamming pressures for high speed motor yachts of both mono and multihull type are given as a function of the design vertical acceleration a_{cg} (defined in Chapter 4) by a relationship containing the significant wave height, hull deadrise, ship speed and other geometrical characteristics of the vessel.

Det Norske Veritas loads for 'High Speed, Light Craft and Naval Surface Craft' (2011) are assessed in Chapter 1 of Part 3 for both HSLC yachts (with speed greater than 25 *knots*) and LC yachts (speed less than 25 *knots*). Loads are subdivided into local loads, represented by slamming pressures and sea pressures, and global loads.

To calculate slamming pressures formulas are provided to determine vertical and horizontal acceleration. Design vertical acceleration (at the centre of gravity) is calculated relative to yacht length, speed and an acceleration factor (fraction of g_0) defined as a function of type and service notation, and service area restriction notation. Horizontal accelerations, both longitudinal and transversal, are also provided as a function of the same parameters defining vertical acceleration.

Dynamic pressures on the bottom, forebody sides, bow and flat cross structures are then calculated by formulas containing, besides the design acceleration, a longitudinal distribution factor, the yacht displacement and the number of hulls, unsupported panel areas, draft, maximum design vertical acceleration and deadrise angle along the hull. Sea pressures acting on the craft's bottom, side and weather decks are calculated separately for load points below and above the design waterline as a function of the vertical distance from the waterline to the considered load point, yacht draught and a wave coefficient. The pressures from liquids in tanks and the loads from dry cargoes, stores and equipment and heavy units are also taken into account.

As for other CS' rules, hull girder global loads considered by DNV consist of hogging and sagging bending moments and shear forces expressed as a function of yacht dimensions and wave coefficient. For twin hull vessels the loads on the transverse connecting structures are also addressed: vertical, transverse and pitch connecting moments are provided by formulas containing displacement and design accelerations at centre of gravity.

In the *Germanischer Lloyd's* Rules for 'Special Craft' (2003), design loads for steel and aluminium yachts of less than 48 *m* are contained in Section 2.D for speeds lower than $7.2 \cdot \nabla^{1/6}$ (where ∇ is the moulded volume in m^3). Design pressures are calculated on hull, weather decks, superstructure and deckhouses, accommodation decks, bulkheads and tank structures. As an example, the hull pressure formula contains the ship scantling length, draught, deadrise angle, panel span and size factors, hull longitudinal distribution factor and range of service. Design pressures on decks and superstructures are determined by similar formulas but with less parameters. For speeds higher than $7.2 \cdot \nabla^{1/6}$ yachts are considered 'high speed' motor yachts and for design loads and scantling requirements reference should be made to the High Speed Craft code (GL Rules Part 1 – Seagoing Ships, Chapter 5 – High Speed Craft). The loads for yachts of less than 48 *m* constructed of composite materials are given in Section 2.E with the same philosophy applied as for steel yachts.

Steel and aluminium yachts with length greater than 48 *m* are briefly considered in Section 2.G, in the sense that it states that for speeds higher than $7.2 \cdot \nabla^{1/6}$ reference should be made to GL Rules Part 1 – Seagoing Ships, Chapter 5 – High Speed Craft, Section 3. For lower speeds GL Rules Part 1 – Seagoing Ships, Chapter 1 – Hull Structures should be applied.

Lloyd's Register (2011) design load criteria are considered in Part 5 of the SSC Rules. Generally the cases of displacement and non-displacement, and mono-hull and multi-hull are considered separately throughout. Chapter 1 states that '*load and design criteria are to be supplemented by direct calculation methods incorporating model tests*

and numerical analysis for novel designs', and details on the allowable direct calculations and instructions for model experiments are then given in Sections 2 and 3 respectively.

The LR philosophy consists of considering local strength and global strength according to the 'rule length', L_R of the vessel (L_R being between 96 and 97 % of the waterline length) as follows:

- for vessels with a rule length of less than 50 m, global strength assessment is not mandatory, and only local strength should be taken into account;
- for vessels with a rule length equal to or greater than 50 m and up to 70 m, consideration of both the local and global strengths is mandatory;
- for vessels with a rule length of over 70 m and up to 150 m, in addition to consideration of local strength, a global strength evaluation is to be carried out either using parametric formulae or using direct calculation methods (3D FEM models).

Local design loads (Part 5, Chap. 2) are expressed as static and dynamic pressures acting on different part of the vessels for non-displacement and displacement craft (Part 5, Chapter 3 and 4 respectively). After 'motion response' determination (relative vertical motion and acceleration), the rules provide 'Loads on the shell envelope' (hydrostatic and hydrodynamic wave pressures, pressures on weather and interior decks), 'Impact loads' (impact pressure for displacement, non-displacement and foiled or lifting device craft, forebody impact pressure for displacement and non-displacement craft), loads on 'Multihull cross-deck structure' and the 'Component design loads' (deckhouses, bulwarks and superstructures, watertight and deep tank bulkheads, pillars, deck area for cargo, stores and equipment). Design values are synthesised in tables for mono-hull, multi-hull and components as a function of local design factor and criteria representative of hull notation, service area, service type, craft type and stiffening type.

Global loads are divided into two categories: hull girder loads, and primary loads for multi-hulled vessels. Hull girder loads are to be considered for strength purposes and distinguished on the basis of their frequencies as follows:

- still water bending moments and associated shear forces arising from mass distribution and buoyancy forces, to be calculated directly as a function of load condition;
- vertical wave bending moments and associated shear forces arising from low frequency hydrodynamic forces;
- dynamic bending moments and associated shear forces arising from high frequency bottom slamming;

Wave bending moment and slamming bending moments are provided by equations as a function of rule length L_R , breadth B , service group coefficient and block coefficient. Primary loads for multi-hull craft arise mainly from the interaction between the hulls and waves.

Registro Italiano Navale (2011, 2011a) design loads are defined in Part B, Chapter 1, Section 5. First design accelerations are defined as the vertical and transverse accelerations at the centre of gravity. Then local loads are defined as hull pressures on the bottom, side and decks for planing and displacement yachts; the differentiation between the two categories depends on whether the relative speed $V/L^{0.5}$ is greater or less than 4 respectively.

For planing vessels sea pressure should be assumed as the higher of two values obtained by the following different formulations:

- hydrostatic pressure depending mainly on yacht length, full displacement and local draught and longitudinal position;
- hydrodynamic pressure defined as a function of the yacht length, maximum design vertical acceleration, longitudinal position and other coefficients taking into account varying deadrise angles along the hull and unsupported panel areas.

In the case of displacement yachts only the first, static pressure formulation is considered.

Global loads are given in Chapter 1, Section 5 as longitudinal bending moment and shear force in still water and in waves by formulas as a function of hull dimensions, block coefficient and a speed coefficient. A direct procedure to take into account the increase in bending moment and shear force, due to impact loads in the forebody area, for the sagging condition only, is available. In this case the vertical acceleration at LCG given by the rules should be considered, which corresponds to the average of the 1 % highest accelerations in the most severe sea conditions expected. For twin hull yachts transverse bending moment and shear force and transverse torsional connecting moment are also given. The minimum section modulus of the midship section is intended to comply with the maximum total bending moment and with the maximum allowable bending stress of the material.

4.3.2 Small Yachts

For small motor yachts with length less than 24 m to be commercialised in Europe, the vessel's hull should be constructed according to the ISO 12215 (2005). With respect to loadings, the relevant parts are contained in ISO-12215 Part 5 (2004) which contains detailed sections for calculation of design pressure for motor and sailing craft. All parts of the vessel are considered such as bottom, sides, decks, superstructures and deckhouses, windows hatches and doors. The bottom pressure, as an example, is calculated by a formulas as a function of the displacement, waterline length, breadth, corrections for longitudinal position x/L_{WL} , the size and aspect ratio of the shell panel and a dynamic load factor which takes into account whether the craft is displacement, semi-planing or planing (as well as whether the craft may be entirely clear of the water for short or long periods of time) given in both parametric equation and tabular form.

Harzt (1998) gives a background to the development of ISO 12215, discussing and explaining the decisions made with respect to design pressures in general and also side and deck pressures specifically. He states that the bottom pressure calculations are based on the Heller and Jasper (1961) and Savitsky and Brown (1976) approaches, and also discusses the origin of the estimates for speed, running trim angle and longitudinal impact factor and design category factor. In Appendix III, Hartz includes comparisons of bottom pressures obtained using various existing rules (VTT, BV, LR, GL, ABS) and notes that '*the load assumptions are differing considerably, which is not surprising, as the step from loads to scantlings is not identical*'.

The GL Rules for 'Special Craft - Yachts and Boats up to 24 m' (GL, 2003) also consider smaller yachts and boats ($6\text{ m} \leq L \leq 24\text{ m}$). Basic principles for load determination are given in tabular form in Section 1, A 'Hull Structures' 1.9. Hull loadings are presented for shell bottom and shell side, as well as 'correction factors for speed' for shell bottom and side and various internal structural members and frames, and then deck and superstructure loadings are specified. Rudder force and torsion moment design loadings are calculated in Section 1, A 'Hull Structures' 3.2.

Further, the RINA 'Rules for the Classification of Pleasure Yachts' (2011) can be used

for smaller vessels, since their applicability is valid for yachts down to 16 *m* in scantling length.

5 STRUCTURAL STRENGTH AND RESPONSE

Following the practice for conventional ships, there are two design philosophies for yacht structural design that can be assumed, namely the ‘first principle’ approach and the use of CS’ rules; often a mixture of both methods is practiced. Designing by CS’ rules provides reliable scantling procedures and widely accepted loads but it doesn’t allow the refinement of structural dimensions and weights. For larger, innovative and more performance sensitive vessels, a first principle approach becomes mandatory. Being based on direct calculations, first principles approach requires rigorous procedures and accurate prediction of the loads acting on the hull structure but it allows the determination of any kind of structural response for subsequent processing. The response of hull structures to different types of loadings results in static stresses and deformations, dynamic stresses in way of vibration, noise and slamming impacts, thin plate buckling and fatigue phenomena.

5.1 Structure Design Methods

The design criteria of motor yacht hull structures are mainly related to their dimensions and speed. For smaller, high speed yachts the structure scantling is mainly performed on a local basis by applying dynamic pressures stemming from planing effects, such as bottom and side slamming. For larger displacement, or semi displacement vessels, the evaluation of hull girder global strength must be performed as well, with respect to both still water and wave pressure distribution.

In most cases the first, rough scantlings are performed by the application of CS’ rules. In the next iteration, a significant reduction in structure dimensions can be pursued by recourse to direct methods based on beam and plate theories. For smaller vessels direct analysis is addressed to local areas such as decks, sides or bulkheads modelled by two-dimensional grillages or orthotropic stiffened plates (e.g. Maneepan *et al.*, 2006 and Sobey *et al.*, 2009); transverse sections can be analysed by two-dimensional frames. A check of longitudinal strength can be carried out as well by simply verifying the main section inertia. This is important for FRP yachts, even if below 50 *m* in length, because of the low elastic modulus of the material (Loscombe, 2001).

