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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

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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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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.

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- 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

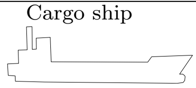
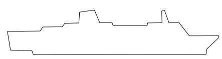
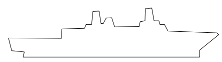
	Cargo ship 	Cruise vessel 	Frigate 
General specification: Similarities	Worldwide operation Open sea and coastal waters All weather High reliability, Year round operation Survive all weather and sea conditions Docking at planned intervals, plus emergency situations Grounding/collision damages	Similar 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

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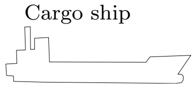
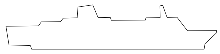
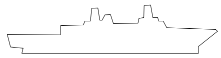
Table 2: Structural similarities

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

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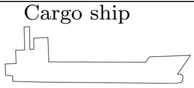
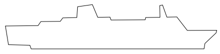
Table 3: Generic differences between naval and commercial structural design

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

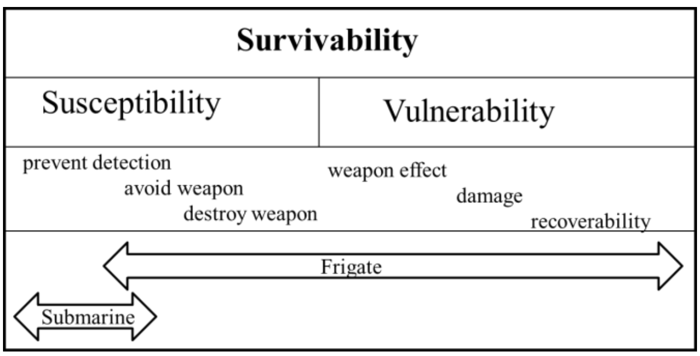


Figure 1: Survivability elements for a typical frigate and a submarine

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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

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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

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Table 4: Limit States/Mechanical Properties

Limit State Criteria	Ordinary Strength Steel ($350\,MPa$)	High Strength Steel ($550-690\,MPa$)	Aluminium: Conventional Welded Plate (5xxx series)	Aluminium: Extruded Panels and Shapes (6xxx series)	Titanium (Ti-6Al-4V Gd. 5)	Fibre Reinforced Plastic: Glass Fibre	Fibre Reinforced Plastic: 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 *all* 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.

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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:

$$\frac{(Value_{lightweight} - Value_{nominal})}{Value_{nominal}} \cdot 100\,\% \tag{1}$$

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 %

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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:

$$I_i = I_o + A \cdot d^2 \tag{2}$$

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

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Table 7: Other Behaviours Affecting Structural Weight in Naval Vessels

Other Properties	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
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 when exposed to sea air, but not seawater.	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 conductively 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 conductively and will rise in temperature quickly. The critical temperature is about 195° C.[1]		Superior resistance to thermal loading. The thermal conductively 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 strength.	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.

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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 $128\,m$ high speed ferry operating at $42\,kts$ (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.

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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 a in 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

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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.

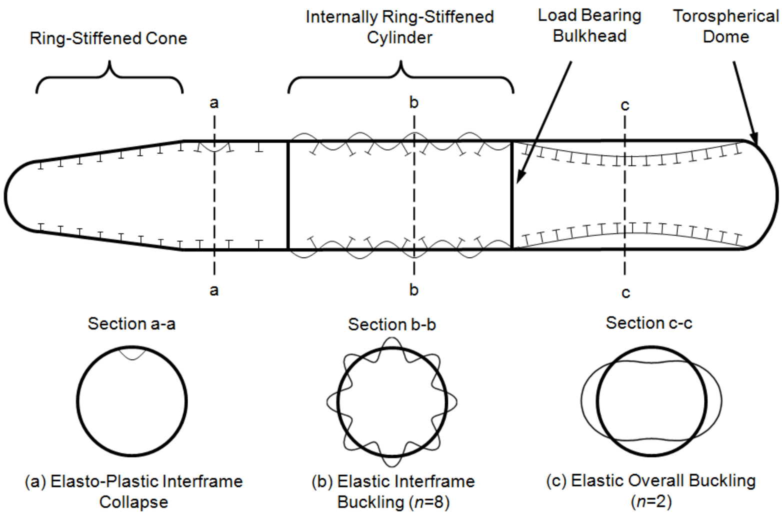

Figure 2: Typical pressure hull structure and buckling modes

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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 ($> 500\,MPa$) 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).

