Navy Ship Underwater Shock Prediction and Testing Capability Study

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14. ABSTRACT

Underwater mines have long been a major threat to ships. The most probable threats are non-contact explosions, where a high pressure wave is launched towards the ship. During World War II, it was discovered that although such “near miss” explosions do not cause serious hull or superstructure damage, the shock and vibrations associated with the blast nonetheless incapacitate the ship, by knocking out critical components and systems. This discovery led the Navy to implement a rigorous shock hardening test procedure. The shock hardening testing culminates in a Full Ship Shock Trial (FSST), in which an underwater explosive charge is set off near an operational ship, and system and component failures are documented.

JASON was asked by the Navy to examine the potential role of Modeling and Simulation (M&S) for certifying ship hardness, with the potential goal of FSST replacement. We were also asked to examine a number of specific questions about Navy M&S and experimental testing capabilities, and the potential role of other organizations in helping with the Navy mission.

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1 EXECUTIVE SUMMARY

Underwater mines have long been a major threat to ships. The most probable threats are non-contact explosions, where a high pressure wave is launched towards the ship. During World War II, it was discovered that although such "near miss" explosions do not cause serious hull or superstructure damage, the shock and vibrations associated with the blast nonetheless incapacitate the ship, by knocking out critical components and systems. This discovery led the Navy to implement a rigorous shock hardening test procedure. The shock hardening testing culminates in a Full Ship Shock Trial (FSST), in which an underwater explosive charge is set off near an operational ship, and system and component failures are documented.

JASON was asked by the Navy to examine the potential role of Modeling and Simulation (M&S) for certifying ship hardness, with the potential goal of FSST replacement. We were also asked to examine a number of specific questions about Navy M&S and experimental testing capabilities, and the potential role of other organizations in helping with the Navy mission.

1.1 Findings

The key issue is to understand whether risks that are currently managed with combinations of component testing and full ship shock trials can equally well be managed (in whole or in part) with M&S.

The Navy has a high quality team of analysts developing relevant M&S capabilities, and that the M&S capability itself is rapidly improving. However the testing program that is needed to validate M&S does not currently exist. A fully validated M&S program could potentially provide a better route to shock hardening than FSST, though validation is critical before FSST replacement can be considered.

1.1.1 Component testing and full ship shock trials

The FSST tests are done at two thirds of the shock level that ships are required to survive, a level that belongs to the elastic regime. Component testing procedures are intended to match the full design level of a ship by means of an empirical metric. We find that the scientific basis of the component testing procedures is lacking, and that component testing procedures do not necessarily match either the time history of a shock impulse on a ship, or the response a component feels at its location on the ship. Component tests are not done for very large components, and do not address the possibility of failure because of the complex coupling of many components.

The major role of FSST is to mitigate these risks; the significance of the risk level is demonstrated by historical shock trials, which have documented critical equipment failure that was not discovered by component testing alone. The shock trial unfortunately occurs too late to have any influence on the design of a ship.

1.1.2 M&S

Numerical simulations currently play an important role in component testing for ship shock through Dynamic Design and Analysis Method (DDAM), a method used for benchmarking components during the initial design process. DDAM is based on data that is nearly fifty years old. Current M&S capabilities offer significant opportunities for updating DDAM.

Navy M&S capabilities are rapidly improving, but the tools–especially those for predicting the structural response of a ship– need to be better validated. Validation and verification of the structural model would be greatly served by directly monitoring ship vibrations and comparing them to M&S predictions. The direct prediction of fragility boundaries of either components or ships by M&S is currently impossible without heavy reliance on model calibration with tests. Department of Energy capability might be drawn upon if M&S validation indicates that much larger simulations are required to achieve the required accuracy.

There is legitimate skepticism within the Navy over the potential use of M&S for qualification. At least part of this concern is due to the lack of proper validation of the structural model of the ship, and uncertainty about acceptable levels of matching between simulation and measurements. There have been no attempts for M&S to predict the failure modes observed in an FSST, so the performance of M&S remains uncalibrated. It is critical that any metric comparing simulation and measurements be constructed to connect directly to ship survivability and component failure.

When properly validated, M&S can provide a new capability for designing ships and hardening them against shock.

The costs of component testing and FSST are more than 100-fold smaller than the cost of the DDG1000 ship class (including acquisition, development and operation) costs. Hence the main opportunities that exist for cost savings through M&S are for the design of ships and their operation.

A shock trial is not the best method for validating M&S predictions of ship vibrations. Since ship vibrations during FSST are elastic, the best validation is to measure the elastic modes directly. Sensors could probe both modal frequencies and their associated dampings during the normal ship operation, and these could be directly compared with M&S predictions.

A validated M&S could contribute to the efficacy of routine maintenance programs if used in conjunction with a sensor suite that monitors the ship's response to natural vibrations.

A validated M&S capability could be extremely useful in evaluating this, by simulating threat scenarios, and by assessing the vulnerability of existing ships to new threats.

1.2 Recommendations

1. We recommend validation of the Navy M&S predictions for elastic structural response (frequencies and damping).

2. We recommend that the Navy should instrument the lead ship to measure continuously the vibration modes and their associated dampings. Such tests should occur before FSST, in order to provide model validation before FSST predictions .

3. It needs to be determined how well present M&S capability can predict the failure modes of components in Full Ship Shock Trials. This can be done by (i) carrying out comparisons of simulations and observation of failure modes on future shock trials, and (ii) carrying out simulations on recent full ship shock trials. Successful prediction or understanding of the failure modes in the historical database is a substantial step forward in the code validation process.

4. Uncertainties in component testing procedures for testing to a given threat level must be better documented and understood. The Navy's validated M&S capability for liquid response should be used to determine whether the Keel Shock Factor is the right indicator of "similarity" between the shock induced by a hostile event and the impulse delivered in the component test program.

5. An analysis of the potential for the combination of continuous monitoring of ship vibrations with M&S to lead to cost savings in both the design of a ship to a fixed threat level, and the operation and maintenance of a ship should be carefully carried out and documented. This analysis should include the potential for cost savings in operations, and reducing the cost per ship through design improvements while still maintaining design margins for surviving realistic hazards.

6. DDAM should be updated using both experiments and M&S, and incorporating current ship requirements.

7. It is critical for the Navy to maintain the high quality of its analysts in M&S.

2 GENERAL INTRODUCTION

Underwater mines have long been a major threat to ships. The most probable threat does not involve direct contact of a ship with a mine, but has the mine exploding in the vicinity of the ship, launching a high pressure wave into the liquid. During World War II, it was discovered that although such "near miss" explosions do not cause serious hull or superstructure damage, the vibrations associated with the blast nonetheless incapacitated the ship, by knocking out critical components. Since this discovery, the Navy has implemented a rigorous shock hardening procedure of on board components. The shock hardening culminates in a Full Ship Shock Trial (FSST), in which a mine is exploded near an operational ship, and systems failures are documented.

2.1 Questions and Answers

JASON was asked by the Navy to examine the potential role of Modeling and Simulation (M&S) for shock hardening, with the potential goal of FSST replacement. In particular we were asked to address the following questions:

1. Is the Navy's current M&S capability sufficient to support assessment of equipment, distributed ship systems, and ship structures to operationally survive underwater shock from conventional weapons and far field underwater explosions due to nuclear weapons? If not, what new M&S Capability is required?

2. Are there M&S capabilities from other organizations (notably DOE) that Navy should consider adopting or adapting in lieu of creation of new M&S capability?

3. Is the objective of equipment fragility prediction reasonable and, if so, which M&S and testing technologies or approaches should be considered?

4. Are the Navy's current and proposed testing methods sufficient to support assessment of equipment, distributed ship systems, and ship structures to operationally survive underwater explosion from conventional weapons and far field underwater explosions due to nuclear weapons?

5. Is the type, quantity, distribution, and fidelity of the test data currently collected by the Navy sufficient to support validation and verification of current and projected new M&S capability required to support Navy shock assessment? If not, are there other testing approaches, data analysis methods, or sensor technologies that the Navy should be considering?

The key issue underlying these questions is to understand whether risks that are currently managed with either component testing or full ship shock trials can equally well be managed with M&S. Our brief answers to the Navy's questions are given below. The rest of this report is dedicated to developing the answers more fully, explaining how we reached these conclusions.

1. The Navy's M&S capability is moving in the right direction, with high quality analysts developing good M&S tools. However, the tools– especially those for predicting the structural response of a ship– need to be validated. Validation will require the development of new testing procedures that probe vibrational response of ships. Once validated there are credible reasons to believe that M&S could substantially augment and improve testing procedures.

2. Department of Energy capability could be drawn upon to help sort out specific issues that are uncovered during validation of current M&S

capability. For example:

- Validation might uncover that current Navy simulations (with $\approx 10^6$ degrees of freedom) are insufficient to reproduce ship vibrations with the required accuracy, in which case the DOE has expertise, experience, and computing resources for implementing larger calculations effectively.
- Validation might demonstrate that the damping of ship modes is not even approximately described by a linear combination of the mass and stiffness matrices. DOE expertise could help sort out computational methodologies for dealing with more general damping matrices.
- Other specific problems could also arise where DOE expertise could be valuable. However we do not think it is advisable to outsource ship M&S. The M&S of surface ships and large naval equipment include a number of aspects that are discipline specific. We believe it is unlikely that analysts with other expertise can contribute significantly to the M&S program without substantial input from naval architects, scientists and engineers.

3. A validated M&S will predict the local environment that a component experiences during a ship shock. Knowing this environment accurately will allow checking the adequacy of the component testing procedures. Determining whether a component will survive this environment needs to be determined by experimental tests for the foreseeable future. The prediction of fragility boundaries of either components or ships by M&S is currently impossible without heavy reliance on model calibration with (expensive) tests.

4. FSST is an important component of current testing procedures, mitigating a number of risks that are not otherwise addressed in the testing procedures. These risks are real, as demonstrated by documented Grade A system failures during an FSST.

If properly validated, the risks mitigated by FSST could well be taken over by a combination of M&S and alternative testing procedures; indeed, there are excellent arguments suggesting that a validated M&S capability could bring entirely new capabilities, including better threat mitigation, and cost reduction/integration during the ship design process.

A successful M&S program requires new tests to measure ship vibrations (natural frequencies and dampings). Alternative testing methods such as air guns might also play a useful role: although the pressure impulse delivered by an air gun is much smaller than FSST requirements, air guns could be a useful tool to probe and test the elastic response of a ship.

5. Current Navy testing methods, as well as the FSST alternative, are insufficient for validation of the structural model. Validation and verification of the structural model requires directly monitoring ship vibrations, and comparing the measurements with M&S predictions.

In studying this topic, we were very fortunate to have briefings from talented scientists and engineers from both the Navy and the Department of Energy. In particular we are grateful to: Tom Julian (OSD); Mike Winnette (SEA 05P); Fred Costanzo (Carderock); Angela Maggioncalda (PMS 500); Charles Hutching (N091); Gregg Harris (Indian Head); Chris Abate (Electric Boat); Tom Moyer (Carderock); Hal Morgan (Sandia); Tony Giunta (Sandia); Bob Garrett (OSD); Dennis Baum (OSD); Joseph Jung (Sandia); Bob Heyburn (NSWC); Kurt Hartsough (Philadelphia 623) Dave Ingler (Carderrock). We would like to thank them all for their contributions to our understanding of this subject. Special thanks are due to our point of contact, Dick Vogelsong, for his outstanding organization and extensive help in understanding all matters discussed herein.

2.2 Legislation

The Navy is required by law to subject major systems and munitions programs to survivability and lethality testing before full scale production. The legislation is set out in title 10, U.S. Code Section 2366. Specifically, the legislation states that *A covered system may not proceed beyond low rate initial production until realistic survivability testing of the system is completed.*

This law is most commonly applied to munitions systems that are produced in large numbers, such as tanks. In that case the system is subject to a combat environment (ultimately destroying the combat system), and survivability is assessed. For ships this procedure is infeasible. The lead ship in a class is itself operational, and also very expensive: for example the lead ship in the DDG1000 is projected to cost upwards of \$3.5B. Moreover, the acquisition of a new ship class takes of order 20 years. By the time the lead ship is delivered, typically five contracts have been issued for subsequent ships in the class.