For larger units, where global loads became predominant, the longitudinal strength is carefully evaluated by simplified two-dimensional hull girder schemes with constant or variable sections. By determining the balance between sectional weight versus sectional buoyancy it is then possible to achieve still water shear force and bending moment distributions. Additional contributions to shear and bending moments from waves can be accounted for by CS’ rules or by quasi-static equivalent wave analysis. Moreover the torsion moment can be addressed from class rules. By this approach it is possible to achieve additional information relevant to structure deformations, increasingly important for verifying window glass integrity. Generally the first scantling iteration for a superyacht considers deformations rather than stresses; a realistic limit for maximum vertical deformations amidships is 1/1000 of the scantling length.

At present the structure of a medium size motor yacht produces a very complex lay out owing to the necessity of reducing the reinforcement dimensions to internal volumes’ advantage, to the presence of large transom and side doors and terraces and to the increasing structure irregularity to match interior arrangements. Large openings, in particular, induce high stress concentrations and, in this regard, the analysis by

numerical models has become mandatory. FEA represents the most detailed level of approach for structural design and it allows to model the structure with any detail, to keep into account asymmetrical structure such as partial decks, longitudinal bulkheads and side doors, and to analyse the structure in its three dimensional form under the contemporary action of different loadings. A review of numerical techniques now available to industry for superyacht design is presented by Köhlmoos and Bertram (2009). FEM methods for static and dynamic analysis are the base for vibration, noise, fatigue strength and ultimate strength assessment. An example of FEM analysis on a large steel motor yacht to control structural deformations and their compatibility with surface fillers is presented by Fincantieri (2010). Using a global analysis the dynamic behaviour of the yacht excited by short wave loads has been determined to quantify the dynamic vertical bending moment and springing phenomena. Then the effects of local slamming have been studied on a hull portion modelled by a very refined FEM model to determine the long term maximum displacement of side and bottom panels.

A similar approach is described by Motta *et al.* (2012) to investigate the stress distribution on a 60 m steel yacht with large side doors and other asymmetrical structural components. Particular care has been dedicated in creating the numerical model, shown in Figure 5, in order to obtain a very refined mesh capable of analysis in the time domain. The numerical analysis is still underway and the results will be compared with tests already carried out in real scale on the same yacht.

The needs of a FEM analysis is particularly felt for multihull vessels for which simplified models based on longitudinal symmetry cannot be used. An example of a FEM application to a catamaran motor yacht is presented by Luco *et al.* (2002). The study is further complicated by FRP hull material: the material properties have been verified by laboratory tests and then modelled by proper multilayer elements. The authors considered three static loading conditions typical of multihulls: hydrostatic pressure, prying moment and torsion, all provided by DNV Rules for high speed crafts.

The present trend in structural design is to perform combined FEA/CFD investigations where pressure distributions resulting from seakeeping analyses are directly applied to a FE numerical model. Such a procedure is compared by Hermundstad and Wu (1999) with a traditional global load method and with a modal method, all applied to a monohull and a catamaran fast vessel.

Superyachts have very large superstructures in order to allow for more interior space; when the superyachts dimension exceed 100 m in length the interaction of superstructures with hull structures should be considered with care. Albertoni *et al.* (2000) made an investigation on this subject modelling a 70 m naval vessel and analysing the contribution of superstructure in terms of stress and deformations. From the analysis,

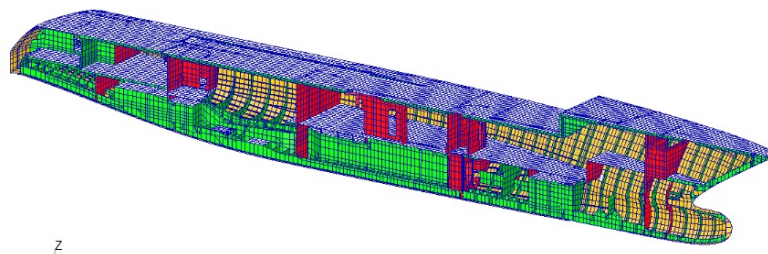


Figure 5: FEM numerical model of a 60 metres superyacht.

for long superstructures, 1 or 2 expansion joints become mandatory in order to keep deck stresses within acceptable values.

5.2 Vibrations and Noise

Vibrations and noise are crucial topics for superyachts and they require detailed calculations from the earliest of design stages to verify the dynamic behaviour of hull structures and their response to exciting loads such as propellers, engines and wave encounters. Even if vibrations and noise are more critical for metallic yachts, FRP units are not immune from these phenomena. The problem is increased by higher comfort requirements and constraints imposed by the ISO 6954 (2000) with respect to the previous ISO 6954 (1984) standard, together with CS notations which ask for even lower levels of vibration and noise. All main CS recently introduced comfort requirements addressing highest admissible vibration and noise levels. Baker and McSweeney (2009), as an example, present a complete analysis of present ABS Rules concerning vibrations and noise published in the ‘Guide for the Class Notation Comfort - Yacht’ (2008). Two notational options are considered: COMF(Y), which establishes a level of comfort based on ambient noise and vibration alone and COMF+(Y) which adds slightly more demanding criteria for noise and vibration, and provides additional criteria for the assessment of motion sickness. ABS Yacht Comfort guide, however, have been recently revised in some aspects. In Table 2, a synthesis of the new version is reported for yacht below and over 45 m in length. Comfort regulations for yachts are also contained in other CS rules such as:

- Bureau Veritas (2011) Part E, Section 5, ‘Additional Requirements for Yachts’;
- Det Norske Veritas (2011), Part 6, Chapter 12, ‘Noise and Vibration’;
- Germanischer Lloyd (2003b), Part 1, Chapter 16, ‘Harmony Class’;
- Lloyd’s Register (2011), Chapter 6, ‘Passenger and Crew Accommodation Comfort’;
- RINA (2011a), Part E, Chapter 5, ‘Comfort on board’.

Some examples of maximum vibration levels are reported in Table 3 for BV, LR and RINA.

From the structural point of view vibrations take place both at global or local level, being the first ones more incisive and difficult to put right after the yacht is built. Even

Table 2: Maximum whole-body vibration according to ABS (COMF(Y)) for yachts below and over 45 m in length.

Yacht length	Notation	Frequency Range	Acceleration Measurement	Maximum Level	
				Underway	Anchor
$L \leq 45\text{ m}$	COMF (Y)	1 – 80 Hz	a_w (v)	89.4 mm/s^2 (2.5 mm/s)	53.5 mm/s^2 (1.5 mm/s)
	COMF +(Y)	1 – 80 Hz	a_w (v)	53.5 mm/s^2 (1.5 mm/s)	45.0 mm/s^2 (1.25 mm/s)
$L > 45\text{ m}$	COMF (Y)	1 – 80 Hz	a_w (v)	71.5 mm/s^2 (2.0 mm/s)	45.0 mm/s^2 (1.25 mm/s)
	COMF +(Y)	1 – 80 Hz	a_w (v)	53.5 mm/s^2 (1.5 mm/s)	35.75 mm/s^2 (1.0 mm/s)

a_w = multi axis acceleration value calculated from the root-sums-of-squares of the weighted root mean square (RMS) acceleration values in each axis (a_{xw} , a_{yw} , a_{zw}) at the measurement point. v = spectral peak of structural velocity in mm/s.

Table 3: Maximum whole-body vibration according to Bureau Veritas, Lloyd's Register and RINA Comfort Rules for yachts.

Location	Bureau Veritas		Lloyd's Register		RINA	
	Frequency	v [mm/s]	Frequency	v_{rms} [mm/s]	Frequency	v [mm/s]
Cabins and lounges	1 – 80 Hz	1.0 – 3.0	1 – 80 Hz	1.8 – 2.5	0 – 100 Hz	1.0 – 3.0
Public spaces	1 – 80 Hz	1.0 – 3.0	1 – 80 Hz	2.5 – 3.3	0 – 100 Hz	1.0 – 3.0
Open recreation decks	1 – 80 Hz	2.0 – 4.5	1 – 80 Hz	2.5 – 3.8	0 – 100 Hz	2.0 – 4.0

v_{rms} = overall frequency weighted r.m.s. value of vibration during a period of steady-state operation over the frequency range 1 to 80 Hz. v = spectral peak of structural velocity.

if simplified models based on variable section girders with concentrated masses remain a valuable tool to calculate approximate values of the first natural frequencies of the hull, only by FEM analyses of the whole structure is it possible to achieve reliable results and to avoid any structure resonance with the exciting frequencies. Given that the propeller blade passing frequency is relatively low (below 10 – 15 Hz) the danger exists more probably for large units over 80 m. The presence of large openings, in addition, further complicates the dynamic behaviour of the hull lowering its natural frequencies and inducing additional torsion modes.

Where local vibrations are concerned, decks and superstructures are the most critical areas; most inconveniences come from high frequency excitations, primarily caused by main and auxiliary engines, and by structural discontinuities and irregularities. Also, in this case a detailed FEM analysis is the only way to individuate and correct problems. As a general rule the only way to avoid vibrations is to keep natural frequency very high and this can be achieved only by increasing hull stiffness. In this regard the longitudinal framing system shows higher natural frequencies with respect to the transverse one; this may be ameliorated by reducing the transverse frame distance and the longitudinal stiffener spacing. As a matter of fact any action towards vibration reduction implies an increase in structural weight: as an example, it has been estimated that for a 95 m megayacht the weight increase to avoid maddening vibrations amounts to more than 100 tonnes.

Köhlmoos and Bertram (2009a) present a specific analysis of the vibrations induced by the propulsive system of a superyacht, performed by the combined use of experimental techniques, FEM and CFD tools. First the hull natural frequencies have been measured by an experimental investigation. In a second phase the excitation sources have been identified by a CFD analysis of the water flow around appendages and applied to a FEM model of the ship to individuate critical areas. By a series of modification of underwater after body performed by CFD simulations and correspondent FEM control of vibration levels of critical areas, the problem has been iteratively solved.

The noise abatement for motor yachts is another strategic issue related to onboard comfort and most difficult to achieve because of powerful and high speed propulsion engines, related gear boxes and highly loaded propellers with reduced clearances. The acoustical implications of motor yachts should be taken into account from the earliest of design phases because any subsequent interventions on an already built unit in most cases doesn't give any improvement. A synthesis of a correct approach to noise assessment on small vessels is presented by Juras (2000); he first analyses the noise sources on board and then the possible actions to reduce their intensity. For propeller (or water-jets), noise solutions are a higher number of blades, skewed blades and appropriate propeller-hull clearance; for engines and gearboxes usual acoustical enclosures

Table 4: Maximum noise levels for superyachts. Values in $dB(A)$ are provided for 'in harbour' and 'sailing' conditions.

Spaces	Lalangas (1983)	ABS (COMF(Y))	BV	GL (cruise ship)	LR	RINA	90 m yacht (2011)
	Harb/Sail	Harb/Sail	Harb/Sail	Harb/Sail	Harb/Sail	Harb/Sail	Harb/Sail
Owner cabin	35/73	40/45	40/50	44/52	50/50	45	40/44
Guest cabins	35/73	45/50	40/50	46/54	53/53	45	43/47
Lounges	40/77	50/50	45/55	52/60	55/55	55	45/50
External decks	50/89	60/65	55/75	64/72	63/63	55	65/70

are the most used tools. Then the noise propagation paths (air-borne, structure-borne and hydrodynamic noise) are analysed together with relevant measures of noise abatement to be adopted in accommodation and working spaces. The author asserts that there are not big differences between steel, aluminium and FRP yachts in the noise dominant frequency range (up to 125 Hz) while better behaviour is shown by wooden vessels. Finally some considerations on the existing noise levels criteria are carried out, underlining that they have been established for large ships and that, for smaller vessels, the noise level on board is generally higher.