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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

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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

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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 ±6 % of the experimental values. Graham (2008) later extended his FE analysis to a fourteenth test specimen, overpredicting 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.

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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.

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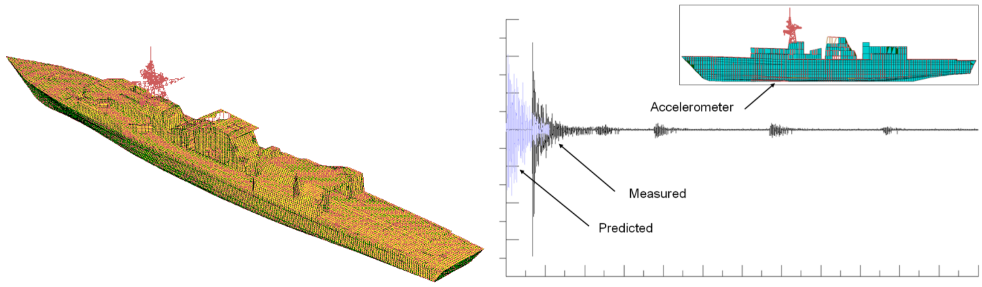

Figure 3: Full ship FE model for the USA prediction correlated with FSST measurement.

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 NASTRAN, 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.

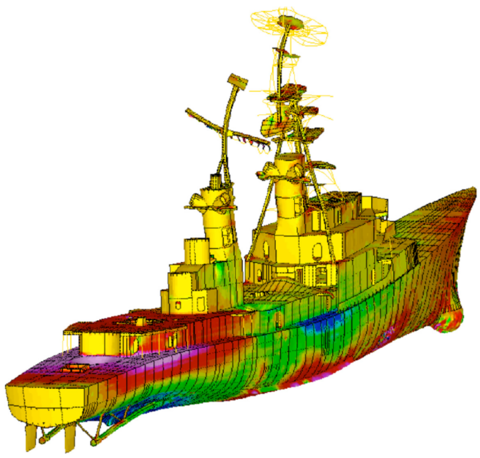

Figure 4: Structural model of the destroyer Lütjens. Deformation under lateral blast.

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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 Nº 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

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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-

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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

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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

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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 $508 \times 508 \times 3.4\,mm$ 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 $E = 207000\,MPa$, $yield = 350\,MPa$ and $Et = 20875\,MPa$ (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

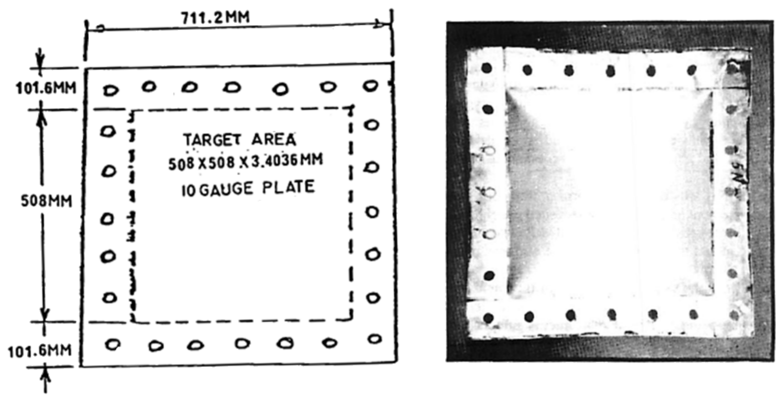

Figure 5: Square plate dimensions and final deformed shape

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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.

$$\left(\frac{\delta}{t}\right)_r = \frac{(3-\xi_0)\left\{(1-\Gamma)^{1/2}-1\right\}}{2\left\{1+(\xi_0-1)(\xi_0-2)\right\}} \tag{3}$$

where

$$\Gamma = \frac{2\rho_p V^2 a^2 \beta^2}{3\sigma_y t^2}(3-2\xi_0)\left(1-\xi_0+\frac{1}{2-\xi_0}\right) \tag{4}$$