By law the secretary of defense is authorized to waive equipment from the requirements of the legislation, but he is not allowed to waive equipment from testing. For ships, this waiver has traditionally consisted of substituting a full shock test with a mixture of component testing at the full requirement, with a Full Ship Shock Trial (FSST) which occurs at 2/3 of the requirement. The FSST is the only test of the ship against an actual weapon. While the FSST involves an explosion set off some distance from a ship, component tests occur in different ways (ranging from hammer tests, to barge tests, in which an explosion is placed at some distance from the barge). The component tests are described in detail below. A critical question is whether the component tests are testing to the same standard as the FSST.

2.3 The Design Requirement –The Shock Factor

The design requirement for ship shock is quantified through a *shock factor*. If W is the weight of the explosive and D is the distance of the explosion from the ship, the shock factor is of the form

$$\frac{W^n}{D}. \tag{2-1}$$

The values of n and the precise value of the shock factor are classified, and are discussed in a classified appendix to this report.

The shock factor is used to set the strength of an explosion for a FSST, which is required to be conducted at 2/3 of the design level. At this design level, we were told during our briefings that the structural response of the ship is described by linear elasticity. The shock factor is also used for designing component tests: for example the floating shock platform (discussed below) is also set off with an explosion at some distance from the barge. The weight of the explosive and the distance from the barge are set with the shock factor at the full design level. For hammer tests in which an impact is given to a component, the test is designed so that the maximum kick off velocity after impact is a pre-specified velocity.

JASON analysis discussed in a classified appendix to this report finds that the shock factor is not physics based, and discusses the consequences of this conclusion.

3 EQUIPMENT FAILURE CRITERIA: SUMMARY AND HISTORICAL PERSPECTIVE

The crux of predicting ship shock survivability is to determine whether a piece of equipment will fail when presented with the impulsive loading of a ship shock.

How can one characterize the failure of equipment? In general terms, components fail when parts inside of them break. Breaking can be caused by a strain exceeding the yield strain for some part of the component. In general whether this will occur depends on the displacement, velocity and acceleration history of the component. Currently, the only practical way to map out this failure surface is to use experiments; there are simply too many ways that something can break for M&S to predict the breaking mode *a priori*. M&S can be used for predicting failure only if the failure mechanism for a particular forcing is determined, and then the simulations are calibrated with experimental tests.

Despite this pessimistic assessment of the role of M&S for predicting component failures, much can be understood about failure and failure criteria by examining the response of a component in the linear regime, where stresses are directly proportional to strains, and where a validated M&S capability is extremely well suited for making predictions. This is because analysis in the linear regime is accurate for low intensity forcing, and as the intensity of the forcing increases the linear regime still applies up to some critical strain. Failure occurs well into the nonlinear regime, where stresses are not linearly proportional to strains, and irreversible changes occur in the components due to the forcing.

3.1 The Historical Basis: Earthquake Hardening of Buildings

Indeed, the historical basis for the Navy's component testing procedures are based on analysis in the linear regime. The historical foundations for these ideas are based on Biot's work on earth quake hardening buildings. Biot began this work in response to a question from his PhD thesis advisor von Karman about why some buildings fall during earthquakes and others do not. Biot's goal was to both answer this question and to provide an experimental methodology to test for whether buildings are earthquake proof. Biot considered a building to be a superposition of linear modes, each of which obeys the equation of a harmonic oscillator. He imagined that each mode is forced by a base that is subjected to a prescribed acceleration $a(t)$. The oscillator equation is

$$\ddot{x} + \omega_0^2 x = a(t), \tag{3-2}$$

where $\omega_0 = \sqrt{k/m}$ is the natural frequency of the oscillator of mass m and spring constant k. Biot observed that the most dangerous modes are those that are near resonant. He then recommended recording the acceleration spectrum of earthquakes–the Fourier Transform of $a(t)$, representing the acceleration amplitude as a function of frequency. Buildings should be hardened so that the modes of buildings could withstand the accelerations they experience, with the main issue being the avoidance of resonance.

3.2 Analysis of Linear Modes for Ship Shock

Although the ship shock problem shares important features with the earthquake hardening of buildings, the two situations are not exactly equivalent, and it is therefore worth analyzing how the modes of components are forced by ship shock. A typical $a(t)$ for ship shock has a maximum acceleration a_{max} which lasts for a time $T_{shock} \sim 10^{-3}$sec. Let us first consider the vibrational response to a component which feels exactly the $a(t)$ produced by the pressure pulse of the shock, without any filtering of this response by the ship.

We would like to predict the maximum acceleration of the component in response to shock. If we assume that (a mode of) the component obeys Eq. 3-2, we need to solve for $x(t)$, and then compute the maximum acceleration $A_{max} = \max_t(\ddot{a}(t))$. We of course do not know the precise form of $a(t)$, but let us assume that $a(t)$ is given by a particularly simple form: $a(t) = a_{max}$ for $t \leq T_{shock}$ and $a(t) = 0$ otherwise.

For a square pulse $a(t)$, Eq. 3-2 can the be solved directly using (for example) Laplace transforms. We assume that $x(0) = \dot{x}(0) = 0$. The Laplace transform of Eq. 3-2 is

$$X(s) = \frac{1}{\omega_0^2}\Big(\frac{1 - e^{-sT_{shock}}}{s(s^2 + \omega_0^2)}\Big).$$

Inverting the transform, the solution is

$$\ddot{x}(t) = a_{max} cos(\omega_0 t), \tag{3-3}$$

when $t \leq T_{shock}$, and

$$\ddot{x}(t) = a_{max}(cos(\omega_0 t) - cos(\omega_0(t - T_{shock}))), \tag{3-4}$$

for $t \geq T_{shock}$. We are interested in the *maximum* acceleration $A_{max} = \max_t \ddot{x}(t)$.The initial acceleration during the shock is given by a_{max}. After the shock, there are two relevant limits that need to be considered.

- $\underline{\omega_0 T_{shock} \ll 1}$ In this limit[1], Eq. 3-4 becomes

$$\ddot{x}(t) \approx a_{max}\omega_0 T_{shock} sin(\omega_0 t). \tag{3-5}$$

[1]This result is derived by noting that $cos(\omega_0(t - T_{pulse}) \approx cos(\omega_0 t) + \omega_0 T_{shock} sin(\omega_0 t)$ when $\omega_0 T_{pulse} \ll 1$

Thus the maximum acceleration occurs immediately with the magnitude

$$A_{max} = a_{max}\omega_0 T_{shock} = V_{kickoff}\omega_0. \tag{3-6}$$

In this limit the maximum acceleration is determined by the product of the kick off velocity

$$V_{kickoff} = \int_0^{\infty} a(s)ds$$

and the resonant frequency of the oscillator. In general this acceleration is much less than the initial acceleration during the shock a_{max}. But if $\omega_0 T_{shock} \ll 1$ then the oscillator does not move very much while it is being accelerated and so one might hypothesize that the damage potential is more limited. T

- $\underline{\omega_0 T_{shock} \gg 1}$ In this limit the two terms in Eq. 3-4 do not nearly cancel, and indeed they constructively interfere to produce A_{max}. The maximum acceleration occurs when the two terms $cos(\omega_0 t)$ and $cos(\omega_0(t - T_{shock}))$ are out of phase with each other and hence

$$A_{max} = 2a_{max}. \tag{3-7}$$

  Hence in this limit the maximum acceleration does *not* depend on the resonant frequency of the spring but instead is twice the maximum acceleration of the input pulse!

Note that since $T_{shock} \sim 10^{-3}$ sec for shock loading, the resonant frequency where one crosses from the first regime to the second is at $2\pi 10^3$Hz, or $\approx$ 6 kHz. Thus we have demonstrated that the maximum acceleration is the product of the kick off velocity and the resonant frequency for frequencies below 6 kHz, and the maximum acceleration is constant (set by the acceleration of the shock itself) above this resonant frequency.

This analysis contains the essence of the shock response spectrum for components on ships, but we must understand how to interpret it. The resonant frequency ω_0 can correspond either to

1. The resonant frequency of the critical part of a component, assuming the acceleration imparted to the component is as assumed $a(t) = a_{max}$ for $t < T_{max}$. This requires that the component is hard mounted to a part of the ship that is itself rigid and close to the impact point.

2. If the component is not hard mounted to a rigid object, but is hard mounted to a ship structure (e.g., the deck) which itself vibrates at frequency ω_0, then the maximum acceleration corresponds to that of the ship structure. Indeed in this case the forcing function assumed for the component is more accurately captured by the response characteristics of the ship.

3. If the component is not rigidly mounted, it is also necessary to take into account the frequency of the mount.

3.3 Failure Modes

With these results in hand we can now observe that there are essentially two different ways a component can fail on a ship in the linear regime.

1. The first possibility is that the component experiences an acceleration beyond its design limit. Hence (assuming that the ω_0 in question is below 6kHz) the maximum acceleration that a component experiences is either $V_{kickoff}\omega_{component}$ or $V_{kickoff}\omega_{shipstructure}$ depending on where the structure is mounted. Hence whether or not the acceleration exceeds the design limit depends critically on not only $V_{kickoff}$ but also either $\omega_{component}$ or ω_{ship}.

2. Another possibility arises for components that are themselves mounted on a flexible structure (e.g., a deck). If this structure oscillates at frequency ω_0, there is the possibility of a resonant interaction of this

forcing frequency with a mode of the component. To estimate how this affects the maximum component acceleration let us assume that the flexible structure has a damping rate γ. Then the maximum acceleration of the component is $A_{structure}\omega_0/(2\gamma)$, where $A_{structure} = V_{kickoff}\omega_0$. If we assume that $\gamma = \omega_0/N$, so the vibrations of the ship structure damp out in N oscillations we have that the maximum component acceleration is $NV_{kickoff}\omega_0/2$. Thus if the structural mode were not strongly damped there is the potential for the component acceleration to greatly exceed that of the structure itself.

4 COMPONENT TESTING PROCEDURES

The consequences of the above analysis is that for components with critical resonant frequencies below 6 kHz, the critical parameters from the ship shock are the kick off velocity $V_{kickoff}$ and the characteristic frequency, either $\omega_{component}$ or $\omega_{ship\ structure}$. The maximum acceleration that a component experiences depends on its own critical resonant frequencies, as well as how and where it is mounted. We now discuss the component testing procedures that are used by the Navy, and analyze their efficacy in light of these conclusions.

NAVSEA Instruction 9072.1A (Shock Hardening of Surface Ships) dictates that every shipboard component be pre-qualified for shock hardening. The acceptable methods of qualification are: 1) test 2) analysis (DDAM) and 3) extension. Test is the primary and most desirable qualification method. Analysis is permitted for qualification only when component weight and/or size precludes a test. Extension is essentially a waiver from the qualification process, granted when a component is a close derivative of a component that was previously shock qualified in test. In granting an extension consideration is not explicitly given to where the component is located in the ship, and whether there are changes in this environment from ship to ship (or ship class to ship class).

The procedures and requirements for shock testing of shipboard components are detailed in MIL-S-901D, "Shock Tests, H.I. (High-Impact) Shipboard Machinery, Equipment, and Systems, Requirements for." While MIL-S-901D was released in March 1989, it remains largely unchanged from the previous revision (901C) issued in 1963. The standard is based primarily on WWII-era analysis of underwater explosions and their effects upon contemporary ships.

4.1 Test Program Overview

The component test techniques are intended to simulate – within a laboratory environment – the effect of shock on a particular piece of shipboard equipment. Specifically, the tests are expected to inflict equivalent damage to a component, as if the component were mounted aboard the ship and the ship were exposed to an underwater explosion of given shock factor. In this case, the relevant shock factor is the full design shock factor. However as we have seen in the previous section as long as the resonant frequency of the equipment modes in question are below 6 kHz, the relevant quantity is product of the the kick off velocity and a characteristic frequency–and indeed, we will find that tests are designed to match $V_{kickoff}$.