On this subject it is interesting to assess the developments of noise levels in time. Lalangas and Yannoulis (1983) report these values for a planing aluminium motor yacht in two different operating conditions: underway at full power and when at anchor with operating generators (sailing/anchor). In Table 4 maximum noise levels are compared among Lalangas (1983), ABS, BV, GL, LR and RINA comfort Rules. Finally the values resulting from real scale measurements on a 90 m superyacht built in 2011 are reported as well. To be noted is a much smaller difference between under-way and at-anchor conditions.

Nevertheless the theoretical noise prediction at the design stage still is not fully reliable. A numerical procedure based on FEM approach has been applied to a container ship by Cabos and Jokat (1998). This procedure simulates the propagation of structure borne noise in complex ship structures, taking advantage of existing finite element models created mainly for strength and vibration computations. An example of an integrated approach to this problem is presented by Colombo *et al.* (1995) for a 30 m fibreglass motor yacht. In this paper the prediction and experimental verification of noise and vibration level is described.

5.3 Buckling, Fatigue and Reliability

Buckling phenomena on superyacht structures are not so frequent but particular attention must be paid to structures made from FRP and aluminium because of their low elastic modulus. Loscombe (2001) proposes a procedure to calculate when it becomes necessary to take into consideration the buckling phenomena of panels on FRP motor yachts. Buckling stress values are provided by a simple formula as a function of glass fibre weight fraction, glass reinforcement weight and shortest span of the panel. Benson *et al.* (2009) present a detailed FEM analysis of the ultimate strength of aluminium

stiffened panels built from marine grade 5083-H116 and 6082-T6 under compressive load. The paper describes a series of nonlinear large deflection FEM analyses carried out on aluminium panels typical of high speed vessel deck or bottom structures, investigating their uniaxial in plane compressive strength assuming interframe and overall collapse modes. The results have been compared to equivalent steel panel analyses.

Given the relatively low yearly usage factor of a motor yacht, fatigue life evaluation is not a limiting criterion in structural design. Nevertheless, a scrupulous designer must not ignore this aspect. The usual procedures based on cumulative damage and crack propagation theories adopted for ships are applicable to yachts as well. A complete procedure to analyse the fatigue life of a 68 m aluminium catamaran is presented by Di *et al.* (1997). A complete FEM model of the vessel has been loaded by fundamental wave loading cases including longitudinal and transverse bending, torsion and splitting moments. Some cracks have been included in the numerical model in order to study the consequence of fatigue damage on the structure. Furthermore, a fatigue life assessment has been carried out by the application of S-N curves and fracture mechanics.

Reliability methods can be applied to superyacht structures as for conventional ships; an example of such an approach to the structural design of a 34 m, steel patrol boat is described by Purcell *et al.* (1988). The structure scantling has been carried out by traditional method and FEM calculations. Full-scale testing have been performed as well to establish a relationship between hull stress and acceleration measurements. On the base of the gathered data the probability of bottom plate yielding has been calculated by Monte Carlo simulation. The described calculation is based on an operative life of 15 years and 2000 hours of operation per year. Considering 8 hours per day, this corresponds to 250 days of navigation per year, totally out of the common run for superyachts.

5.4 Yacht Motions

Even if not strictly a structural item, the response to waves is crucial for the onboard comfort and for seasickness arising. The latter heavily influences the good or bad mood of owner and guests and a preliminary analysis of the vessel characteristics on this subject is advisable. This important issue is discussed by Dallinga and Van Wieringen (1996) in terms of comfort criteria, hydrodynamic characteristics, 'mission' related criteria (e.g. operability) and prevailing wave climate. Design indications to obtain a comfortable vessel and methods of zero speed stabilisation are given as well. Van Wieringen *et al.* (2000) extend this work using both motion simulator tests and long-term ratings for both passengers and crew. Theories of motion sickness, general operability criteria and design considerations are also presented by Stevens and Parsons (2002) for fast vessels.

6 MATERIAL SELECTION

Given that the driving philosophy in designing and building motor yachts is the cost reduction, the choice of the construction material also depends on their specific mission and dimensions. Materials are chosen for their appropriateness in the same way they are for vessels with other missions. As with commercial and government vessels, motor yacht material selection is predominantly based on cost (both initial and life-cycle) and weight. Insulation properties, predominantly noise and thermal, and vibration damping, are often emphasized. Unlike those other vessel types, some yacht materials are chosen for their aesthetic qualities.

On account of the demand protraction wooden yachts below 24 m in length continue to be built by a restricted number of long experience shipyards. In the higher range between 24 and 45 m, even if fibre reinforced plastic (FRP) is the most diffused material, aluminium alloy has a wide application, especially for high performance, one off realisations. The upper bound of this category represents the FRP dimensional limit owing to its low mechanical properties and elastic modulus; at the same time steel begins to become the standard. For vessels over 45 m global loads assume important values and steel becomes the only possible choice. Aluminium alloys continue to be an interesting alternative to steel material for high performance vessels while it is the standard for superstructure construction. A very detailed analysis of the advisable materials for high speed vessels is presented in the paper by Jackson *et al.* (1999) which can be considered a real point of reference on this subject. All the mechanical properties, including the specific strength and rigidity of various types of steel, aluminium and FRP are tabulated and compared with each other. The titanium Ti-6Al-4V alloy is considered as well.

A comparison of steel, aluminium and FRP as possible alternatives in the construction of a large motor yacht has also been carried out by Marchetti (1996) with regards to mechanical properties, fatigue life, impact strength, corrosion, vibrations and noise propagation, reparability and hull weight. A similar work has been published by Boote (2004) in which the structural scantling of a 55 m yacht has been performed for steel, aluminium and FRP construction. The three solutions have been compared in terms of shell, longitudinals and frame weights; the final comparison, made for the yacht at half load displacement showed that the FRP version had a displacement 9% lower than the steel one, while for the light alloy version the difference rose to 17%. The main advantages and disadvantages of each construction material are synthesised in the previous ISSC 2009 Report of V.8 Committee about sailing yachts and they remain the same for motor yachts. In this chapter current trends in material selection and the associated production methods specifically in the motor yacht industry are described.

6.1 Wood

In the last two decades a return of the oldest material for yacht building has been observed. Even if the traditional hull construction based on solid wood has become more and more difficult due to the low availability of exotic woods such as mahogany, teak, okoumé and iroko, new construction techniques based on plywood and laminated wood, coupled with new bonding products derived from the composite industry, allow the best advantages of wood's mechanical properties and light weight to be exploited. In addition modern techniques more efficiently protect and seal the wood from moisture. Mahogany continues to be the most wanted for solid parts and plywood, while red cedar is the most suitable for laminated strips. Other less exotic woods like oak, ash, elm and spearwood are used for structural components, depending on local availability. Moreover wood continues to be the basis material for refitting and repair and the most diffused material for interior and furniture on modern yachts and active research in processing techniques is continuously carried out by designers to achieve new visual effects in wood for furnishing.

6.2 Metallic Materials

The steel types used for yacht building are the same as those used for ships and are well described by the CS' rules. For displacement vessels of low/medium size dimensions, with transversely framed systems, mild steels with yield strength below 235 MPa have

been widely utilised since the sixties. Then the necessity to reduce structural weight drove the use of high tensile steels with yield strengths up to 390 MPa and, at present, almost all motor yachts are built with these alloys. For vessels with high performance requirements and medium/large dimensions, aluminium is the best choice: AlMg 5083 is the typical aluminium/magnesium light alloy used for hull construction, particularly resistant to salt environment and very suitable for welding. If properly protected by sacrificial zinc anodes the problem of its vulnerability to galvanic corrosion are easily overcome. Recently, for structural parts not in contact with water, the 6000 series of alloys are successfully used because of the lower cost. As aluminium's mechanical properties are heavily influenced by welding procedures, their values are commonly provided in unwelded or welded conditions. A complete review of aluminium light alloys for marine constructions, with main characteristics and research trends is presented by Sielski (2007). Both steel and aluminium hulls suffer shell deformations caused by welding processes and reworking and/or fairing correction by filler is always required.

The use of titanium has increased recently due to the reduction in its cost and a certain interest has been devoted to this material also in the field of yacht construction. While titanium has been used in a variety of marine applications since the fifties, its cost was prohibitive for most uses (Williams, 1970). At the present time it is roughly twice as expensive as stainless steel for equivalent strength while having roughly 57 % of stainless steel's density. Grade 4 and 5 titanium are mostly used in large components under large loads, such as hydraulic cylinder rods, padeyes and cleats. Other components include exhaust components, stanchions, seawater piping, valves and ventilation components (Lazarus, 2011). Some failures have pointed out however that while the stainless parts they replaced would show small amounts of observable damage prior to failure, the titanium parts developed fatigue cracks that were easily missed, necessitating a periodic inspection process. A promising area is for titanium rudder shaft bearings due to their low corrosion.

6.3 Fibre Reinforced Plastics

After several experiments of partial fibreglass boats (aluminium frames with FRP shell) were started in the forties, the first FRP motor boat was a 41 ft sportfishing boat, built in the USA in 1959 utilising a combination of polyester resin and E-glass fibres manufactured with a hand lay-up procedure in a female mould. Since then great progress has been reached in composite technology applied to vessel construction. While E-glass fibres remain the basic reinforcement owing to their acceptable mechanical properties and low cost, for more specific applications, where higher strength and stiffness are required, together with lower weight, aramid and carbon fibres are more suitable.

As the distinct advantage of FRP composites is the ability to tailor the property directionally to suit specific applications, a very large number of fabrics have been made available on the market for glass, aramid and carbon fibres. The most used in the boatbuilding field are listed by Boote *et al.* (2006) together with their most relevant construction technologies; many issues relative to regulations and new manufacturing are contained as well.

Thus, unbalanced woven roving, with a higher fibre percentage in the warp direction, are used to increase the hull stiffness in the longitudinal direction; the use of rovimat (mat and roving stitched together) allows the lamination process to be significantly sped up. Biaxial fabrics are used to increase a hull side's resistance to shear forces

and torsional moments. Unidirectional reinforcements are used on beam flanges to increase stiffener modulus keeping weight low. Where resins are concerned, isophthalic and orthophthalic polyester resins are progressively replaced by vinyl ester resins to increase the composite resistance to the marine environment, with particular reference to osmotic blistering. For particular applications with aramid and carbon fibres and when greater fatigue or impact resistance is desired, the most expensive and efficient epoxy resin is generally used. Arvidson and Miller (2001) showed the higher shear strength of the epoxies and vinyl esters allow the elimination of 'tie' layers of random-oriented mat, significantly reducing weight.