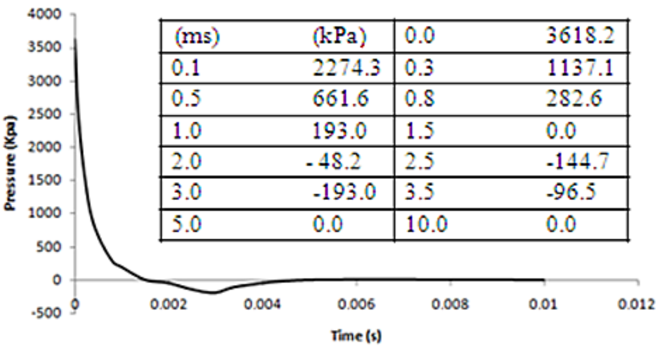
$$\xi_0 = \beta\left\{(3+\beta^2)^{1/2}-\beta\right\} \tag{5}$$

$$\beta = \frac{b}{a} \tag{6}$$

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.



(ms)	(kPa)	0.0	3618.2
0.1	2274.3	0.3	1137.1
0.5	661.6	0.8	282.6
1.0	193.0	1.5	0.0
2.0	- 48.2	2.5	-144.7
3.0	-193.0	3.5	-96.5
5.0	0.0	10.0	0.0

Figure 6: Load time history

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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	×-sh	×	64.2	36.2	355.7	342.2
SS	5×5	ABAQUS	explicit	×-pp	×	64.2	36.0	272.6	236.6
SS	12.7×12.7	LS-DYNA	explicit	×-sh	×	62.7	36.0		
SS	12.7×12.7	LS-DYNA	implicit	×-sh	×	62.7			
SS	5×5	ABAQUS	explicit	×-sh	×	64.2	36.0	292.5	261.9
SS	5×5	ABAQUS	implicit	×-pp	×	64.2	36.0	406.8	352.4
SS	5×5	ABAQUS	implicit	×-sh	×	64.2	36.0	406.8	352.4
SS	20×20	ABAQUS	implicit	×-pp	×	64.2	35.8	226.1	210.7
SS	10×10	ABAQUS	explicit	×-pp	×	64.2	35.7	292.3	262.2
SS	10×10	ABAQUS	explicit	×-sh	×	64.2	35.6	295.8	264.9
SS	20×20	ABAQUS	implicit	×-sh	×	64.2	35.4	251.6	319.8
SS	10×10	ABAQUS	implicit	×-pp	×	64.2	35.3	211.6	350.0
SS	20×20	ABAQUS	explicit	×-pp	×	64.2	35.0	396.0	350.0
SS	20×20	ABAQUS	explicit	×-sh	×	64.2	35.0	397.4	350.0
CL	5×5	ABAQUS	implicit	×-sh	×	118.8	34.3	401.8	352.0
CL	10×10	ABAQUS	implicit	×-sh	×	118.5	34.1	401.5	351.2
CL	5×5	ABAQUS	explicit	×-pp	×	118.8	34.1	396.8	350.0
CL	5×5	ABAQUS	explicit	×-sh	×	118.8	34.0	399.2	352.5
CL	10×10	ABAQUS	implicit	×-pp	×	118.5	33.8	384.9	335.7
CL	10×10	ABAQUS	explicit	×-pp	×	118.5	33.7	329.8	333.0
CL	10×10	ABAQUS	explicit	×-sh	×	118.5	33.7	346.2	341.1
CL	5×5	ABAQUS	implicit	×-pp	×	118.8	33.4	227.3	350.0
CL	20×20	ABAQUS	implicit	×-sh	×	118.5	33.3	339.1	351.2
CL	20×20	ABAQUS	implicit	×-pp	×	118.5	33.0	307.0	380.0
CL	12.7×12.7	Dytran	explicit	×-sh	×	117.0	33.0		
CL	12.7×12.7	LS-DYNA	explicit	×-sh	×	115.74	33.0		
CL	20×20	ABAQUS	explicit	×-pp	×	118.5	32.8	194.0	230.9
CL	20×20	ABAQUS	explicit	×-sh	×	118.5	32.7	187.0	235.1
CL	42.3×42.3		explicit	×-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	×-sh	×	117.4	26.0		
CL	12.7×12.7	ANSYS		×-sh	×		23.0		
	42.3×42.3			×	×		23.0		