MIL-S-901D describes the three test apparatus which are used for component shock qualification: the Lightweight Shock Machine (LWSM), the Medium Weight Shock Machine (MWSM), and the Floating Shock Platform (FSP) and its variants. These are described in detail in the following section. Choice of test apparatus depends primarily upon component size and weight, but may also be dictated by the component's classification. The components are primarily classified by their anticipated mounting technique ("Equipment Class"), their mounting location, and their necessity for the ship's function ("System Grade"):

Equipment Class. *Class I* components are those which are "hard-mounted" aboard the ship, i.e. bolted or welded directly to the ship's structure. *Class II* components contain resilient mounts between the component and the ship structure, usually specifically intended for shock absorption. If the travel allowed by the resilient mounts is greater than 3", the Class II component must be tested on the FSP. The importance of these distinctions is that, as we have seen, the resilience of the mount directly affects the maximum acceleration of the component.

Mounting Location. *Hull-Mounted* components are those which are intended to be mounted on main structural members, including structural bulkheads, structural stiffeners, and shell plating above the waterline. *Deck-Mounted* components are those which are intended to be mounted on decks, platforms, and non-structural bulkheads below the main deck, and anywhere above the main deck. *Shell-Mounted* components are those which are intended to be mounted on shell plating below the water line. *Wetted-Surface Mounted* components are those which are intended to be mounted external to the hull, below the waterline. All Shell-Mounted and Wetted-Surface Mounted equipment must be tested on the FSP; it is *recommended* that all Deck-Mounted equipment also be tested on the FSP, regardless of size and weight.

System Grade. *Grade A* components are those which are essential for the safety of the ship's crew and/or the ship's continued combat capability. *Grade B* components are those which are not essential for crew safety or combat capability, but which may become hazardous to either the ship's crew or nearby Grade A components if damaged. *Grade C* components are those which are neither essential nor pose a hazard if damaged by shock. Grade A equipment must be tested while operational (e.g., energized, pressurized, motors running, etc.). Grade B equipment need not be running while tested, unless the operational state increases the risk of hazard. Grade C equipment does not need to be qualified by shock testing.

Component shock tests are *not* designed to probe systems-level failure, as only one component is tested at a time. In some cases, a single component system may not be tested wholly. Shipboard components are required to be tested at the "principal unit" level, but allowances are often made for subsidiary component or subassembly testing when the principal unit is too large or heavy even for the largest FSP. Furthermore, if a principal unit fails a shock test due to localized failure in a subsidiary component, a successful shock test of the hardened or redesigned subsidiary component alone suffices for shock qualification of the entire component principal unit. For these reasons, shock tests of equipment subcomponents are not uncommon. When components or subsidiary components that are a part of a larger system are tested, it is required to simulate all relevant shipboard connections (e.g., incoming pipes, drive shafts, etc.). Following the guidance of MIL-S-901D, it is the contractor's responsibility to determine which connections are relevant, and how best to simulate them during the test.

The acceptance criteria for shock qualification by test are clear: Grade A components must continue to function during and after the test (with allowances for momentary malfunction), and Grade B components simply must not create a hazard (e.g., leaking toxic fluid, fire, etc.). Grade A components must undergo both functional testing and inspection after the test, whereas Grade B components need only be inspected for hazards. In general, no instrumentation is required to test against these acceptance criteria.

4.2 Experimental Apparatus

Substantial insight into the operating characteristics and historical development of experimental apparatus for component shock qualification is provided in NRL Report 7396, "Shipboard Shock and Navy Devices for its Simulation" (Clements, 1972). The following descriptions rely upon information provided in this report, as well as the official test procedures described in MIL-S-901D.

4.2.1 Lightweight Shock Machine (LWSM)

The Lightweight Shock Machine (LWSM) was the first test apparatus developed for component shock qualification, dating back to 1940. The

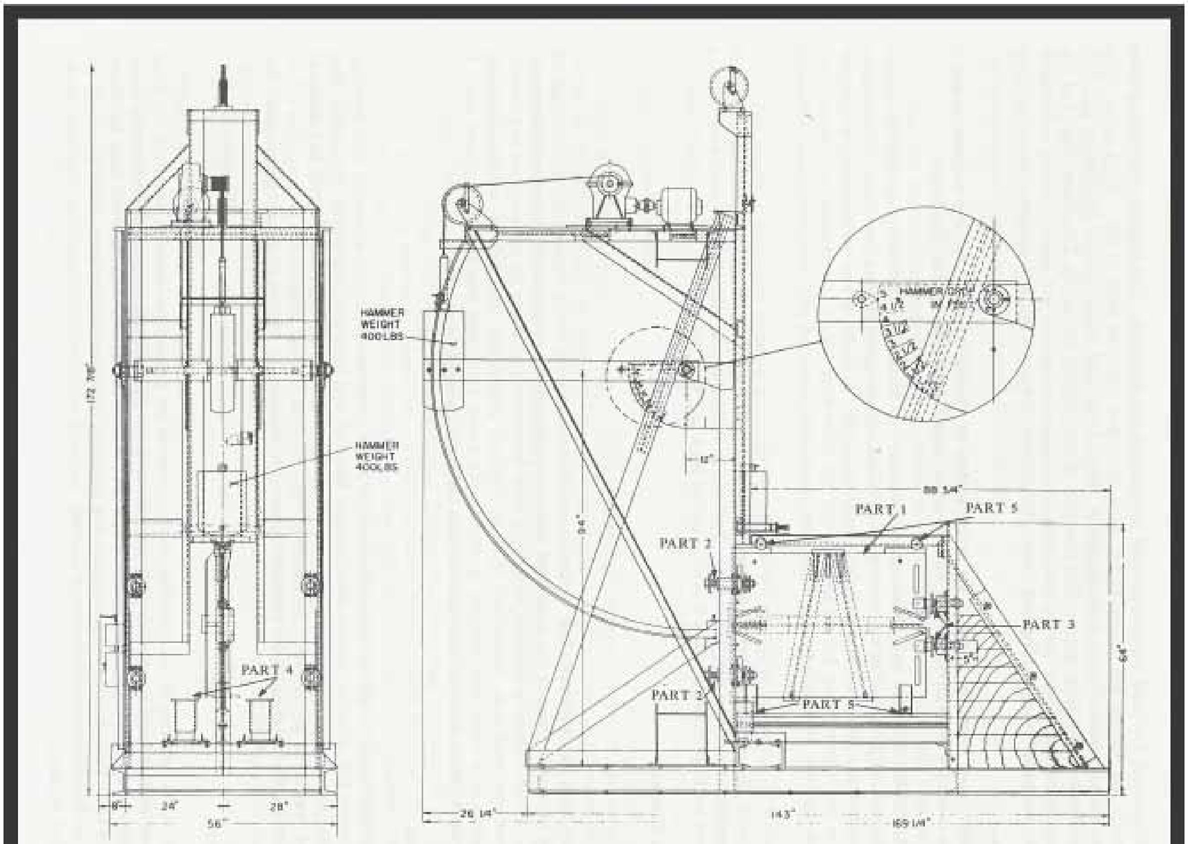

Fig. 1 — The Lightweight Shock Machine (LWSM). Part 1 is the anvil-plate, Parts 2 (4 each) are the forward springs for Back blows, Parts 3 (2 each) the forward springs for Edge blows, Parts 4 (2 each) the forward springs for Top blows, and Parts 5 (6 each) the anvil-plate guide rollers for Edge blows. The LWSM is drawn with the anvil-plate oriented for Edge blows.

Figure 1: The Lightweight Shock Machine (courtesy K. Hurtsough).

LWSM is used primarily for testing Hull- and Deck-Mounted components weighing up to 250 pounds. A diagram of the LWSM is given as Figure 1. The LWSM imparts a shock load upon a component by hammer impact. The component of interest is mounted to the anvil table (part 1 in Figure 1), and the 400lb pendulum-hammer is raised to a specified height. When the hammer is released, its potential energy is converted to kinetic energy. This kinetic energy is conveyed to the anvil table by elastic collision, generating a shock pulse through the anvil table and into the component. Following the guidelines set forth in MIL-S-901D, the component receives a number of "blows" of increasing intensity and in different mounting orientations. Intensity of the shock is controlled by the height of the hammer.

Clements (1972) reports on experimental analysis of the acceleration pulse generated by the LWSM system, at the anvil table. He describes this pulse as a half-sinusoid with 2ms duration and 32-840g amplitude, depending on the hammer height, load orientation, and accelerometer location. The shock felt at the component depends not only on these pulse characteristics, but also on the impedance characteristics of the mounting foundation (anvil table + fixtures). The same anvil table, essentially a heavy metal mass, is used for all components. The mounting fixture is specified by the type and orientation of mounting that the component will see on the real ship; there are six standard fixtures to choose from. All of the these mounts are generally rigid, though, allowing much of the high-frequency energy of the impact to transmit to the component. In this regard, the foundation impedance of the LWSM most closely simulates hull and bulkhead mounting.

4.2.2 Medium Weight Shock Machine (MWSM)

The Medium Weight Shock Machine (MWSM) was first developed in 1942, based on the Navy's satisfaction with the LWSM and the need to shock qualify even heavier items. The MWSM can accommodate components ranging in weight from 250lbs to 6000lbs. A schematic of the MWSM is given as Figure 2. Like the LWSM, the MWSM utilizes hammer impact to convey shock loading upon the component of interest. In this case, the component is fixed atop a massive (4400lb) anvil table, which is struck by a 3000lb hammer after it has swung through 270° of rotation. The MWSM is generally permanently installed at a test facility, such that the hammer swings through a hole in the floor and strikes the anvil table from below (see Figure 2).

Prior to testing on the MWSM, a foundation for the component is installed on the anvil table. Following guidelines prescribed in MIL-S-901D,

TABLE II. Test schedule for medium weight shock machine

Group number	I	II	III
Number of blows	Note (1)	Note (1)	Note (1)
Anvil table travel, inches	3	3	1.5
Total weight on anvil plate (pounds) (Note (2))	Height of hammer drop (feet) (Notes (3) and (4))		
Under 1000	0.75	1.75	1.75
1000 - 2000	1.0	2.0	2.0
2000 - 3000	1.25	2.25	2.25
3000 - 3500	1.5	2.5	2.5
3500 - 4000	1.75	2.75	2.75
4000 - 4200	2.0	3.0	3.0
4200 - 4400	2.0	3.25	3.25
4400 - 4600	2.0	3.5	3.5
4600 - 4800	2.25	3.75	3.75
4800 - 5000	2.25	4.0	4.0
5000 - 5200	2.5	4.5	4.5
5200 - 5400	2.5	5.0	5.0
5400 - 5600	2.5	5.5	5.5
5600 - 6200	2.75	5.5	5.5
6200 - 6800	3.0	5.5	5.5
6800 - 7400	3.25	5.5	5.5

Note (1) Items requiring testing in 2 orientations get 2 blows per group; items requiring testing in 3 orientations get 3 blows per group.

Note (2) Total weight on anvil plate is the sum of equipment weight plus weight of all mounting fixtures. Weight limits are as defined in section 3.1.2.(b).

Note (3) The height of hammer drop shall be measured by means of the existing markings on the scale of the machine, no corrections being made for the added anvil table travel for the blows of groups I and II.

Note (4) For submarine frame mounted equipment, hammer drop heights for groups I, II, and III shall be 3, 5.5, and 5.5 feet, respectively.

Figure 2: The Medium Weight Shock Machine (courtesy K. Hurtsough).

this foundation consists of a number of "car-building" channels and rails for support, as well as one of five standard mounting fixtures. Two of the fixtures position the component at a 30° angle with respect to horizontal, with the intent of testing athwartship shock hardness. The anvil table itself is mounted atop the striking surface with a series of massive bolts. These bolts are configured so as to allow as much as 3" of vertical travel of the anvil table. Upon being struck by the hammer, the anvil table-foundation-component system accelerates vertically until it hits the bolt stops, then accelerates downward due to both elastic rebound and gravity. The initial acceleration impulse depends upon the height of the hammer before release. As for the LWSM, a progression of blows are prescribed for the MWSM in MIL-S-901D. In this case, the prescribed hammer heights are given as function of component weight – see Figure 3.