Sandwich plating, commonly used for decks and then even more often for hull sides as well, are generally built with glass fibre skins and balsa or PVC cores; various densities are available to match different resistance requirements. Where cores are concerned the best solution in terms of weight and stiffness is represented by both Nomex or aluminium honeycomb. In this case particular care is required when bonding skins and core to each other. Sandwich construction is used in place of single skin construction to reduce weight and improve vibration damping and provide greater thermal insulation. When the weight reduction becomes mandatory more sophisticated materials are used for skins such as carbon and aramid. Core selection becomes an important consideration with trade-offs for each of the popular types. Cores are often selected by their shear strengths and the strongest for its density is end-grain balsa wood. Balsa's main drawback is its tendency to rot if exposed to moisture for a long period of time, requiring careful fabrication in the boat and adherence to high quality standards during repair or modification. PVC cores are growing in use with the cross-linked varieties more appropriate for deflection limited designs and the linear PVC cores more suited for impact resistant designs. Polyurethane cores are used when insulation is a primary concern, while honeycomb cores of aramid, aluminium or polyethylene are used when reduced weight is the main goal. Honeycomb cores are often combined with thicker, 'cosmetic' face sheets for joinery.

At present environmental sustainability is becoming more and more inherent to the FRP marine constructions owing to the large quantity of material needed to build a yacht. An intense research is addressed to new materials that can be easily recycled and that can be derived from sources that are unlikely to be depleted or finite. Particular attention has been devoted to natural fibres such as flax or hemp encased within a polylactic acid resin matrix. In his paper Gravid (2011) reports a comparison between natural and glass laminates characteristics, with particular attention to mechanical properties. Malmstein *et al.* (2011) have been investigating the use of sustainable structural composites for FRP construction, in particular looking at the durability of castor oil and linseed oil based resin systems combined with glass fabric to long term exposure to water. These systems show promise in mechanical properties in comparison to epoxy/glass composites (in the case of castor oil derived resin and glass) and polyester/glass (in the case of linseed oil derived resin and glass).

7 STRUCTURAL ARRANGEMENTS

Structural arrangements of yachts show diversities in accordance with hull length, hull forms, speed range and construction materials employed. In the following the most important structural characteristics and developments are outlined for each construction material.

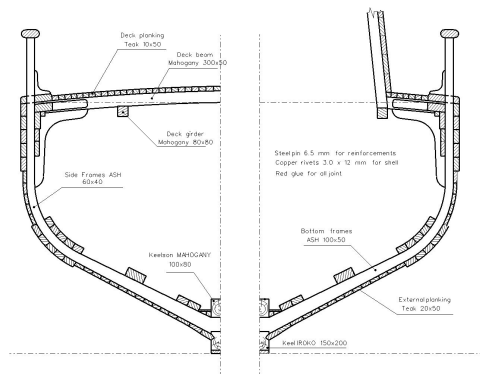


Figure 6: Typical main section of a wooden displacing boat with timber structural elements.

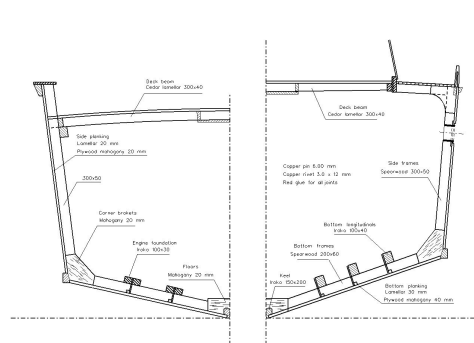


Figure 7: Typical main section of a wooden planing boat with plywood and laminated components.

7.1 Wood

Wooden boat construction is often defined as an ‘art’ rather than a simple profession. The solutions to work, to bend, to glue and to join solid wood have approached perfection through the centuries. The displacing, slow vessels with round hulls were built with the traditional solid technique with closely spaced frames and floors and few longitudinals, with keelsons to support engines; the low speed (often below 10 *knots*) and the reduced dimensions did not require additional longitudinal stringers (see Figure 6). The need to reduce the weight and to increase the speed though has directed builders to new solutions based on the use of plywood. With flat or single curvature bottoms and sides it became easier and cheaper to build straight frames connected by plywood floors and brackets by means of ‘red glue’ and copper rivets.

This solution continued to be largely used up to the Sixties to build fast patrol boats and relatively large motor yachts with just some innovations represented mainly by the introduction of glued lamellar wood. Lamellar construction allows the building of long and thick keels in a unique piece to include the required curvature. With multilayer planking, there are reduced transverse frames and, in general, a reduction in joints and structural mass, improving the craft’s performance through reduced resistance. In Figure 7 an example of a main section of a 20 *m* wooden yacht is presented. From 1900 to 1970 many motor boats have been built with this technique in the United States and in Europe. A review of the present criteria and methodologies for wooden yacht construction is presented by Vesco (2005). In his paper many drawings relative to a 21 *m* planing yacht built in wood are contained together with a synthesis of rules relevant to wood scantlings and many interesting photos of the construction sequence.

Nowadays a number of shipyards continue to use wood for motor yacht construction, pushed by an increasing demand of enthusiasts of this material. The average dimensions of modern wooden motor yachts are around 20 – 25 *m* in length, but vessels up to 30 *m* are not so rare. Even recently in Dubai a wooden motor yacht has been built that measures 47.5 *m* in length and a 140 *m* wooden sailing yacht is under construction in Turkey.

7.2 Steel and Aluminium

As previously mentioned the use of steel in motor yachts coincides with the introduction of steam engines to power ships. Steel vessels at first had a typical transverse,

bolted structure with close frames and longitudinal primary reinforcements with the same, well tested lay-out coming from wooden constructions. To save weight and to overcome the difficulties in assembling plates some units had wooden shells and deck planking.

This lay-out soon showed its limits with regard to speed performances because of its high weight. The definitive change in steel vessel structure came at the end of the World War II with the invention of welding, thanks to which it was possible to reduce significantly weights and costs, to increase strength and stiffness and, as a consequence, the length of ships. In addition welding made it possible to realise new and more fashionable hull and superstructure lines, this last aspect being particularly attractive for yacht designers. In its beginnings, welded large steel motor yachts were built with normal steel with a traditional transversely framed structure composed of secondary frames 500–800 mm spaced and web frames every three or four intervals. Longitudinal reinforcements were limited to one central and two or more lateral keelsons on the bottom, with reinforcements on the side and deck girders (see Figure 8). To reduce weight and improve stability superstructures were built in aluminium light alloy with riveted joints as the welding technique for this material became only reliable later in the Sixties. The hull-superstructure connection was made by screw bolts in such a way to insulate steel from aluminium and avoid dangerous galvanic action.

Nowadays a bimetallic joint is widely used consisting in an aluminium /steel strip explosively clad together. The steel side of the strip is welded to the main deck and the superstructure is welded to the aluminium side of the strip. A detailed description of the bimetallic strip concept and construction is presented in the paper by Young and Banker (2004) together with its most important marine applications.

With the increase of conventional ship dimensions hull structural lay-out moved from a transverse scheme to a longitudinal one, in order to increase the longitudinal strength and stiffness. The same trend was assumed for motor yachts where the longitudinal structure was particularly appreciated for its reduced weight.

Longitudinal framing system on superyachts is characterised by widely spaced, deep transverse frames, typically between 1000 mm, for aluminium vessels, up to 2500 mm for steel ones, depending on dimensions and speed; lower values are often assumed in the bow and stern areas to better withstand slamming and collision loads. Frames support longitudinal stringers, generally bulb or angle profiles, closely spaced (between 300 and 600 mm) to minimize shell thickness (see Figure 9).

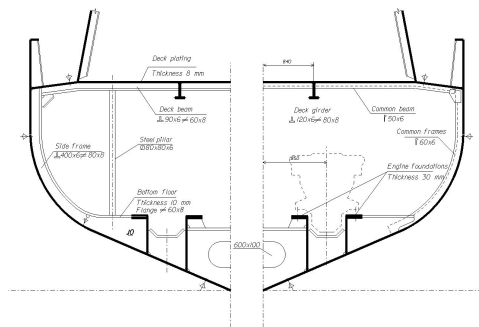


Figure 8: Main section of a displacing motor yacht in welded steel with transverse framing lay out.

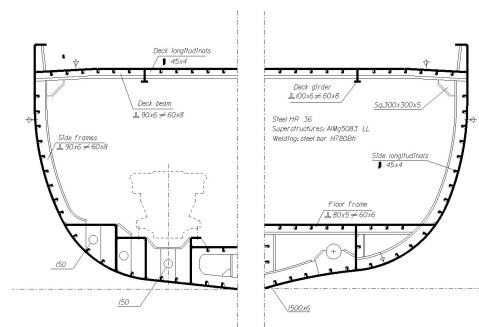


Figure 9: Main section of a modern motor yacht with longitudinal framing structure.

Longitudinal framing gives a higher section modulus without any weight increase with respect to transverse framing but, from the construction point of view, it requires higher construction times and costs because of the larger number of connecting members. For this reason, generally small yachts are built with a transverse structure while, for longer ones when hull girder loads increase significantly, the longitudinal structure is always assumed; the transition length stands between 50 and 80 m.

A third solution is represented by hybrid structure in which bottom and decks are longitudinally framed and sides are transversely framed. This lay out represents the best in terms of resistance to longitudinal bending and to side loads and it is particularly suited for yachts sailing in icy water. On the other hand a hybrid structure is the most expensive one and shipyards are reluctant to adopt it.

So far when a new design is starting, the choice of the framing system often requires a deep investigation taking into consideration strength, costs and other factors like noise and vibration. Schleicher (2003) in his paper about the 100 *knots* super yacht, together with the main properties of suitable construction materials (high strength steel, aluminium and FRP), presents a comparative analysis of hull weights relative to framing systems. For the three considered materials the weight per metre is plotted versus stiffener spacing (from 1000 to 2000 mm) for both transverse and longitudinal framing systems. Another systematic comparison between longitudinal and transverse framing system has been carried out by Roy *et al.* (2008) on an 85 m steel yacht for which the two lay-outs have been fully developed, using the Lloyd's Register SSC Rules, for a section of hull 20 m in length. The considered length of 84 m is very close to the limit length of a 3000 GRT vessel for which MCA LY2 is still applicable. The study presents results in terms of weight, number of structural parts to be assembled and welding length. Other factors are considered as well, in particular the influence of framing system on noise and vibration.

While small and medium size yachts are fitted with only one main deck, on larger vessels (over 60 m) an intermediate deck is inserted between the main one and the double bottom. The two deck arrangement allows additional space below the cabin deck generally devoted to crew personnel, and a technical tunnel where most piping and cables can be fitted and easily inspected (Figure 10).

Present trend asks for the introduction of large openings in the hull transom and sides in order to give to cabins direct access to the external (balconies) or to allow tenders to be easily lifted and recovered into garages or to enter directly into inner harbours; on some yachts, other openings can be found at fore on the main deck for tender recovering. The presence of these large openings, often not symmetric, has a negative effect on the hull watertightness first of all, and then on the hull beam

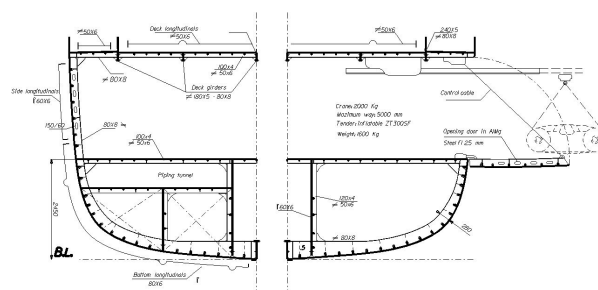


Figure 10: Main section of a 60 m steel yacht with intermediate deck and side door.

strength and dynamic behaviour; by the way they interrupt the structure integrity and continuity. For these reasons opening doors should have the same resistance of the integral hull structure and very strong closure mechanism and hinges which allow for a perfect closure and watertight are necessary. In the same way proper stiffening frames should be implemented in the hull around the opening to avoid local deformations (with consequent water entrance) and global bending and torsion effects. Door hinges should be dimensioned properly to resist to the high accelerations induced by the yacht motions in waves. To reduce inertia closing doors are built in aluminium light alloy.