SS – Simply Supported boundary,
CL – Clamped Boundary,
pp – perfectly plastic,
sh – strain hardening

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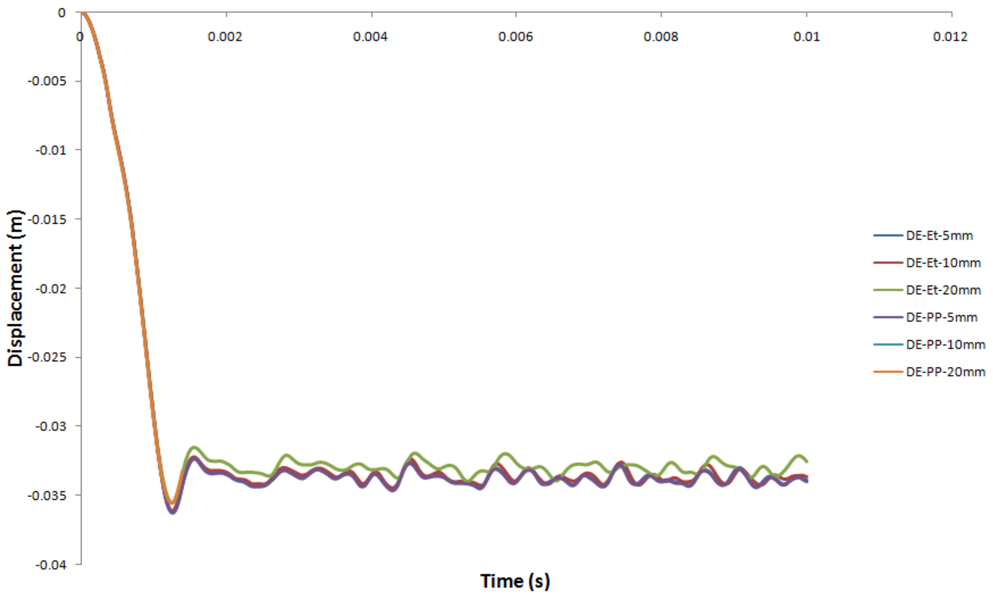

Figure 7: Results of displacement vs. time for explicit solution and strain hardening (Et) and perfectly plastic (PP) nonlinear FEA with differing mesh size

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 nonlinearity 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

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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.

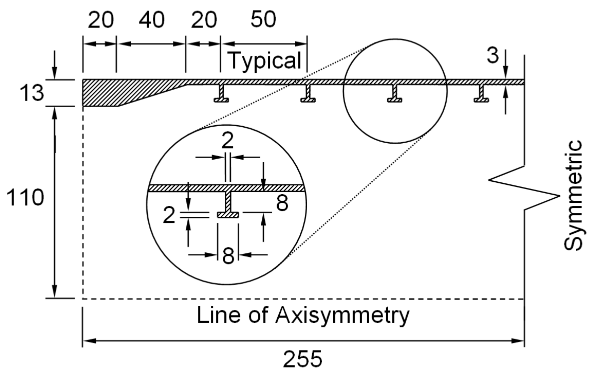

Figure 8: Nominal dimensions of test specimens

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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 ±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.

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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.