TABLE II. Test schedule for medium weight shock machine

Group number	I	II	III
Number of blows	Note (1)	Note (1)	Note (1)
Anvil table travel, inches	3	3	1.5
Total weight on anvil plate (pounds) (Note (2))	Height of hammer drop (feet) (Notes (3) and (4))		
Under 1000	0.75	1.75	1.75
1000 - 2000	1.0	2.0	2.0
2000 - 3000	1.25	2.25	2.25
3000 - 3500	1.5	2.5	2.5
3500 - 4000	1.75	2.75	2.75
4000 - 4200	2.0	3.0	3.0
4200 - 4400	2.0	3.25	3.25
4400 - 4600	2.0	3.5	3.5
4600 - 4800	2.25	3.75	3.75
4800 - 5000	2.25	4.0	4.0
5000 - 5200	2.5	4.5	4.5
5200 - 5400	2.5	5.0	5.0
5400 - 5600	2.5	5.5	5.5
5600 - 6200	2.75	5.5	5.5
6200 - 6800	3.0	5.5	5.5
6800 - 7400	3.25	5.5	5.5

Note (1) Items requiring testing in 2 orientations get 2 blows per group; items requiring testing in 3 orientations get 3 blows per group.
Note (2) Total weight on anvil plate is the sum of equipment weight plus weight of all mounting fixtures. Weight limits are as defined in section 3.1.2.(b).
Note (3) The height of hammer drop shall be measured by means of the existing markings on the scale of the machine, no corrections being made for the added anvil table travel for the blows of groups I and II.
Note (4) For submarine frame mounted equipment, hammer drop heights for groups I, II, and III shall be 3, 5.5, and 5.5 feet, respectively.

Figure 3: Test schedule for the MWSM, as given in MIL-S-901D. Groups I and II correspond to 3" of allowable anvil table travel; Group III corresponds to 1.5" of table travel.

NRL Report 7396 (1972) suggests that the MWSM was designed to reproduce the impulse intensity of the LWSM – thought to invoke equivalent damage as the full design shock factor – by matching the kick-off velocity of the anvil table in the two instruments, for a component of the same weight. JASON analysis indicates that in fact the schedule of hammer heights prescribed in MIL-S-901D (Figure 3) produces approximately identical kick-off velocity of the anvil table, across the spectrum of component weights. This analysis is summarized in Table 1. It may be inferred, then, that for the

Table 1: JASON calculation of MWSM anvil table kick-off velocities, corresponding to the component weights and hammer heights prescribed in Figure 3.

2*Total Weight on Anvil Plate [lbs]	Group I	Group II	Group III
	Approx. Anvil Table Kick-off Velocity [ft/s]		
Under 1000	4.25	6.50	6.50
1000-2000	4.08	5.77	5.77
2000-3000	3.90	5.23	5.23
3000-3500	3.85	4.97	4.97
3500-4000	3.91	4.90	4.90
4000-4200	4.00	4.90	4.90
4200-4400	3.91	4.99	4.99
4400-4600	3.82	5.06	5.06
4600-4800	3.97	5.12	5.12
4800-5000	3.88	5.17	5.17
5000-5200	4.00	5.37	5.37
5200-5400	3.92	5.55	5.55
5400-5600	3.84	5.70	5.70
5600-6200	3.87	5.48	5.48
6200-6800	3.82	5.18	5.18
6800-7400	3.77	4.91	4.91

purpose of designing these laboratory instruments, the design shock factor was interpreted as an initial kick-off velocity of the component's foundation. While these kick-off velocities may in fact closely match the kick-off velocities of components under full ship shock, the associated impulse may not match in displacement (initial or accumulated) or peak acceleration.

Analysis of the impulse generated by the MWSM indicates loading similar to the LWSM. Clements (1972) reports that the initial pulse, which is always the most dominant, is a half-sinusoid with 1ms duration and 220-580g amplitude, depending on the hammer height. Loading on the MWSM is complicated by the presence of the bolt stops at 3"; these induce a second shock load on the component as its foundation acceleration reverses in direction. Finally, there is a third shock load as the component comes to rest. While these ensuing shocks are always decreasing in magnitude with respect to the initial shock, they occur much closer in time than the subsequent loadings in a real shock test (e.g., cavitation closure, bubble pulse).

In order to fully understand the loading on the component, the foundation impedance must also be considered. In the MWSM, foundation impedance is determined primarily from the car-building/rail channels. Clements, in 1972, writes "the evolution and intent of this mounting system have since been lost and remain today a subject of speculation." He suggests that the number and configuration of these channels prescribed in MIL-S-901D is such to keep the channels themselves from yielding under a certain acceleration load. The unintended consequence is that the anvil table-foundation-component system will always have a natural frequency between 55 to 72 Hz. For some components, this will not be a realistic foundation. The deviation was likely thought to only provide an extra margin of conservatism.

The validity of the MWSM fixturing has been questioned in a number of theses written at the Naval Postgraduate School (see, for instance, Corbell, 1992, Cox, 1993, or Flynn, 1994). These theses suggest that the unrealistic test environment of the MWSM will – in some cases – introduce vulnerabilities rather than an extra margin of conservatism. They note that, for one, the shock loading imparted to the component is short in duration, high-impulse, and contains high frequency components. This is distinct from the shock loading observed on many deck-mounted equipment well above the waterline. Furthermore, the MWSM standard fixturing only allows for specific frequencies of excitation, which may or may not be the most critical for a given component. Each component system has a certain frequency or frequencies of excitation which will elicit the greatest response. It is excitation at these frequencies which is most likely to damage the shipboard component. The NPS theses cited above suggest "tuning" the MWSM fixturing, such that the component is excited at these critical frequencies. If the components survive this tuned shock excitation, they would then be assured of surviving the envelope of shock spectra associated with a full design shock. While this method of tuning the fixture may in fact prevent unexpected failures during the FSST, it is conservative in nature and may actually lead to increased costs associated with unneccessary shock hardening. This method does not leverage off of current modeling and simulation techniques, which may be able to predict exactly how a given component (with known shipboard location) will be excited during an underwater explosion event.

4.2.3 Floating Shock Platform (FSP)

The Floating Shock Platform (FSP) and its variants are used to conduct heavy-weight ($>$ 6000lb) component testing. In addition to the FSP (28'x16', 60,000lb capacity), this family of test beds includes the Intermediate Floating Shock Platform (IFSP; 40'x20', 250,000lb capacity), the Extended Floating Shock Platform (EFSP; 46'x16', 100,000lb capacity), and the Large Floating Shock Platform (LFSP; 50'x30', 400,000lb capacity).

All of these platforms are essentially floating barges which are intended to test individual components via a scaled-down version of the FSST. The FSP test is distinct from the LWSM and MWSM tests in significant ways: 1) the test involves an actual underwater explosion rather than a simulated impulse 2) there are no standard mounting fixtures for the FSP 3) there is simultaneous loading in the vertical and athwartship directions and 4) the test beds are owned and operated by the government (all LWSMs and MWSMs are owned and operated by contractors, with government certification).

The loading in the FSP test, like in the FSST, is created with an underwater charge of HBX explosive at specified depth and distance from the barge. The exact test specifications are given in MIL-S-901D, and repeated here in Figure 4. The FSP specifications apply to the FSP, the IFSP, and EFSP, even though, when loaded, the masses of these different test platforms vary by a factor of 50. The charge weight and distances given for Shot 4 correspond to the full design shock factor, which should be 50% greater in intensity than that felt by the ship in the FSST.

TABLE III. Test schedule for heavyweight shock testing

Test conditions	Standard floating shock platform	Large floating shock platform
Depth of explosive charge below water surface (for all shots)	24 feet	20 feet
Explosive Charge weight/composition	60 lbs/HBX-1	300 lbs/HBX-1
Shot direction[1]:		
Shot 1	Fore-and-aft	Fore-and-aft
Shots 2,3, and 4	Athwartship	Athwartship
Standoff[2]:		
Shot 1	40 feet	110 feet
Shot 2	30 feet	80 feet
Shot 3	25 feet	65 feet
Shot 4	20 feet	50 feet

[1] For the fore-and-aft direction shot, the explosive charge shall be located relative to the floating shock platform so as to represent an underwater explosion occurring off the bow or stern of the ship in which the equipment is to be installed (see 6.2.1). Athwartship shots shall be oriented to represent explosions abeam of the ship.

[2] Refers to the horizontal distance between the explosive charge centerline and the near side of the floating shock platform.

Figure 4: Test schedule for the Floating Shock Platform and its variants, as given in MIL-S-901D.

While the heavy weight shock test program most closely resembles the FSST, it still is distinct in significant ways. For one, the test charge weight and depth are specified such that the explosive bubble vents on its second expansion. Thus, there is no bubble pulse loading. Furthermore, the charge is scaled down by a factor of 100 from the 2,000lb and 10,000lb charges typically used in the FSST. Combined with the large variation in FSP weights, it is unclear whether this charge is actually generating an impulse which is greater in intensity than that observed in the FSST. Finally, the test platforms do not have the complex hull and deck structure of a real ship, which tends to attenuate the pressure pulse before it reaches a deck-mounted component; the deck-mounted environment is simulated in the heavy weight testing by means of the component fixturing.

While there are no standard component fixtures for the heavy weight shock testing, MIL-S-901D requires that the custom fixturing meet specific impedance characteristics. For Class I components (hard-mounted), the fixturing is to be designed such that the system of the component equipment and fixture together has natural frequencies of 25Hz or greater in each principal direction. For Class II components (resiliently-mounted), it is to be designed such that the component/fixture assembly has a natural frequency of 12 to 16 Hz in the vertical direction only.

4.3 DDAM

An M&S procedure is already a critical part of the component testing procedure. This is DDAM, Dynamic Design and Analysis Method. DDAM is primarily used as a design tool, to guide contractors in designing components that should pass the shock test, and is also used in itself to qualify components that are too large to test.DDAM is well integrated with commercial vendors, with many (e.g., Abaqus–see www.abaqus.com/ddam) having integrated the DDAM methodology into their software procedure.

The essential idea behind DDAM comes from Biot's invention of the acceleration response spectrum for earthquake hardening buildings. In that context Biot recommended that every normal mode of a building should be able to withstand the largest earthquake that has ever been observed to force that frequency.

In turn, an acceleration response spectrum is employed for ship board components to shock; the acceleration spectrum follows the general principles outlined above. For lower frequency components the critical acceleration is the product of a kick off velocity and the modal frequency. Above a critical frequency the components must be able to withstand a critical acceleration. Figure 5 sketches the typical acceleration spectrum that DDAM assumes.

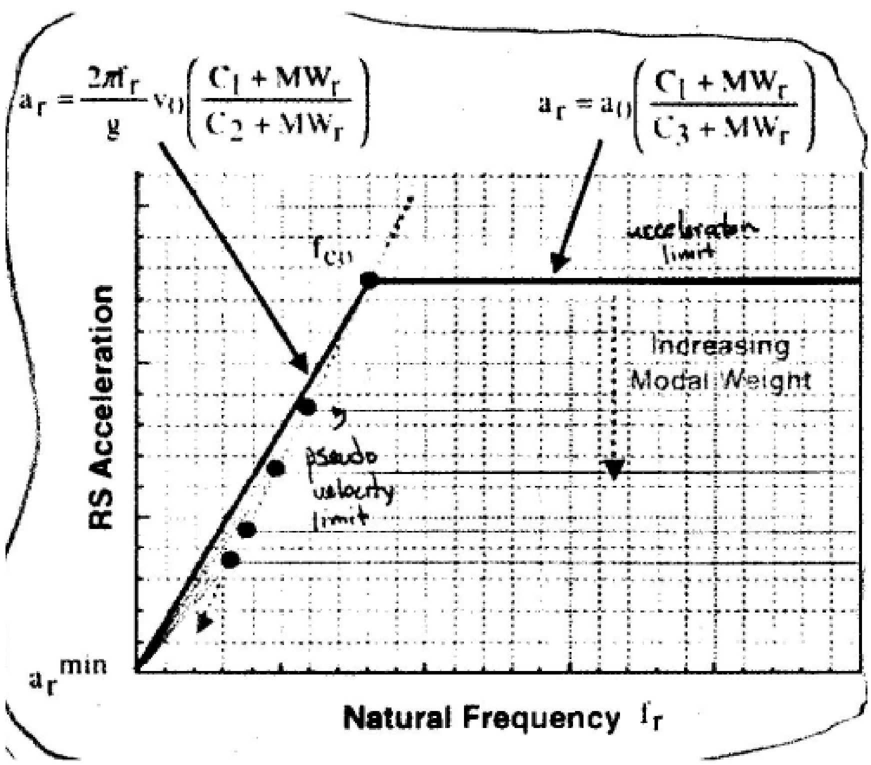

Figure 5: Acceleration spectrum used in DDAM (courtesy of B. Heyburn).