As already mentioned in Chapter 5, the only way to evaluate the consequences of large openings on the structural behaviour of the hull, both statically and dynamically, is by a preliminary FEM analysis of the complete vessel. Even if the cost of hull structures for a steel yacht is about 10 % of the total price (compared with more than 50 % for a bulk carrier), an accurate scantling can have a significant positive effect on cost reduction. This is the aim of the optimization procedure presented by Motta *et al.* (2011) based on the use of the LBR 5 software and applied to a 60 m superyacht. The conclusions show a reduction of the structure weight of up to 8 % with respect to the initial design carried out according to CS' rules and direct methods.

The limit of welding, for which it was not possible to join plates thinner than 3 mm, made steel suitable for yacht having lengths in excess of 40 m. Below this limit steel can be replaced either with FRP or with aluminium light alloy. Both offer excellent results with regards to lighter displacement and aluminium is also particularly suitable for the construction of one off units or for small series production. Aluminium light alloy was originally difficult to construct due to accessible and cost effective welding technology and therefore only saw application in aerospace and in military patrol boats. By the beginning of the Sixties, the yacht industry was able to take cost advantages from the progress in TIG (Tungsten Inert Gas) and MIG (Metal Inert Gas) welding techniques. The structural lay-out of aluminium boats is not so different from steel vessels and only some restrictions should be respected, mainly due to the different mechanical characteristics and welding behaviour of this material. Where the framing system is concerned Kaneko and Baba (1982) suggest avoiding transverse structures for values of speed-length ratio, $V/L^{0.5}$, greater than 4.

Modern aluminium yachts are generally longitudinally framed with shorter spacing with respect to steel (not more than 1000 mm) and symmetric section stiffeners, such as T or flat bars, are recommended to reduce the risk of lateral buckling. Very interesting design aspects are presented by Henrickson and Spencer (1982) for an aluminium crewboat, including the bottom structural analysis based on a 'limited' reliability approach and the evaluation of the fatigue life. Another very rich information source is represented by Lalanga and Yannoulis (1983) in which the design and construction of a 25 m aluminium motor yacht is presented. They provide simple formulae to calculate bottom plating thickness and longitudinal and transverse reinforcement moduli as a function of design pressure. With this procedure a saving of about 40 % of the hull structure weight is declared by the authors. The concept of weight saving is particularly stressed in the paper by Rusnak (1999) about the design and construction of a 40 m sportfisherman built in aluminium, with a speed of more than 30 knots. The author writes that *in general the design philosophy for structure was to optimize the overall structure to save weight, with particular emphasis on reducing plating thickness throughout*. This was achieved by many solutions such as *lightening holes and scallops added wherever possible to reduce weight*.

7.3 Fibre Reinforced Plastics

Since its first applications, dated at the end of the World War II, FRP spread throughout the yacht industry and, in a very short time, it became the most diffused material for small and medium size pleasure and work boats. The first structural lay-out consisted in a thick, single skin shell stiffened by 'box reinforcements' having a longitudinal framing system with web frame interval between 1000 and 2000 mm; in Figure 11 the main section of a typical semi-planing yacht with single skin hull is shown. Reinforcements have top-hat sections (also called 'box', or 'omega') with empty or PVC cores. This latter solution is now preferred because of the advantage of a simpler construction (the PVC core works as a male mould on site) and because the empty 'top-hat' beams absorbed and trapped water inside. Secondary stiffener sections in FRP constructions are not smaller in scantlings as usually observed for metallic structures where the ratio of web height between secondary versus primary reinforcements must be below 0.5. In fact, while structural connections or crossing beams represent weak points in steel and aluminium structures because of welding, in the case of FRP joints and crossings the mode of construction requires glass overlapping and extra material and they subsequently become stiffer zone. This helps compensate for FRP's low Young's modulus and achieves higher hull stiffness, avoiding structure deformations when sailing at high speed or in rough seas. In addition the number of stiffeners is reduced, therefore reducing production cost.

Despite their ease of fabrication, top-hat-type stiffeners do not have standard cross section parameters. Tsouvalis and Spanopoulos (2003) provide design curves for tophat-type cross sections meeting specific scantling requirements and Maneepan *et al.* (2006) looked at tophat stiffener lay-up optimisation. The geometric parameters considered are the crown thickness and width, the web thickness and height, the flange width, the web angle and the flange angle. The mechanism of shear stress transfer between web and flange in FRP beams is not the same as for steel so the determination of the effective breadth cannot use the same rules assumed for steel structures; on this matter Boote (2007) made a parametric investigation using FEM models to individuate linear regressions to be used in FRP structure scantling.

The low elastic modulus of this material precluded the building of very long vessels and the effort of designers and engineers was always devoted to increasing stiffness, more than the resistance, of FRP. This task has been partially achieved by 'sandwich' construction, which made it possible to obtain more rigid hulls eliminating, at the same time, secondary stiffeners thus achieving a simpler and lighter structure. In addition

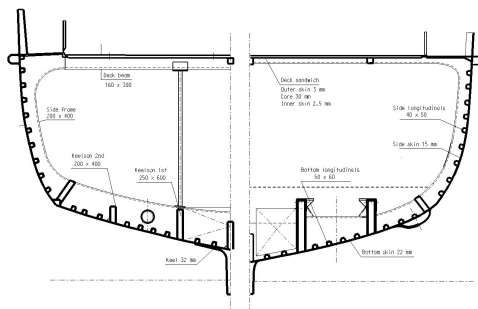


Figure 11: Main section of a 40 m FRP displacement yacht with single skin hull.

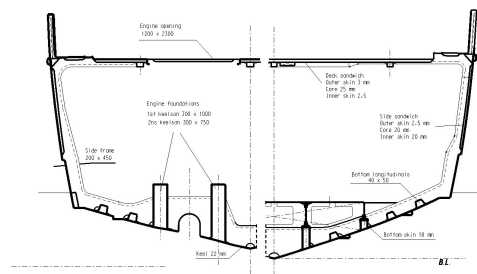


Figure 12: Main section of a 25 m FRP planing yacht with sides and deck in sandwich.

the use of more sophisticated fibres, like carbon and Kevlar, together with new lamination techniques (resin infusion and resin pre-impregnated fibre systems), contributed to obtain stiffer sandwich panels and to increase yacht lengths up to 45 *m*. Nevertheless carbon and epoxy laminates need more sophisticated production technologies based on the availability of large ovens to cure mouldings at high temperatures, not a cost effective production technique.

Classification societies had always been very careful in accepting sandwich for the whole hull shell because of the low resistance of the external thin skins to impacts. On the other hand shipyards are in favour of sandwich panels because it avoids secondary stiffeners, further simplifying the construction sequence, and because it allows smoother external surfaces without the shrinkage marks of internal frames. At present the use of sandwich plating, while utilised for the entire hull on sailing yachts, is only well accepted for deck and sides for motor yachts; for bottom structures single skin remains mandatory, especially for high speed vessels. In Figure 12 the main section of a planing motor yacht with a single skin bottom and sandwich side and deck is shown.

A further complex point for FRP yachts is the hull to deck joint; as reported by Pfund (1999) this is a critical aspect for boats over 30 feet for which hull girder loads become significant. He gives many suggestions for optimization with regard to strength and aesthetics, the latter not to be underevaluated at all.

Fuel storage on board FRP vessels is generally done in stainless steel tanks or in structural tanks integrated in the hull structure in way of a double bottom. The first solution is now well tested and reliable enough where safety and odour are regarded. Structural tanks, in principle, showed many problems especially with regards to fuel seepage. This has been solved by specific treatments such as gel-coating or some other impermeable barrier coat on the inside of the tank. Eikenberry (2009) presents common problems of installation and maintenance of different types of fuel tanks in aluminium, polyethylene, stainless steel and fibreglass. Collision and watertight bulkheads are FRP made with 'top-hat' stiffeners; dividing bulkheads are made of plywood sandwich with insulating panels as the core.

8 PRODUCTION METHODS

8.1 Wood

The traditional building methods of solid wood boats still relies heavily upon the ability and experience of shipyard craftsmen. However the production methods for wooden boats have been simplified by the introduction of plywood and lamellar wood, together with new epoxy based bonding systems which has significantly changed the hull structure lay-out. The lamellar multi layer shell and glued reinforcements avoid pin holes and joints, thus reducing the beam sections and increasing the frame distance. Lighter hull structures are therefore more achievable without sacrificing strength and stiffness. Many motor yachts are still built from wood therefore, typically up to lengths of 30 *m* and speeds around 20 to 25 *knots*. The same advantages of these laminated techniques are used in wooden yacht refits which are, at present, a consistent, profitable and prevalent industrial activity.

A new trend consists in building yachts by combining wood with composites: cedar strips are glued on a structural grid and covered by carbon reinforcements laminated with epoxy resin. Vacuum bagging is widely used for a better structural performance. Wood remains the primary structural material and it works as a mould for the external composite. Epoxy as an adhesive and a coating works much better than polyester

resin with regards wood durability. Composite fabric is usually just a surfacing material that does not significantly contribute to structural strength. This construction method, suitable for medium size sailing and motor boats, has been described in detail by Fox (2001). Boote and Morozzo (2005) presented an experimental investigation to determine the resistance of multilayer beams of lamellar wood and carbon reinforcements for the construction of a racing yacht.

8.2 *Steel and Aluminium*

The construction procedures of a steel/aluminium yacht depend mainly on shipyard facilities and practice. As a general rule hull and superstructures are realised separately. Traditionally the hull was built on a launch slipway, starting from the keel and then adding all frames up to the deck and finally enveloping the whole with the hull shell. At completion the hull is launched and outfitted afloat. Nowadays, the hull is built in a shed allowing better working conditions, especially in colder climates, and then launched by means of trolleys and cranes. Similar procedures are used for aluminium vessels with the only difference that they are often built upside-down to take advantage of the deck as a flat support and reference surface. The lower weight facilitates the overturning operation. Nowadays, the standard procedure for larger yachts (over 40 m) consists in building the hull by blocks, as normally done for conventional ships. Block dimensions depend on their weight and on the lifting capacity of yard cranes; in case the weight is too high blocks are divided in height and, sometimes, in breadth by smaller modules. Modules and blocks are then assembled together in a slipway or in a basin and outfitted when the hull is completed. Preliminary block outfitting is not so common because it would require a very detailed and time consuming design procedure which is not advisable for yachts because of frequent design changes required by the owner. From the shipyard point of view this is the real difficulty in yacht construction management. Accordingly, the attraction of employing 'concurrent engineering' is being increasingly recognised as a boon to superyacht production.