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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	:. 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-

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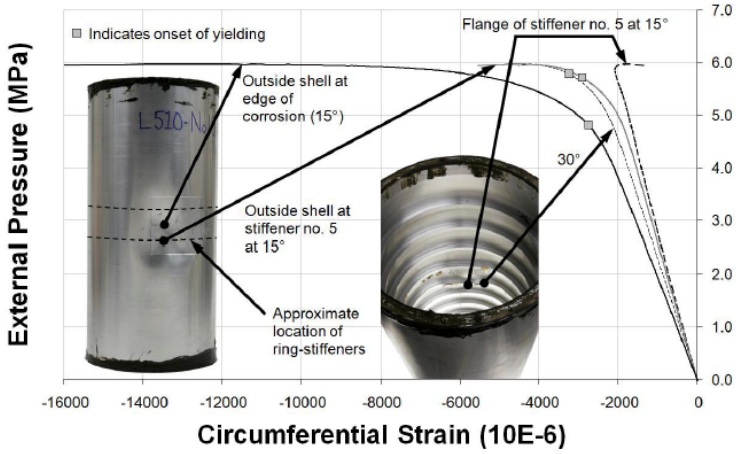

Figure 10: Experimental result of cylinder with corrosion patch

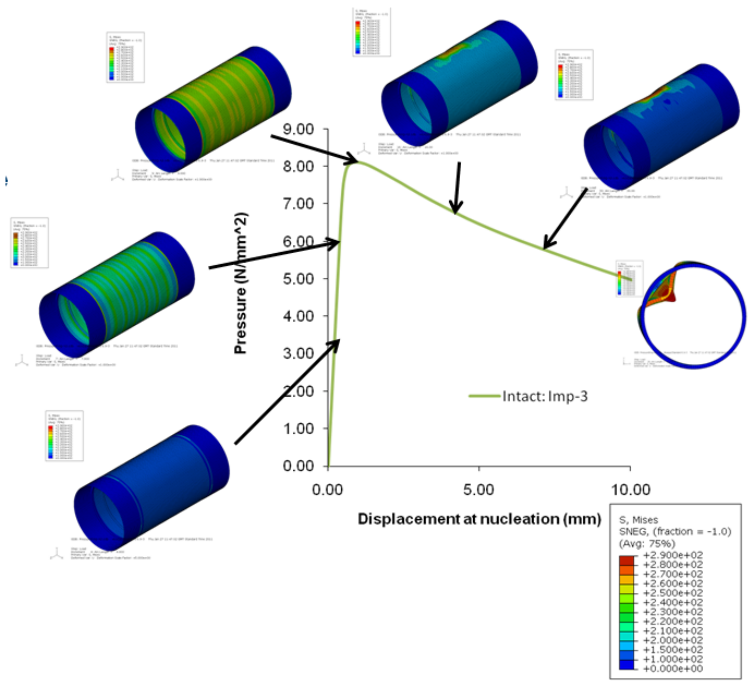

Figure 11: Elasto-Plastic collapse process for undamaged cylinder

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×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

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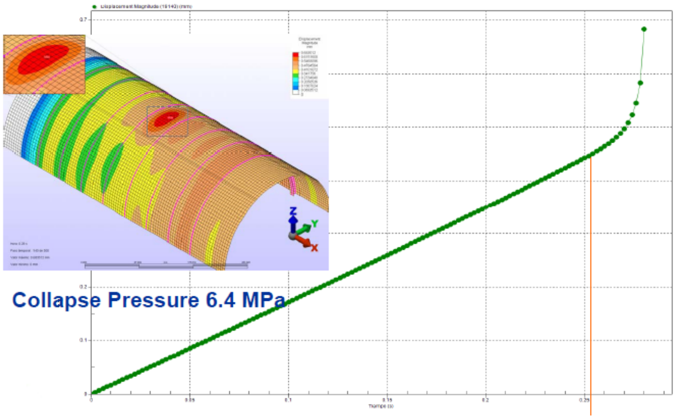

Figure 12: Results for model with corrosion patch

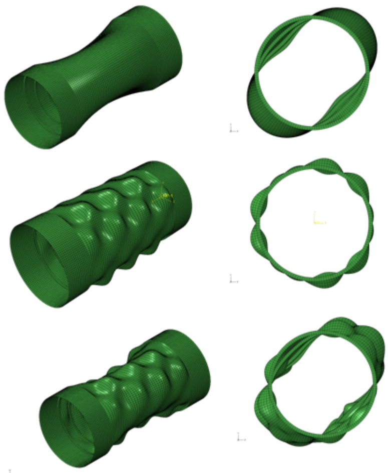

Figure 13: Overall, interframe and combined OOC shapes

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-

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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

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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 theretical/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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