As outlined above, this shape of acceleration response curve is well founded assuming a single mode response to an impulse. On the other hand, DDAM makes two critical assumptions that are not generally true for component response on ships:

1. First DDAM completely neglects the possibility of resonant interactions of components with the motion of the ship itself (induced by the blast). The shock response in Figure 5 is exactly that predicted for a square

pulse impact, and does not apply when the component is forced by ship modes. This could lead to important discrepancies, in particular the possibility of a resonant interaction of the component with the ship.

2. DDAM assumes that a single mode interacts with a short timescale response. This is false for large components that are large enough to experience the excitation of multiple ship modes simultaneously.

4.4 Analysis of Component Testing Procedures

The component testing procedures are rather consistently designed to match the kick off velocity of the design requirement. In accordance with our analysis of the response of components to shock, this is a very reasonable design criterion. Nonetheless, there are a variety of issues and simplifying assumptions with the component testing procedures. In particular:

1. The component testing procedures are designed for components that are rigidly mounted with resonant frequency below 6kHz.

2. If a resonant frequency of a component were above 6kHz, the matching of kick off velocity is inappropriate. In that case it is necessary to match the acceleration of the test to that of the shock impulse. Available evidence suggests that the accelerations for the component tests do not match (and were not designed to match) the ship shock response.

3. If a component had resonant frequency below 6kHz but were mounted to a flexible structure (with frequency below 6kHz), the initial acceleration the component would feel would be different than assumed in the component tests, unless the component is tested on a mount with a resonant frequency equal to that of its ship environment. A component which is hardened against 14Hz excitation may pass component

qualification but fail catastrophically under real ship shock because the critical foundation excitation is actually at 10Hz.

To guard against resonant failure (not during the initial acceleration), it is also necessary for the component to be tested on a mount that matches the damping. To our knowledge there is no consideration of this in the testing procedures.

4. The testing procedure makes a critical assumption about the acceleration a component feels during a shock. Namely it assumes that the shock loading during the initial pulse is the most damaging. But there are a variety of loading mechanisms that cause additional frequency content in the input acceleration $a(t)$. In particular ship shock explosions produce bubbles that undergo cavitation, and surface and bottom reflections. These reactions could be resonant with a component. A typical timescale for cavitation closure is of order 10^{-1} sec, so this would correspond to resonant frequencies of order 10Hz. The timescales for bottom and surface reflection can be quite variable depending on where the explosion is located relative to the bottom and the surface.

5. The LWSM seems most appropriate for qualification of Class I, Hull-Mounted components. Nonetheless, the majority of lightweight ($\leq$250lb) components are Deck-Mounted, and are still qualified on the LWSM. While it is recognized that the LWSM does not provide the most realistic shock loading environment for deck-mounted equipment (Clements, 1972), the deviation is thought to only provide an extra margin of conservatism. Furthermore, the LWSM test is the least expensive test to conduct ($\sim$ \$5k).

6. The component testing procedures can not be used on all components. Some ship components are too large to test and these are qualified with DDAM. But as discussed above the assumptions behind DDAM fail when the structure is as large as the spatial extent of the ship modes

themselves. Hence DDAM is most critical to use in the situations where it is the least certain.

7. Finally the possibilities of system failures (cooperative failures between components) is not at all addressed.

These remarks make it clear that the component testing procedures do not mitigate the full risk of a ship surviving a shock at the design limit. Even after component testing is complete, there is the real possibility that a Grade A component might fail. Since by definition any such failures are unacceptable ,the Navy is obliged to carry out additional tests, and this is the role of the FSST.

5 FULL SHIP SHOCK TRIALS

5.1 Background

The full ship shock trial (FSST) is a test of a ship to survive a (single) underwater explosion at 2/3 the design level. Since it is a full ship trial it does address many of the issues described above: in particular it probes whether the components survive shock in their environment on the ship; it probes the possibilities of system failures, and large components that could not be otherwise tested.

This is a significant risk mitigation procedure, and indeed as outlined below, historically FSST's have documented failures in Grade A components. On the other hand, it is a mistake to assume that an FSST mitigates the entire risk–if a ship passes an FSST without mishap it is still possible that a ship in a military environment could experience a Grade A failure below the design threshold. This is because:

1. By their very nature FSST's cannot explore every potential explosion geometry; clearly the damage of a blast is strongest for the equipment closer to the impact site on the ship and hence such components have an increased probability of failure.

2. The shock loading from an FSST is just one example of a potential shock loading characteristic. If an explosion were launched at a different place, the bottom and surface reflections of the blast wave would differ and these could provoke different resonant frequency failures.

It should be noted that both of these risks could be mitigated by a validated M&S *program.*

The FSST is carried out on the initial ship of a class, as soon as possible after production. A typical ship schedule is shown in Figure 6. A series of shock tests are carried out at increasing shock factor up to 2/3 the design load. The test is carried out and managed by the ship acquisition office. During our discussions with Navy personnel it became clear that there are tensions in the FSST process that are at least in part caused by this management structure. Shock trials cost time and money, and FSST occurs at exactly the time where there is the least incentive to go back to the drawing board to fix any issues that arise. We were told that although retrofits are recommended for equipment failures, there are tensions relating to schedule and costs about whether these retrofits would be implemented.

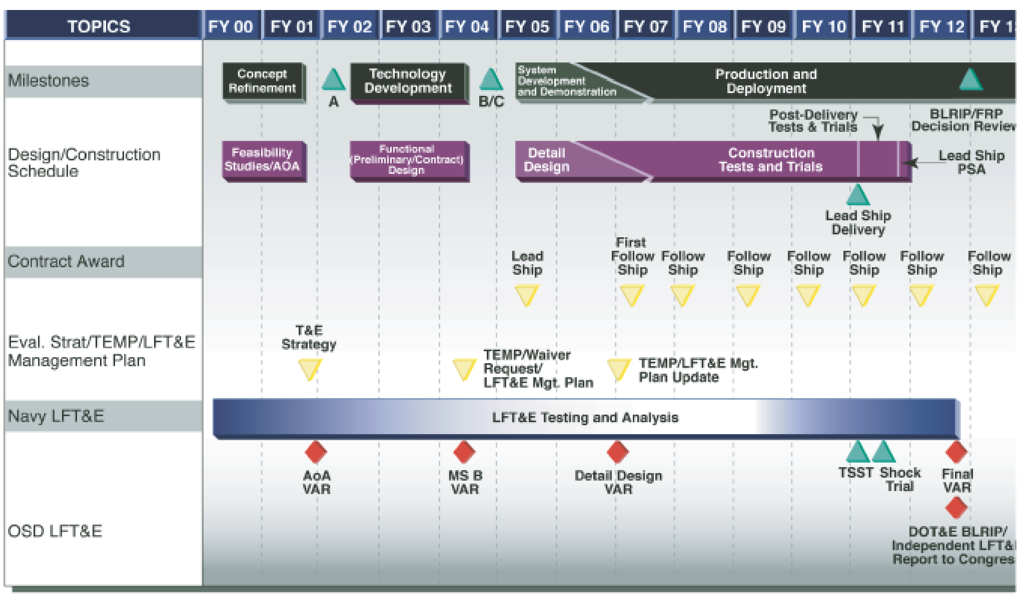

Figure 6: Typical acquisition schedule of a ship class.

There are, however, clear benefits of the shock trials; a clear illustration of this can be seen in a summary of the most recent two shock trials, on the DDG53 and DDG81, respectively.

5.2 Results of Recent FSST Tests

The FSST is the only test which can identify systems-level, shock-induced failures in shipboard equipment and infrastructure. Due to size and weight restrictions of the component test platforms, even some critical components can only be wholly tested during the FSST. If no failures were identified in the FSST, then the value of the FSST would primarily be in certification, i.e. “checking the box.” Review of the FSST reports for DDG 53 and DDG 81 indicate that this is not the case.

5.2.1 DDG 53

The DDG 53 (USS John Paul Jones) FSST was conducted in June of 1994. The USS John Paul Jones is one of 57 destroyers in the Arleigh Burke class, *20 of which were already in production at the time of this FSST.* The Shock Trial Officer’s memorandum (17 November 1994) indicates that the “John Paul Jones Shock Trial is considered the most successful trial of any surface combatant.” In three of six Mission Warfare areas, the DDG 53 saw no degradation in full operational capability during the FSST. In the other three categories, the restoral times were generally much better than those observed in previous FSSTs. Of the 11,772 shock-qualified Grade A and B components (4,460 unique) aboard the USS John Paul Jones, only 15 failed or malfunctioned.

Nonetheless, the consequences of the component failures were in some cases severe. A subset of these few component failures were generally the drivers behind the ships’ time to restore full operational capability, which is still measured in hours. Other component failures, though not severe in consequence during the FSST, would likely be more extensive and significant in consequence under full design shock loading. Degradation in major ship systems were observed as a result of component, subcomponent, and subassembly failures. The Shock Trial Officer indicates that six "mission-critical" systems in particular "will require further investigation *and design review*." [italics ours] Those systems are:

- AN/SPY-1D Radar
- MK 99 Fire Control System
- Close-in Weapon System (CIWS)
- 5"/54 Gun
- SQS-53C Sonar
- VHF Antennas

Degradation in some of these systems may be considered downstream effects, such as loss of command and control functionality, power connectivity, or cooling water. These are the types of systems failures which can only be probed in the FSST. Some of the degradation, however, was a result of a local mechanical failure that *should* have been mitigated by a prior component test. The assumption, then, is that either the component [subcomponent, subassembly] was configured differently once aboard the ship, or else the component test did not properly simulate the design shock load and support impedance. After all, the component test is supposed to probe response under shock loading which is 50% greater in magnitude than the FSST (as measured by the keel shock factor).

The DDG 53 FSST summary report expresses a similar uncertainty in the universality of MIL-S-901 (component test guidelines). While noting that most MIL-S-901 shock-qualified components remained operational, it states that "the shock trial also revealed some areas in need of improvement." In particular, the shock-qualification procedures for the Stern Tube Seals are called out. This is a Grade A component which was shock qualified prior to the FSST, but failed in multiple instances throughout the ship. The report suggests that the component test environment was not appropriate, and that the issue should be revisited. The same recommendation is given for the switchboard circuits, though none of these failed during the FSST. The implication is that these are very critical components which can not afford to be tested and shock-qualified in an inappropriate environment. Data was collected in the location of each of the switchboard circuits, with the express purpose of gaining more accurate information on the accelerations, velocities, frequencies, etc.

5.2.2 DDG 81

The DDG 81 (USS Winston Churchill) FSST was conducted in May and June of 2001. Like the DDG 53, the DDG 81 belongs to the Arleigh Burke class of destroyers. The DDG 81, however, is the second ship of Flight IIA, which incorporates significant design and equipment changes with respect to Flight I ships (e.g., DDG 53). The DDG 81 FSST was conducted primarily to test the shock survivability associated with these changes, as well as shock hardening features added in response to the DDG 53 FSST lessons learned. In many regards, the DDG 81 FSST was less successful than the DDG 53 FSST. Long restoral times - some measured in days - were observed in the Mission Warfare areas. In the executive summary of the FSST report, these "serious failures and significant mission degradation" are attributed to relaxed manufacturing standards on both the shipbuilder and the equipment vendors. In addition, many systems failures described in the report are linked to errors in "damage control" (crew response).

Unlike the DDG 53 FSST summary report, the DDG 81 FSST report does not provide an explicit count of the number of shock-qualified Grade A and B components which failed. It may be surmised from the text that approximately the same number of components failed or experienced degraded capability ($\sim$15). It may also be surmised that systems-level failures were more prevalent, wide-spread, and long-lasting. The extent to which these systems-level failures were a direct result of failure in shock-qualified components is unclear.

The DDG 81 FSST report does, however, clearly indicate the utility of the FSST. The report lists twelve specific shock-hardening modifications accelerated for the DDG 81 FSST, all based on observations and recommendations from the DDG 53 FSST. Some of these, such as the modifications made to the AN/SPY-1D radar, were performed on large, mission-critical systems and resulted in improved survivability. Furthermore, the DDG 81 FSST generated its own set of lessons learned: "Post-trial assessment of failures experienced during the shock trial resulted in shipboard and equipment modifications, identified areas requiring further investigation or analysis." Numerous recommendations were made for "backfitting" all ships in the class, as well as for performing quality assurance inspections on specific critical subcomponents on ships constructed in the future.