An important issue is represented by the study of new welding techniques oriented to reduce distortion defects and consequent man hours spent in reworking. This problem, rather pervasive in steel constructions, is dramatic in large aluminium structures. Russell and Jones (1997) present a detailed analysis of laser welding advantages and disadvantages with regard to traditional Gas Metal Arc (GMA) and Tungsten Inert Gas (TIG) processes. The main advantages of laser welding are summarised in controlled and predictable component distortion, high joint completion rates, and easy integration with CAD/CAM and CIM operations. In the specific case of aluminium superyachts a reduced distortion means a high saving of filler and fairing work. On this same matter extruded aluminium panels with incorporated stiffener profiles offer many production advantages, especially for ease of deck construction where problems associated with weld distortions during stiffener joining can be mitigated.

8.3 *Fibre Reinforced Plastics*

Production methods for FRP motor yachts are very similar to those adopted for sailing yachts. The main difference is represented by the more complicated shapes of hull and superstructures which necessitate building the vessel from a higher number of components and, thus, a higher number of moulds. For smaller units, up to six to eight metres, two separate moulds for hull and deck/deckhouse are sufficient. The after body of the hull mould is generally a separate part to allow gangway stern shapes. By this solution, it is possible, at a relatively low cost, to make aesthetic changes to the after body and to obtain slightly longer vessels just by substituting

the aft part of the mould. For bigger vessels, deck and superstructures are built by separate moulds as well, but the hull mould is preferably divided into two longitudinal shells to avoid lifting operations when extracting the hull. Then a number of minor components are laminated to complete the structure and the internal outfit.

The majority of FRP motor boats are built by a hand lay-up technique by which every reinforcement layer is laid into an open, female mould and manually wetted and rolled. As FRP material resistance is a compromise between the as high as possible glass content and complete glass wetting, the final material quality depends heavily on workers' experience and shipyard daily environmental conditions (dust, humidity and light conditions). The uncertainty of the material quality is further increased by the need to mix the resin with a catalyst to prime the hardening process: this action, generally carried out manually, heavily influences the material workability time and obliges workers to prepare small quantities of resin before lamination, thus wasting a lot of time. This inconvenience is overcome by the spray lay-up process by which resin and catalyst are sprayed at the same time and with correct proportions on reinforcements by a spray-gun fed by pneumatic air equipment. It is also possible, with a proper gun, to spray cut glass fibres together with resin to obtain an on site chopped strand mat. However, it is not easy to control the glass volume and the resulting thickness and again the material quality depends on worker skill. Nevertheless spray lay-up has the advantage of obtaining a constant, optimal resin/catalyst ratio, a longer workability time and yard efficiency in terms of production, but it still requires rolling operations to consolidate the laminate.

Apart from any other technical concerns, the most serious FRP problem is represented by the styrene fumes released in the working environment during the chemical process of the resin hardening in the mould, which have been proved to be toxic for human health. To overcome pollution new lay-up procedures in closed moulds have been developed and/or are under study. The first, well known solution is represented by the vacuum bag or vacuum consolidation procedure in which an airtight sheet, usually nylon, is used to cover the fibre stack in the mould. Reinforcements are wetted out as with hand lay-up. A set of plastic pipes properly placed in the mould and connected to one or more vacuum pumps allow atmospheric pressure to drive out the excess resin thus increasing glass percentage in the laminate with consequent better mechanical properties.

An improvement to this method, removing the disadvantages of the hand lay-up step and the difficulty in positioning and rolling wet reinforcements is represented by the 'vacuum infusion' process: the main difference with respect to vacuum bag is that reinforcements are placed in the mould when dry, without prior wet out; this allows a major accuracy in the positioning phase and it makes it possible to laminate the hull shell and stiffeners in one shot with significant time savings. The laminate stack is then covered by peel-ply, breather materials, vacuum distribution pipes and an airtight bag. Using vacuum pumps resin is first sucked into the dry laminate stack and then evacuated if in excess. The final result is a very compact product with a high glass percentage, good material quality and repeatability. This latter aspect becomes more and more crucial for large scale production which is currently only relevant for FRP boat construction. Infusion asks for a great care in the preparation of the mould, laminate stack and vacuum circuits: a small error in whatever phase can cause the loss of the whole material. From the environmental point of view by utilising the infusion process personnel have no contact at all with resin and no toxic fumes are dispersed into the working area during polymerization; on the other hand a large amount of

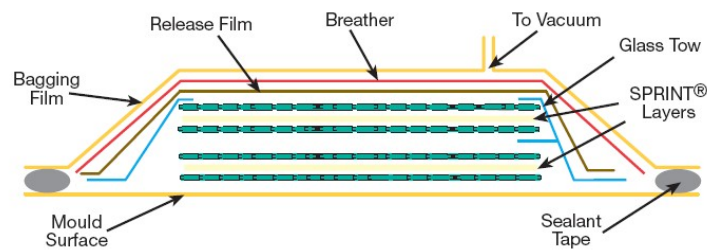


Figure 13: SPRINT system main components.

reject material (peel-ply, breather materials and plastic pipes) is produced for each moulding.

Vacuum bag and vacuum infusion are widely used to build sandwich panels as in one single operation the two skins can be bonded to the core. Moreover infusion is particularly suitable in the case of honeycomb cores, both Nomex and aluminium, because of the reduced bonding surface. It is then possible to produce large components entirely from sandwich construction, such as complete superstructures or decks, in one single process or to laminate partial areas in sandwich within a single skin hull or deck.

As specified in the previous ISSC 2009 V.8 report, vacuum infusion is a general term by which several similar procedures are addressed. Besides well known methods such as SCRIMP a new process has been patented as SPRINT (SP Resin Infusion Technology). SPRINT materials consist of a layer of fibre reinforcement either side of a pre-cast, pre-catalysed resin film with a very lightweight tack film on one face (Figure 13). The material therefore has the appearance of a dry reinforcement, which has resin concealed at its centre and it is produced by a process that differs from conventional prepreg so that the fibres in the reinforcements remain dry and not impregnated by the resin. SPRINT layers are laid up in the mould and vacuum bagged as for conventional prepreg. When the vacuum is applied, the air transport properties of the dry reinforcement enable air trapped in the fibre bundles and between layers to be easily removed, reducing the void content to extremely low values. When the temperature is then raised for the cure, the resin film softens and flows into the air-free reinforcement.

The benefits and drawbacks of the infusion process have been widely assessed in the ISSC 2009 V.8 report for sailing yachts and they remain for motor yachts. In short, whatever the type of process, vacuum infusion allows to reduce pollution in the work environment and to increase FRP mechanical properties, reaching glass percentage in the laminate close to an average of 60 % in weight (Boote *et al.*, 2006). At present the trend within shipyards is to apply more widely this methodology to bigger vessels and components and in many cases they build FRP motor yachts with lengths over 20 m completely by the infusion technique, and other FRP components, such as decks and superstructures, for vessels up to 40 m.

The present trend to control production cost is represented by modular construction by which the vessel components are moulded separately and then assembled by bonding. From this perspective an accurate study of the minimum number of moulds and their optimization becomes very important for the industrialization process and cost reduction. The first applications of this method regards the realization of FRP counter-moulds in which the housing for furniture and fittings, and some furniture

themselves, are included in the mould to speed up the interior furnishing; the tray was then glued to the hull structures, partially contributing to the hull strength. The second step is to build separately the hull shell and the reinforcement grid (also called 'spider structure') in two separate female moulds and then to glue them to each other; in this case the bonding procedure and the choice of the best suitable type of adhesive is more complex. Strand (2002) gives a comprehensive set of guidelines about modular construction, highlighting the problems coming from gluing fresh, uncured polyester laminates to cured ones. The solution is to use methacrylate adhesives which are proven to have strong adhesive capabilities coupled with acceptable elasticity.

9 OUTFITTING

Outfitting covers the whole fit out of a vessel from the interior to the exterior, engineering to aviation, bridge integration, luxury owner supplied items or toys and stores. As far as the conventional naval architect is concerned, weight estimates consider machinery separately from outfitting which includes the rest of the engineering systems and interior fit-out, the total lightship being made of hull structure, machinery and outfitting (a rational approach to weight estimation of fast crafts can be found in Daidola and Reyling, 1991). In this report however, outfit considers all installations that are not fixed parts of the ship hull. The issue of outfitting on structural requirements is crucial not only in terms of the fundamental systems that allow the operation of a vessel but in the case of luxury sailing yachts and superyachts in the features that provide the definition of luxury, including heli-decks, large open volumes, swimming pools and internal harbours and garages. However the complexity of outfitting varies between vessel types and the regulations underpinning vessel design and operation can vary between the very limited applied to a private yacht to those unrestricted charter vessels carrying more than 12 guests that therefore require cargo or passenger ship certification.

The role of CS in the outfitting process is considerable. While a class surveyor might inspect a hull moulding on two or maybe three occasions, the bulk of the surveyor's contact with the vessel will be in the pre-outfit and outfitting stages of the build. Particular attention in the early outfitting stages is paid to the structural complications referred to in the following sections, but clearly as outfitting progresses there is an increasing focus on systems installation. As stylists and designers become more innovative with materials and furnishings, including the increasing use of glass (Freivokh *et al.*, 2010), dependent on the classification of the superyacht, materials used in outfit must meet SOLAS approval. The relevant standards in all aspects of super yacht design and operation are reported in more detail in Chapter 3.

Outfitting also needs to be considerate upon the basic requirements of a charter crew required to operate the ship and the Maritime Labour Convention (MLC) ensures a minimum requirement for crew space which from an owner perspective can impinge on space available for guest and owner designated areas (The Superyacht, 2011).

9.1 Structural Challenges

A typical flow chart of the construction and outfitting process of a superyacht is shown in Figure 14. Outfitting is by far the longest and most complex process in superyacht production taking typically up to two thirds of the production time and up to 80% of the superyacht production cost. The inclusion of large spaces, maximising internal volume and integrating systems to accommodate comfort, luxury and toys present significant structural challenges and issues relating to compliance vary from

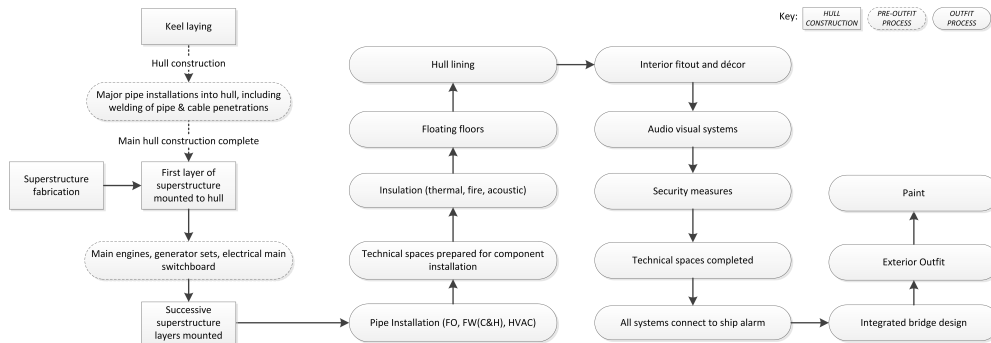


Figure 14: Generic hull construction and outfitting sequence for superyachts (Meijers, 2003)

the impact of the International Convention on Load Lines upon window sizes, to side shell openings, hybrid material connections (for example, aluminium superstructures explosively welded to steel hulls) and the stowage of toys to discontinuities in primary structure.