5.3 Issues with FSST

The results of the ship shock trials for the DDG53 and DDG81 make explicit the utility of the ship shock trials in assessing the ability of grade A components to withstand shock. Nonetheless, there are a variety of issues with the shock trials that were emphasized to us during our briefings.

1. Cost was emphasized as a major concern. The cost of the DDG53 ship trial was \$28.3M, while that of the DDG81 was \$43M.

2. Environmental concerns (in particular the risk of hurting marine mammals) are significant. Indeed, environmental issues delayed the DDG53 FSST by 4 months, because of legal battle with environmental group (NRDC). This delay resulted in a change in the weather pattern at the test site, which caused further delay. Ultimately this led to the cancellation of two of the four test shots. Although there were no lawsuits prior to DDG 81 FSST, originally an FSST was scheduled for the DDG 79 (first of Flight IIA); the environmental documents were not done in time so tests were done on the second ship in the flight (DDG 81).

3. Environmental delays aside, the FSST occurs too late to have any impact on the design process of the ship. (See Figure 6.) This is because the FSST occurs at best after the delivery of the primary ship in the class and by the time this ship is delivered of order 5 ships have been contracted. Note in the DDG53 test there had been 20 ships contracted at the time of FSST, and for the DDG81 FSST did not even occur on the first ship in the class.

4. There is institutional skepticism about whether the trial corresponds to real threats, or whether it is necessary at all.

5. Retrofits resulting from FSST are very expensive and are apparently not always carried out.

6 MODELING AND SIMULATION

M&S is already well used by the Navy through DDAM. The most complete way to go beyond DDAM is to simulate a blast directly. The simulational problem can be broken down into three independent parts:

1. The water response Given an explosion how is the pressure impulse propagated through the water, and what is the impulse that hits the ship?
2. The structural response How does the ship structure respond to this impulse?
3. The component response Given the structural response, predict the vibrations of the components and potentially their failure modes.

Current capabilities are well on their way for the liquid and the structural response, and we analyze current Navy capability for these capabilities below. As mentioned before, the possibility of reliably predicting component failure with numerical simulations is so far beyond current capabilities, and would be so expensive (because of the need to extensively calibrate all component failures with experiments) that we view this as extremely implausible. It is our assumption in what follows that the proper way to extend M&S capability is to still maintain component testing, but to use M&S to mitigate the various risks in the component testing protocols we have outlined above.

6.1 The DYSMAS Code

Since our assessment of Navy capability will be through the use of the

DYSMAS code, it is worth beginning our discussion with a brief overview of the DYSMAS code.

The DYSMAS (DYnamic System Mechanics Advanced Simulation) code was codeveloped with the Germans. It consists of the coupling of a Boundary Element Method solver for bubble and fluid dynamics (DFBEM), an Euler solver for Shock and Fluid Dynamics (Gemini); a Lagrange solver for the structural response (DYNA-N and Paradyn), coupled together with a standard coupler interface. The code was initially developed for undersea weapons applications (such as the design of warheads) but now it is also being used and discussed for Live Fire Testing and Evaluations. Indeed, the largest userbase for the code is currently Live Fire Testing and Evaluations.

DYSMAS was chosen as the main Navy code in the early 1990's, after competing its capabilities against several alternatives on four typical UNDEX problems. As stated by Gregg Harris, DYSMAS is "not as elegant as other research codes, but it is more mature". DYSMAS was co-developed with the Germans under two international agreements. In the first project agreement, there was small scale validation and code upgrades for UNDEX applications. In the second project agreement (2003-2007) there was validation on a full scale ship. This latter event, carried out on a decommissioned German destroyer, the Lütjens, was a highly significant test, the data for which plays a significant role in our assessment of current Navy capabilities.

The Lütjens test consisted of a series of explosions and comparison with pre and post test simulations. The finite element model for the Lütjens was developed by IABG and NSWC CD code 65.

6.2 The Liquid Response

We organize the discussion as follows. After a brief consideration of the basic physics of the liquid response, we discuss Navy capability in the context of the recent Lütjens trial, designed to test the DYSMAS code.

6.2.1 Basic physics

An explosion leads to a pressure wave launched into the liquid, and this pressure wave interacts with the ship. At the simplest level we can approximate this pressure wave as a spherical wave resulting from the initial explosion. The theory of sound waves in liquids dictates that the pressure amplitude at the hull of the ship is given by

$$P_{hull}(t) = \epsilon_{HE} \frac{\ell}{R} e^{-t/\tau}, \tag{6-8}$$

where ϵ_{HE} is the energy density of the explosive, ℓ is the linear dimension of the explosive, R is the distance between the explosion and the ship, and τ is the timescale over which the pressure decays. The decay time scale is given by

$$\tau = \frac{\ell}{c}, \tag{6-9}$$

where c is the speed of sound in water. Note that if ρ is the mass density of the explosive, the size ℓ is related to the explosive weight W through $W = 4\pi/3\ell^3\rho$. Evaluating the formulae for the special case of TNT we have

$$P_{hull}(t) = 1.2 \times 10^5 \frac{W^{1/3}}{R} e^{-t/\tau}, \tag{6-10}$$

$$\tau = 0.03 W^{1/3}, \tag{6-11}$$

where the pressure is measured in pounds per square inch and the timescale in milliseconds. These formulae agree well with the correlations for explosive strength often reported in the literature.

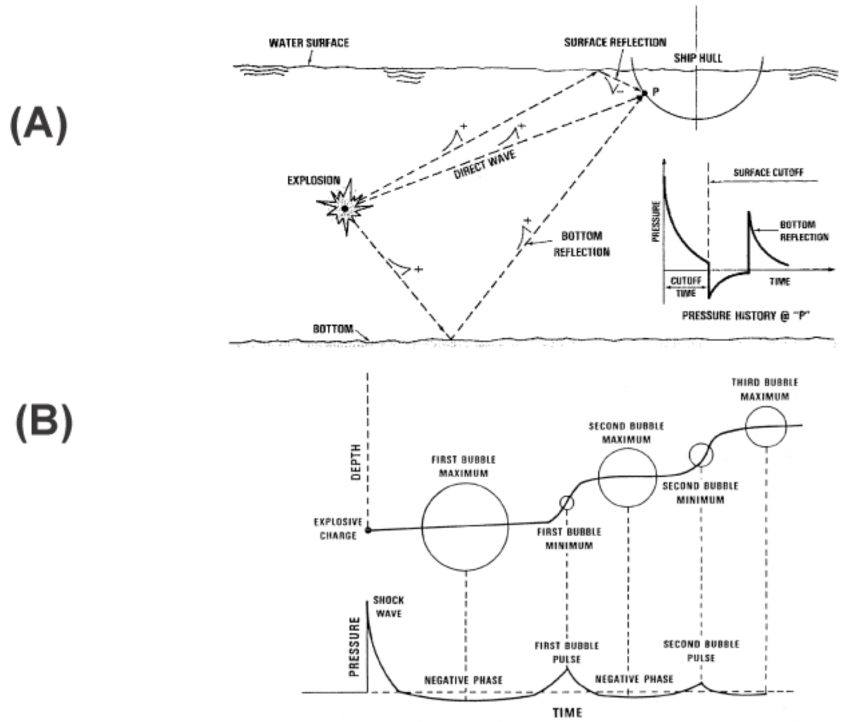

Figure 7: Complications to the simple acoustic propagation picture for the liquid response to an explosion. (A) Bottom and surface reflections lead to additional pressure impulses at the hull of the ship. (B) Bubbles produced as a result of explosions oscillate, leading to their own pressure waves. (Courtesy of Fred Constanzo)

There are two complications to this simple picture of the explosion that can be significant, as illustrated in Figure 7. First, the pressure wave from an explosion can reflect off of both the bottom and upper surface. This causes additional pulses at the hull of the ship. Secondly, it has long been known that explosions under water lead to the formation of bubbles which themselves emit pressure waves.

6.2.2 DYSMAS capabilities: liquid response

The liquid response capabilities of DYSMAS were tested during the Lütjens test. Pressure transducers in the liquid surrounding the test measured the liquid response and these were compared to simulations. The results were impressive. Figure 8 shows the pressure response as a function of time. The code (dotted red line) captures the measurements (solid line) including the initial pressure impulse and the reflections. The simulations indicate that the secondary pressure pulse is due to a bottom reflection, and the pressure pulse 120 msec after the initial pulse is due to cavitation closure.

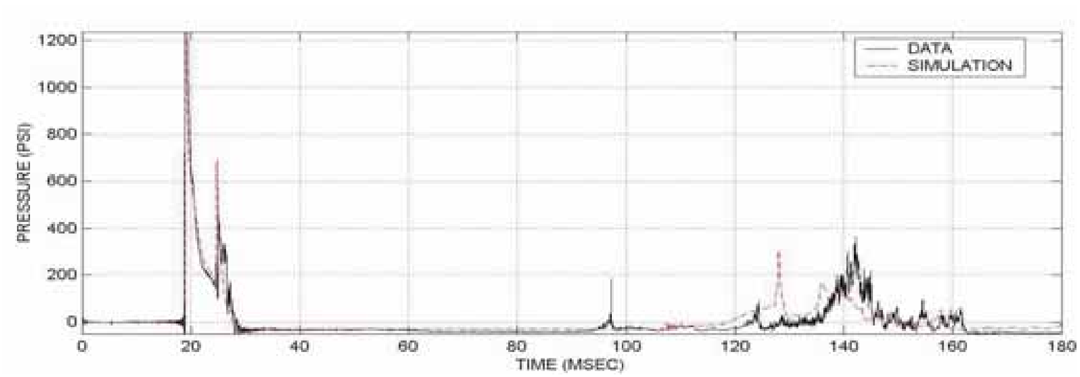

Figure 8: Comparison between predicted pressure response and measured pressure response for the Lütjens test. (Courtesy of Gregg Harris)

Figure 9a shows quantitative agreement between the initial pressure pulse and the measurements. The red curve in this figure shows a simulation with 5cm cells, whereas the blue curve shows a simulation with 5mm cells. It is seen that the simulations converge onto the measurements when resolution is increased. Figure 9b shows the pressure impulse per area as a function of distance to the explosive. As expected from the simple physical arguments presented above, the impulse decreases like R^{-1}.

These comparisons demonstrate that the DYSMAS code does an excellent job of reproducing the liquid response to the explosion.

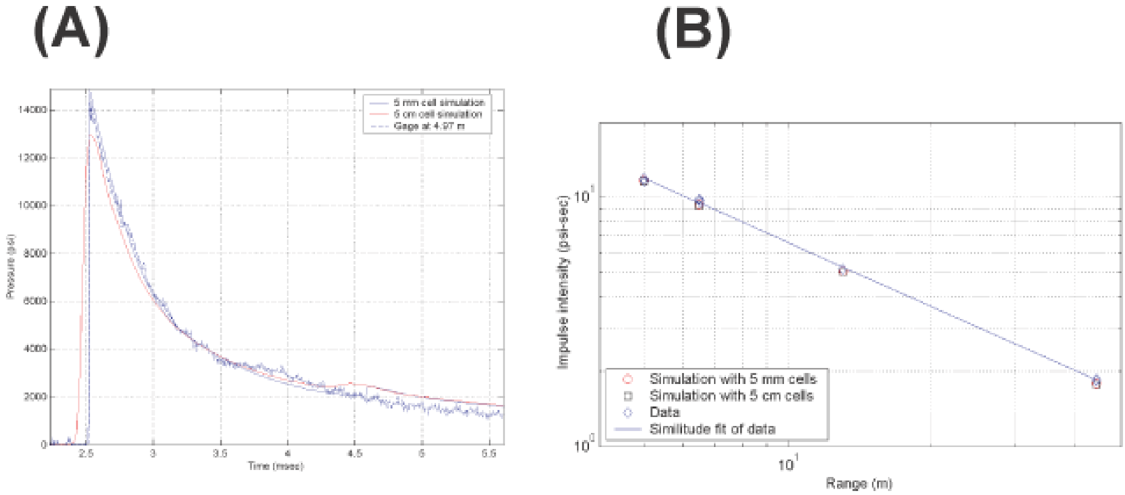

Figure 9: Comparison between predicted pressure response and measured pressure response for thh Lütjens test. (A) Quantitative comparison between initial part of pressure pulse and measurements, for two different resolutions. The blue curve is 5mm resolution whereas the red curve is 5cm resolution. (B) Pressure pulse strength as a function of distance from the explosive. (Courtesy of Gregg Harris)

6.3 Structural Response

We now consider the basic physics of the structural response, and DYSMAS's ability to predict it.

6.3.1 Near impact

Near the impact point, the structural response of the ship shows a characteristic signature. Initially the vertical velocity obeys

$$v(t) = \frac{I(t)}{m_{ship}} = \frac{A \int_0^t \, ds p(s)}{m_{ship}} = \frac{A p_{max}(1 - e^{-t/\tau})}{m_{ship}}, \tag{6-12}$$

where $I(t)$ is the vertical impulse to the ship up to time t, and m_{ship} is the mass of the ship. After the initial impulse is over, the ship has been pushed

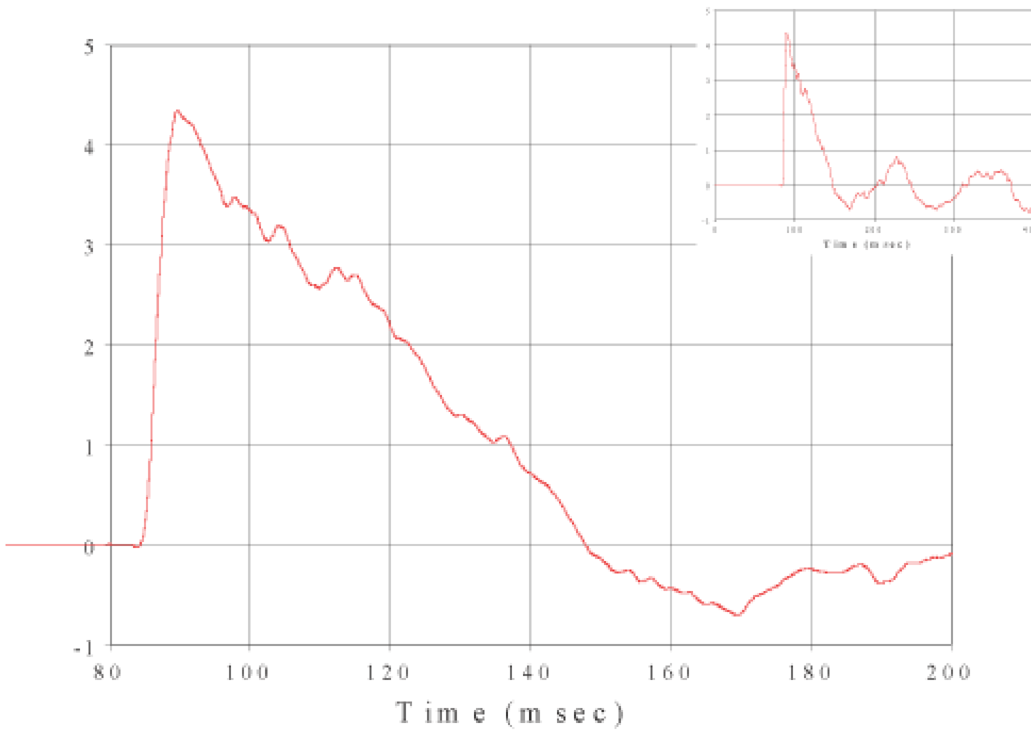

Figure 10: Initial response of the ship due to the explosion. INSET. Response of the ship over slightly longer timescales (Courtesy of Gregg Harris)

out of the water (this happens because of cavitation bubbles that occur near the point of impact); the ship then falls, accelerating with gravity. Figure 10 shows the response of the ship during the initial impulse and deceleration. After this initial behavior the ship vertical velocity oscillates with time, at a frequency of the relevant normal mode (the mode that is most excited at the point of the measurement). The simulations capture the initial response of the ship quite well.

6.3.2 Entire ship

The structural response of the entire ship involves the excitation of normal modes of the ship. Numerical simulations have evolved to the state where they can in principle capture this excitation. Figure 11 shows the finite element model used during the Lütjens test. The model contains 292,000 different finite elements and of order 10^6 degrees of freedom. Figure 12 shows a snapshot of a bulkhead of the ship oscillating after the blast. The entire ship moves upwards, and strong localized deformations are seen in the deck modes, with different parts of the decks moving at different frequencies from each other. It is unfortunate that the complete movie cannot be included as part of this report: the movie gives a visual display of the complexity of the response to be captured and also why equipment mounted in different ways in different locations on the ship needs to be examined separately.

More precise comparisons between simulation and experiment can be seen in Figures 13,14,15,16. The location of the blast is shown in Figure 13 with the yellow marker. The blue curve gives the measurements near the blast and the red curve gives a calculation. Figure 14 shows comparisons on different sides of the ship, with the blue curve representing measurements, the green curve representing pre-test simulations and the red curve representing post-test simulations. Some phasing errors are observed, and the simulations show fast short time oscillations that are not present in the measurements. Displacement comparisons are shown in Figure 15, with qualitative shapes quite consistent between simulation and measurements. Figure 16 compares the mast motion to the simulations: this is the worst comparison; for the atwhartship movement the two curves are uncorrelated and oscillating with different frequencies.

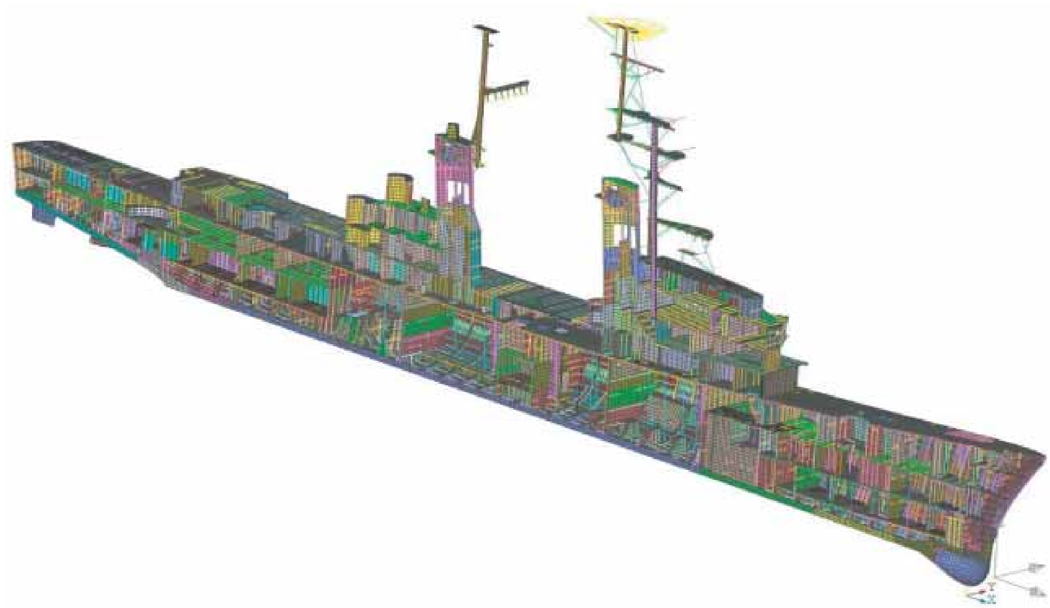

Figure 11: Finite element model of the Lütjens destroyer. (Courtesy of Gregg Harris)

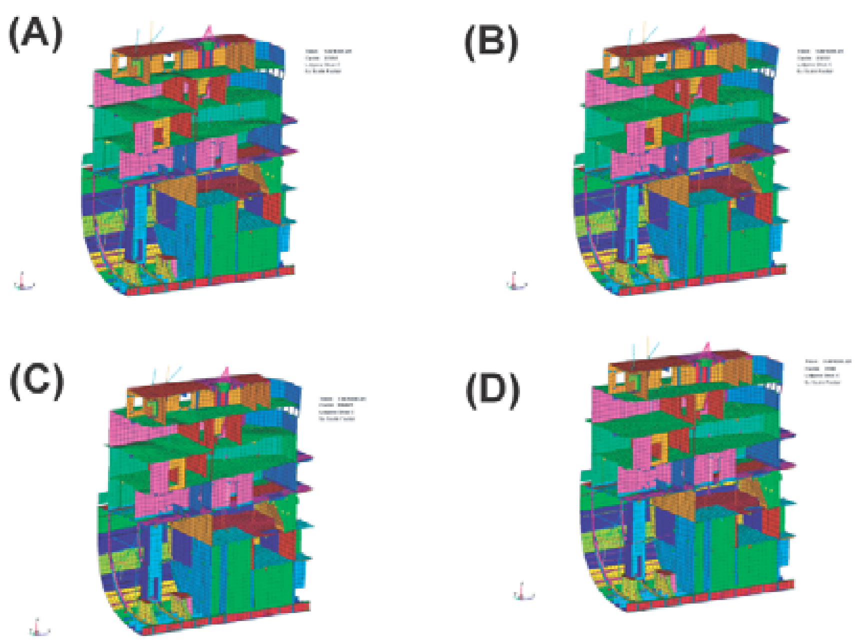

Figure 12: Four snapshots from a DYSMAS simulation of a bulkhead after the UNDEX event. Note the strongly localized deck modes that are excited by the blast.(Courtesy of Gregg Harris and Fred Constanzo)

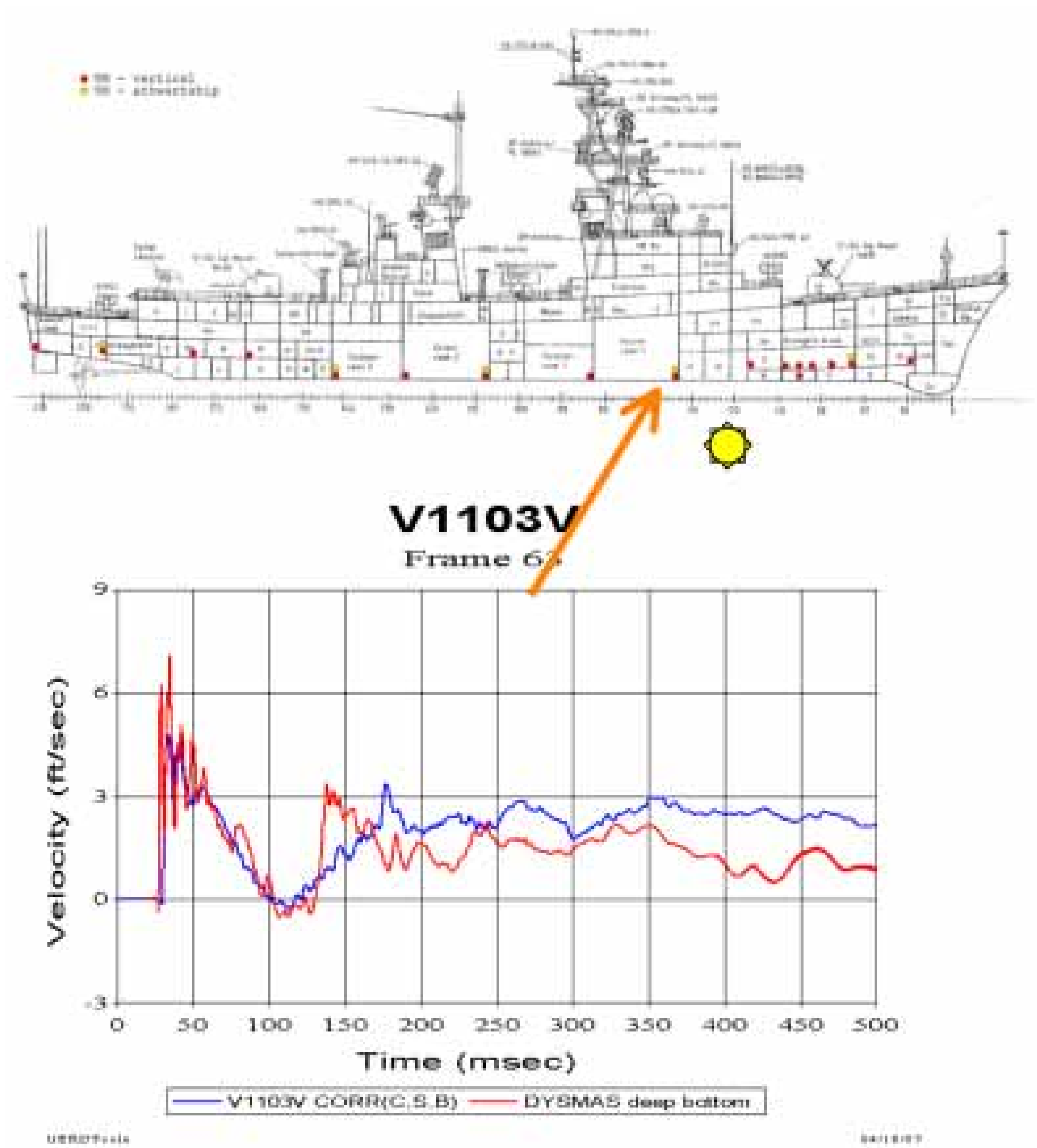

Figure 13: The yellow marker denotes the location of the blast. The blue curve gives measurements of the velocity at the indicated point on the ship, while the red curve gives posttest simulations. (Courtesy of Greg Harris)