Optimising structural arrangements to ease outfitting imposes again the problem of transverse versus longitudinal framing structure. As vessels become larger, the relative increase in bending moment is influenced by the square of the length (and relatively modest increases in beam). Accordingly, as reported by Roy *et al.* (2008), the longitudinal framing appears to be more efficient because of deep and less frequently spaced transverse frames. With longitudinal framing, the length of pipe runs for outfit can increase but the weight savings in this alternative construction are penalised when the longitudinals' depth is increased to take cutouts required for HVAC systems to make transverse turns. 'Tween deck height is increased and longitudinal stiffening can lead to deep recesses in way of hull windows which often conflicts with client and interior design requirements. Hybrid framing systems whereby the side shells are transversely framed, allowing greater flexibility in vertical routing of services and decks longitudinally stiffened, appear to offer the best compromise for structural efficiency, maximised internal volume and accommodation for service runs. In addition longitudinal and hybrid framing best matches dimension increases. In Figure 15 two examples of the typical routing and 'tween deck space available for running service pipework and cabling is shown.



Figure 15: Typical 'tween deck depths for service runs (images courtesy of BMT Nigel Gee).

Superyacht systems are concealed and run in the most space economical manner possible. This can make access for installation (and still worse access for rework) extremely difficult, especially in the later build stages. As such the larger the amount of pre-outfit (particularly underfloor/behind carcass system installations) which can be carried out prior to the joining of hull and deck shells, the less access is a concern and the more man-hours are saved. As vessels get larger, the likelihood of penetrating primary structure for service runs increases. Meunier and Fogg (2009) presented research findings that show adding cut-outs within components such as structural bulkheads, will create an area of local stress concentration. With advances in 3D CAD/CAM software and availability, Meunier and Fogg reasoned that ensuring system penetrations are added in a non-critical areas for structurally efficient design is achievable at the earliest of design stages.

For craft constructed from composites penetrations in stiffening elements, such as deck beams, girders and frames, will be subject to both local reinforcement prior to penetrations being cut and local consolidation to ensure maximum structural continuity. Design criteria govern the maximum size, minimum spacing and overall geometry of penetrations to best preserve the global effectiveness of the stiffening. Finally the finishing details will be specified to maximise the strength and life of the penetration/stiffener interface.

Similarly, through hull penetrations must be carefully designed to maintain the local structural integrity of the hull. For composite vessels, the largest penetrations will be implanted into the hull mould to guarantee structural integrity and wherever a significant penetration is foreseen the local hull core will be chamfered out to reduce risk to the structure. Standardised fittings will then be used for monolithic laminate penetrations with specified fitting and finishing details and procedures. Where it is necessary to remove hull core in way of a penetration the core will be consolidated and made watertight in way of the cut-out to avoid water ingress as a result of damage in service.

A description of most effective outfit solutions on a motor yacht is presented by Lalan-gas and Yannoulis (1983): to reduce noise and vibration diffusion through the hull and superstructure spaces all the interior technical outfit is arranged with 'floating' floors and walls in such a way as to isolate as much as possible passenger areas. Insulating systems consists of paints, filler and panels applied to the internal cabin surfaces. In some cases to reduce the noise from hull wash, some planks are directly applied to the internal shell at the waterline. All these devices heavily influence the final displacement of the ship and a very refined structure scantling becomes mandatory.

9.2 Rework and Refit

Luxury superyacht builders report that rework can add thousands of man-hours to a project and may be a result of design or production errors at an earlier stage, unforeseen complications or client specified changes. The severity of the rework requirement clearly varies depending on the cause and timing, but in general terms, the earlier any required rework is carried out the fewer complications it will in turn result in.

For steel ships, a lot of the final challenge in production rests in fairing and painting the hulls to achieve the gloss finish required by the owner. Exterior rework to rectify weld distortion and fit is often avoided by using this fairing process and significant yard investments can be made in automated fairing compound applicators. Epoxy fairing compounds are stable as a coating but even with the low density bulking property of

added glass microspheres, an average application of 20 mm for a 60 m yacht equates to an extra mass of 20 tonnes (approximately 2.5 % displacement mass).

Structural deformation (especially in aluminium) due to thermal loading is a big problem as well, both in terms of adhesion of fairing compound but more so in terms of cosmetic rippling, exaggerated by high gloss paints (dark or light). As the displacement mass is not so much of a concern to the superyacht designer (these vessels are rarely optimised in terms of power to weight) and if internal volume remains unaffected, then thicker plates provide less thermal distortion and little impact to cost (hull construction materials account for approximately 10 % of the overall yacht production cost). A problem exists however if structural mass for a stiffened plated section increases faster than increasing stiffness gains, then resonant frequency drops and vibration amplitudes from machinery noise, hull/water interaction and propeller excitation increase.

Owing to the huge number of existing vessels and their intrinsic value, the maintenance and refit market of yachts is a growing activity in yacht industry and it represents a source of steady flow, with the consistent by product of maintaining the value and good conditions of yachts, this aspect determinant for the top brands. Refit in particular represents a real new resource especially in recession periods, like the present one seems to be, and it is mainly oriented to big ships for which the value of the steel vessel is large enough to worth the business. This activity is carried out by conventional shipyards together with new constructions or, even more often, by specialised societies. Even if mainly oriented to interior work, often refit covers structural matters as well, especially in the case of older units. Most common interventions regard the modification of stern steel structure to achieve larger bath areas and/or to add a stern door to allow garage access, the addition of bulbous bow, the lengthening of aluminium superstructures, the addition of fixed or folding helicopter landing areas. Particular attention must be devoted in refit planning because all these works deeply influence ship weight and stability conditions and must be carried out in accordance to in force CS' rules. Refit project and work are often more difficult than for a new construction because it is impossible to know what to expect until the beginning of operations. In addition there are fewer degrees of freedom with respect to a new construction because it is not possible to change more than to a certain extent the aesthetic and functional nature of existing structures. Some important aspects of yacht refit are presented by The Superyacht Intelligence (2011) together with a long list of recent refit work carried out by most important refit shipyards.

9.3 Stability and Fire

Volume and expected mass of outfitting are determined early on in the design stage: Hulseman *et al.* (The Superyacht, 2010a) point out that it is important in the apportioning of available space and volume that the outfitters are consulted early so that the requirement for technological system space for, especially interior, outfit is recognised and properly accounted for. Vessel statics and operating dynamics are affected by the mass disposition: for a superyacht where capacity is important, the deadweight to displacement is low compared to, say, a cargo vessel which necessitates large deadweight carrying capability. In the latter design stages, superyacht stability can therefore be affected off initial design by changes in fitout to satisfy fickle customer requirements, although rarely is this shown to be significant. What is of more concern is the disposition of the deadweight, which whilst low (15 % -20 % of total weight) is constituted principally by consumable fuel load (60 – 80 %) which is deep in the vessel

(Roy, 2006). This results in light arrival conditions which are challenging with regard to static and dynamic stability criteria. It is the norm therefore that stabilising devices are fitted for comfort at anchor which puts more burden on structural requirements to accommodate these.

A large challenge is in the use of recreational fun tools, shortly called ‘toys’, whilst at anchor. Most superyachts are fitted with big tenders, jetskis, sports cars and so forth; helicopters are the present vogue. As an example the superyacht *Le Grand Bleu*, carries two tenders: a 62’ Sunseeker and a 72’ Baltic sailing yacht, and is equipped with two helipads. These changes in static stability must be accommodated by increased structural design and ballast arrangements but without compromise to internal volume. At present the concept of a dedicated vessel supporting the mother ship to carry toys is realised by the ‘shadow yacht’, the ‘toy box of the sea’ as defined by Sime *et al.* (2009); in their paper the ideal technical requirements for such a kind of vessel are described and a number of possible design solutions are presented.

Helicopters provide significant outfitting challenges in that the regulations governing the platform design and supporting infrastructure often clash with customer requirements and exterior styling. Articulating and folding platforms are the common solution but come with incumbent structural design impacts. A clear summary of design guidance for helidecks is presented by Strachan and Lagoumidou (2009).

MCA-LY2 requires all enclosed compartments in the hull and below the freeboard deck that are provided with access possible through openings in the hull (for example, inner harbours and garages) should be watertight doors fitted with alarms connected to the bridge. The actual openings in the hull should comply with SOLAS II-1/25-10 External Openings in Cargo Ships.

Swimming pools and SPA baths are considered to be ‘recesses’ (under LY2) and as such, as it is not practicable to drain them within the 3 minutes requirement, intact and damage stability must be considered accounting for the mass of water and free surface effect. Damage stability is assessed through ICLL or LY2. Vessels of 80 m L_{OA} and above need a SOLAS one-compartment standard of subdivision. As vessels become increasingly longer, 2 compartment standard of subdivision becomes more normal which has positive benefits for exterior designers in the siting and provision of life-saving appliances. Refit or major alterations require new inclining experiment checks on lightship stability when either the displacement has increased by over 2 % or the L_{CG} has changed position by more than 1.1 % or the V_{CG} has changed by more than 0.25 % or at renewal survey every 5 years.

According to insurance claim records, the greatest danger to superyachts in terms of financial loss is fire in harbour (The Triton, 2006). Under the MCA-LY2 Regulations 14B.2 and 14B.2.14, all accommodation and service spaces except those not of high fire risk (sanitary spaces, etc) for a superyacht carrying up to 12 passengers must have an automatic sprinkler, fire detection and fire alarm system. This is not however mandatory for a superyacht that falls under the SOLAS Regulations if it carries less than 36 passengers. However if the automatic system is installed, then the fire integrity standard of the bulkheads and decks can be reduced according to SOLAS II-2/9.2.2.4. So at first sight it appears that there are structural and outfitting cost savings to be made by certifying the vessel as a passenger vessel and satisfying SOLAS rather than gaining certification through MCA-LY2. However, a big impact in the construction of a yacht under SOLAS rather than MCA-LY2 is in the restricted use of combustibile materials and how the fire doors are constructed (Fanciulli and Moretti, 2009). Fire integrity of

divisions (under SOLAS or LY2) needs to be maintained at openings and penetrations which can lead to practical complexities following Gurit's findings (Meunier and Fogg, 2009) on increased primary structure penetrations with the increasingly larger vessels being built. One example impacted by luxury outfit requirements comes in the shape of saunas and steam rooms where an 'A' class boundary is required.

9.4 Security

The increasing size of the superyacht fleet, their inherent unit value per ton and the value of the guests belongings and the yacht's freedom to roam make them attractive targets for criminal and terrorist activity. The subject of security on ships and yachts is regulated by the International Ship and Port Facility Security Code (ISPS, IMO 2002) which is a comprehensive set of measures to enhance the security of ships and port facilities. The ISPS Code was adopted by a Conference of Contracting Governments to the Solas 1974, convened in London (December 2002). The Code aims to establish an international framework for co-operation between Contracting Governments, Government agencies, local administrations and the shipping and port industries to detect security threats and take preventive measures against security incidents affecting ships or port facilities used in international trade and to establish relevant roles and responsibilities at the national and international level.

From the operative point of view some measures can be adopted to enhance the security of yacht when at rest and sailing. Figure 16 shows some of the common security measures being incorporated into superyachts:

- thermal imaging cameras for day and night vision mounted in high and protected positions;
- underwater cameras mounted at fore and aft to verify approaching divers;
- underwater lighting to control the yacht surroundings both below and above water;
- gangway entry video-phone to control entrance when in port;
- radar based detection systems to individuate approaching craft;
- long range acoustic guns, high pressure water guns, pepper guns;
- 'shadow yacht' carrying security guards to support the mother ship.

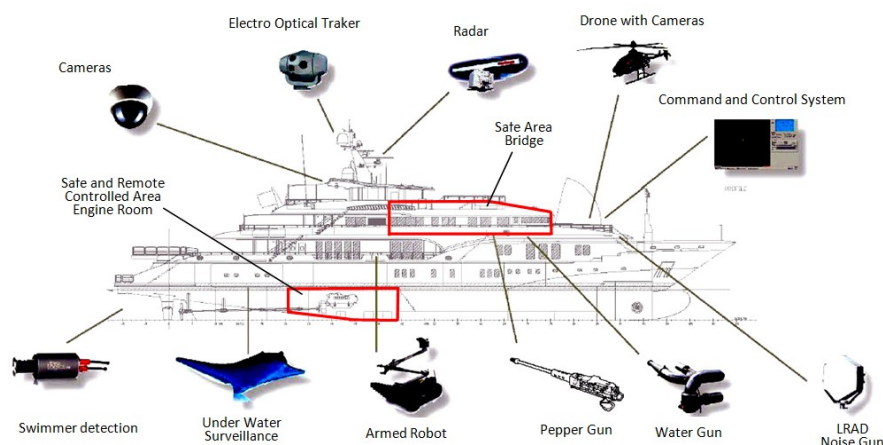


Figure 16: Security devices available for superyachts (by courtesy of Nobiskrug Shipyards)

Two issues regarding structural capability therefore exist. Firstly, the introduction of security measures requires integration into the yacht's electronic systems, placing more burden on the limited 'tween deck space and structural penetrations for cable runs. Secondly, extreme protection is provided by the hardening of safe areas of the yacht, typically the bridge, to mitigate high velocity rifle rounds and blasts. The engine room must be locked remotely and all essential cable trays protected in order that the bridge has full control over the yacht (The Superyacht, 2010). All the considered solutions can be carried out on existing yachts but could be more efficient if integrated in the design phase and realised during the vessel construction. The integration of a system always means higher efficacy and lower costs with respect to a retrofit intervention.

10 SAILING YACHTS

The report of the V.8 ISSC Committee on Sailing Yacht Design (2009) had an extensive discussion on materials selection, fabrication techniques, and design procedures for sailing yacht hull, rig and appendage structures. This chapter is an update of that report, citing the recent work that has been published and some other papers not quoted in the 2009 report.

10.1 Hull Design and Structures

During the past 3 years, the majority of the research into sailing yacht problems seems to be in the application of advanced numerical techniques to sailing yacht design. In particular, computational fluid dynamics (CFD) and finite element analysis (FEA) are being used to more accurately predict the loads on a yacht hull and the responses of the hull structure to those loads. Many CFD applications investigating the determination of loads for structural design are available in literature. Similarly the use of finite element analysis (FEA) in analyzing sailing yacht structures is increasing with the improvements in software and hardware. Fornaro (2011) discusses in detail the entire process of using FEA to analyze the behaviour of composite yacht structures from pre-processing to post-processing. The pre-processing includes meshing, ply properties, laminate definitions, element orientation, global ply tracking and load case development. Post-processing topics include principal stresses, failure indices and strength ratios. Most FEA analyses for composites use linear static solution methods that imply an assessment of strength based on the first-ply failure. Nonlinear solutions allow progressive ply failure analysis (PPFA) by sequential degradation of stiffness for the first and subsequent plies failing until complete failure of the laminate has occurred. The results of PPFA are a better understanding of the nature of failure in a given area and the amount of reserve strength following initial ply failure.

The optimization of composites is much more difficult than for isotropic materials because of the increase in the number of possible design variables such as number of plies, ply thickness, fibre orientation, core material and thickness, etc. Anderson (2008) provides an overview of common optimization routines and briefly discusses their good and bad attributes. Achieving a robust, optimum solution for a composite structure requires not only a good understanding of FEA and composite structures, but also the optimization process being used and how composite structures are manufactured; attempting an optimization without this knowledge will likely result in problems.

Most composite yacht hull and deck structures use some type of cored construction to save weight and cost. High pressure slamming loads can cause significant damage to cored hulls. Two recent papers discuss the design of cored composite structures for dynamic slamming loads: Battley *et al.* (2008) experimentally characterized the

hydroelastic responses of composite hull panels. Panels were tested at a deadrise angle of 10° and a range of impact velocities. Results showed that stiffness has a significant effect on the responses of the panel to a slamming-type load. Flexible panels had reductions in the peak pressure at the centre of the panel and increases near the chine edge of the panel, possibly due to the panel deflections that caused a reduction in the local deadrise angle. Islin and Lake (2008) studied low cycle-high elongation fatigue performance of foam core materials. Four cores were tested including PVC foam, two cross-linked PVC foams and a styrene acrylonitrile (SAN) foam. When subjected to slamming loads, significant differences were found between the cores. The three PVC cores retained or in some cases increased the area under their post-fatigue residual strength load-deflection curve. On the other hand, the SAN core showed a significant reduction in shear energy absorption and elongation after fatigue loading, indicating that this material may not be suitable for areas which would be subject to recurring slamming events.

A wide analysis of most relevant aspects of large sailing yachts made in composite materials is contained in the paper by Meunier and Fogg (2009) where they take into particular consideration hull girder strength, fore and aft structural requirements and the influence on hull structure weight. Comparative analysis of single skin and sandwich solutions is carried out. The paper closes with some considerations about structural versus aesthetics and comfort requirements. The structural behaviour of cruise and racing yachts from the comfort point of view is illustrated by Payne and Siohan (2008) who highlight the common conflicts that arise when integrating structures with the interior requirements. Battley (2011) in his paper considers specifically structural characteristics relative to slamming loads on sailing yachts.

10.2 Mast and Rigging

There have been relatively few publications on developments in mast and rigging analysis and design compared to the sails which they support. Rizzo and Boote (2010) present the structural design of mast and rigging from a practical viewpoint, highlighting main idealization concepts of structural behaviour. After a detailed illustration of the analytical available procedures and applicable rules, they discuss more complex scantling procedures, with particular attention to nonlinear finite element analyses, able to take into account nonlinear large deformations and slacking behaviour of rigging and sails. Some applications on a typical modern sailing yacht rigging are carried out as well. The design of mast and rigging is made more difficult by the uncertainty of sail loads transmitted to the rigging. The measurement of these loads in real scale is becoming a necessity especially for large sailing yachts. A measurement system to be fitted on Perini Navi sailing yachts has been developed recently by the shipyard at the Department of Naval Architecture of the University of Genova (Rizzo *et al.*, 2009).

Chapin *et al.* (2011) have considered fluid structure interaction in the design of yacht sails and rig using a viscous flow solver and a nonlinear finite element code which are loosely coupled. By iteration and using a genetic algorithm, optimum sail shapes can be investigated, and the loads transmitted to the mast and rigging estimated. Augier *et al.* (2011) have carried out a full-scale study in a J80 yacht, making simultaneous measurements of navigational parameters, yacht motions, sail shape and loads in the standing and running rigging in unsteady sailing conditions. These measurement results were compared to a fluid-structure interaction numerical model and a good comparison was found.

The advantages of using streamlined carbon fiber rigging as opposed to conventional

round rod rigging are discussed by Martin *et al.* (2011). By using a VPP for an IMS 40 yacht, they found that victories of 3 to 10 boat lengths could be obtained for both windward/leeward and Olympic courses.

10.3 Appendage Design and Construction

Similar to the advances in hull design and construction, most of the published work in the last few years has been on the application of advanced numerical methods. However, the paper by Keuning and Verwerft (2009) gives a new method to compute the lift forces on a keel and rudder of a sailing yacht based on the extensive data obtained from testing the Delft Systematic Series of yacht hulls. The final results are formulas for the lift on the keel and rudder that take into account the interference effects of the yacht hull, the aspect ratio, the sweep back and the downwash effects of the keel on the rudder. Orych *et al.* (2008) use potential flow methods coupled with a boundary layer code in order to study the effects on keel winglets on the lift and drag. Hutchins (2008) used a RANS code in conjunction with a VPP to determine the effects of candidate bulb shapes on the overall yacht performance. Canting keels are increasingly popular for high performance racing yachts. However, the canting keel imposes unique loading situations on the yacht structure. The structure needs to be strong enough to withstand the very high loads generated by slamming and grounding and yet light enough to not counteract the advantages of the moving ballast in the first place. Campbell *et al.* (2006) discuss the development of the Volvo Open 70 Rules regarding structural requirements for canting keels with particular regard to the safety considerations. Cowan and McEwen (2006) discuss the relative merits of various structural configurations and the use of FEA to analyze the keel configurations. Other practical aspects of canting keel are presented in Tier *et al.* (2006).

11 CONCLUSIONS

Superyachts, both motor and sailing, are very special marine products which lie outside the common criteria for the design and construction of conventional ships. Even if, in many cases, performance requirements continue to be the driving key of the project, the most binding aspects concern more the interior and external design rather than structural issues. Thus the stylist becomes the project leader and the engineer has to manage to fit the boat around the stylized design. This sometimes gives lots of problems/restrictions on the structural engineering side as well, but also commits the engineer to develop very clever and, often, innovative structural engineering solutions.

Furthermore, given the high intrinsic value of superyachts, every owner wants something special, new, and better than what the other owners have. This again makes yachts an ideal platform for research and development of engineering techniques and technologies to reach maximum passenger comfort, highest luxury levels and structural improvement as well. Regardless, the reliability and safety of the vessel is expected and this is reflected in the design and scantling of hull structures. From this point of view, whilst small and medium size yachts have their own rules and design procedures from Classification Societies, whereas larger yachts fall within conventional ships or HSC Regulations, the following trends and research expectations are common:

- light structures to reduce ship weight, construction cost and fuel consumption;
- structure optimization to allow for larger internal volumes;
- reduction of vibration and noise;
- material developments with particular emphasis on new composite 'eco' products and related emerging technologies.

As for conventional ships, many problem areas are still unexplored or, at least, unsolved. As far as future research on superyachts is concerned, the following aspects deserve for further investigation:

- direct application of structural optimisation techniques during the earliest design phases;
- parametric procedures for hull structural scantlings which can rapidly accommodate the changes requested by the owner, with low cost and with the possibility to evaluate the consequences of different alternatives;
- increase in the size of FRP vessels in order for superyachts to benefit from relatively low vibration behaviour and hull maintenance;
- integrated use of CFD and FEM techniques to achieve and apply realistic loads on innovative structures.

Finally, special consideration must be directed to outfitting: while building the hull structure takes one year, at least two or more years are required for completing the vessel. Particular attention should be devoted to improve outfitting design and production methods by use of automation techniques, such as fairing and painting, or modular construction for piping and furnishing. Given that safety and reliability remain imperative, it's the 'toys' and the systems that the owner is more interested in, and although this in itself is not strictly a structural issue, it does have serious consequences in terms of structures.

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