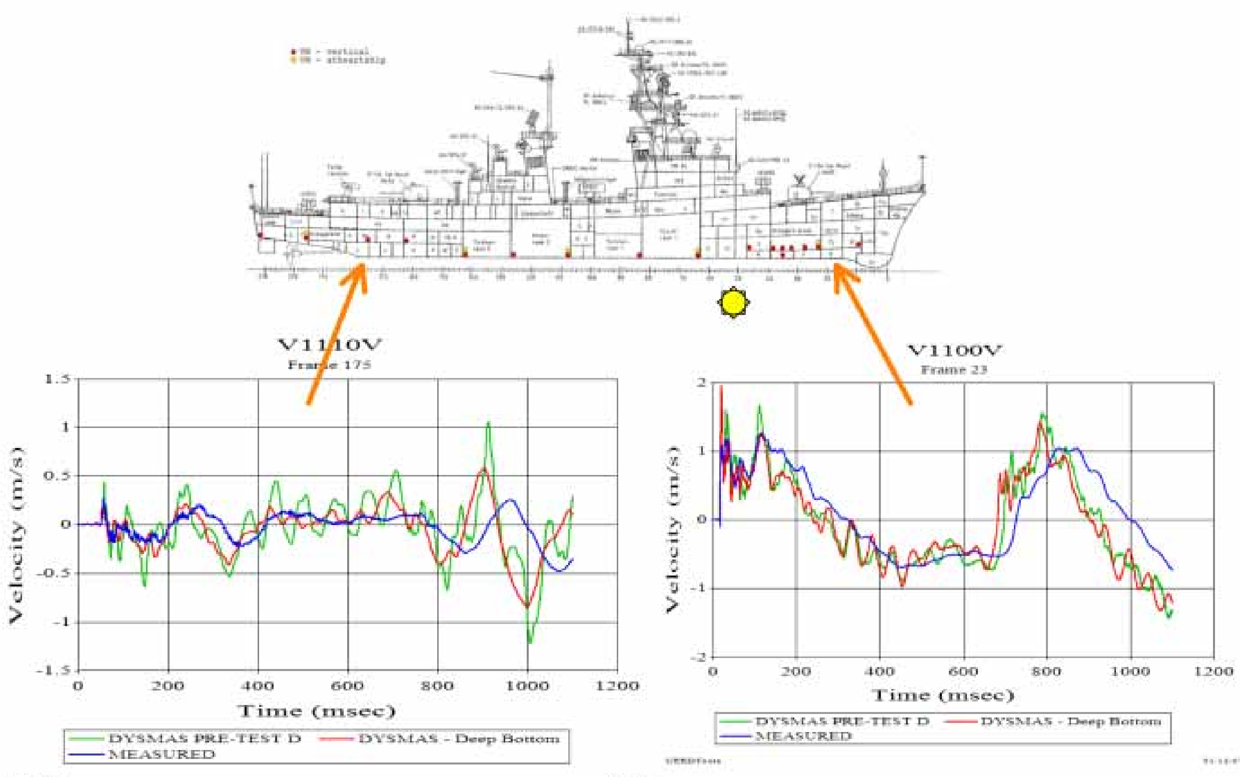

Figure 14: The yellow marker denotes the location of the blast. The blue curve gives measurements of the velocity at the two indicated points on the ship, while the red curve gives posttest simulations and the green curve pretest simulations.(Courtesy of Gregg Harris)

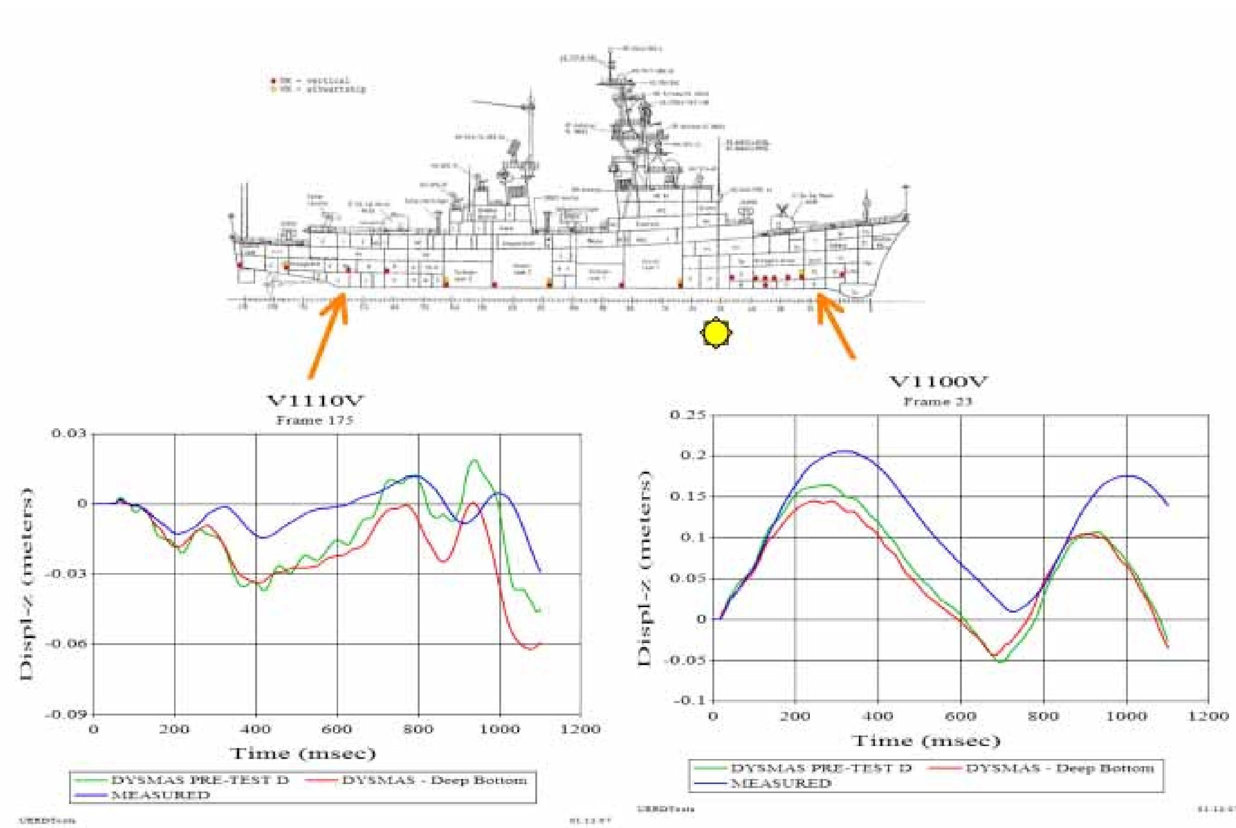

Figure 15: The yellow marker denotes the location of the blast. The blue curve gives measurements of the displacements at the two indicated points on the ship, while the red curve gives posttest simulations and the green curve pretest simulations.(Courtesy of Gregg Harris)

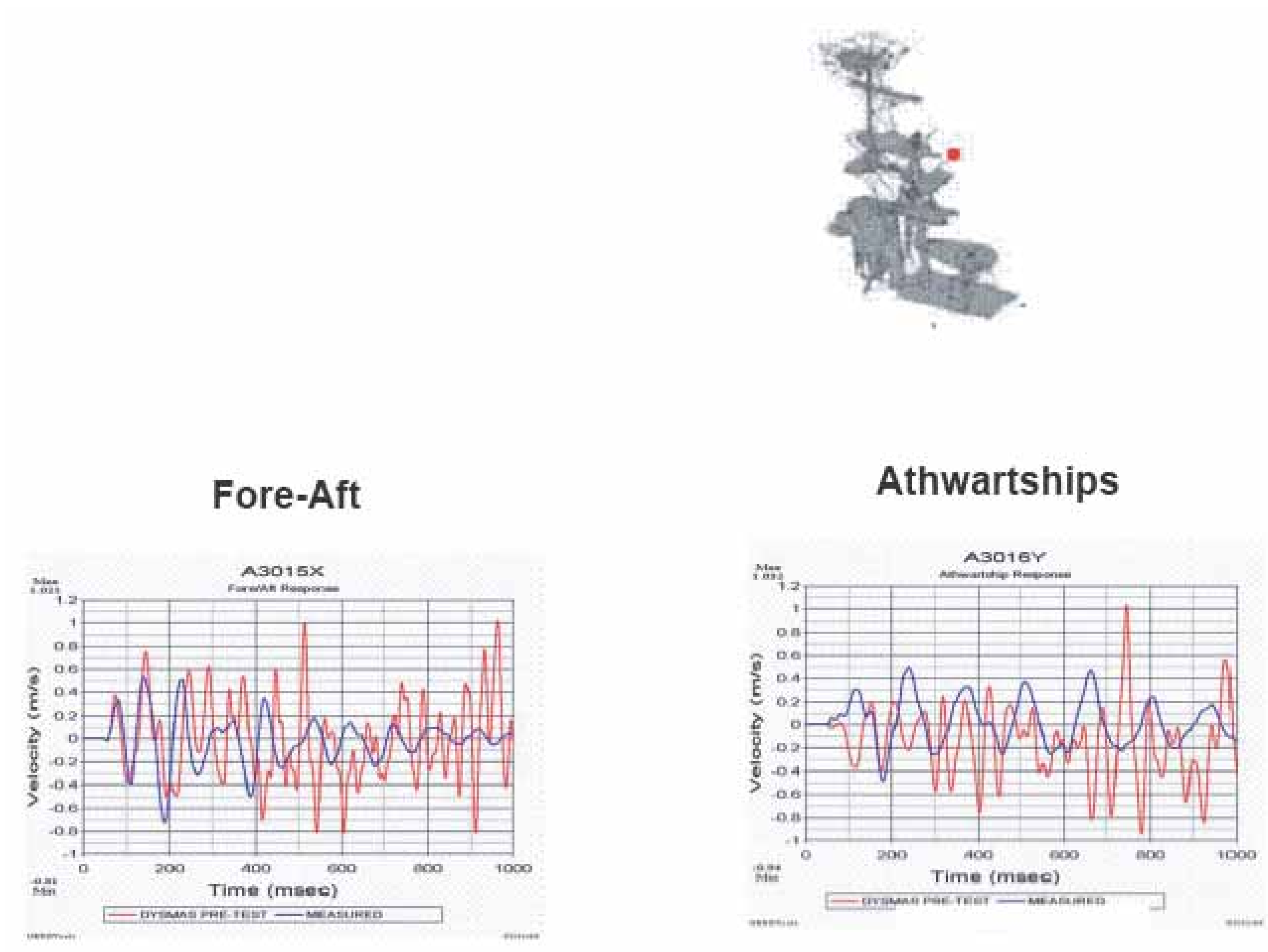

Figure 16: The blue curve gives measurements of the velocities of the mast motion in the fore-aft and athwartship direction, while the red curve gives pretest simulations.(Courtesy of Gregg Harris)

6.3.3 DYSMAS capabilities: structural response

The agreement with the short time structural response is excellent, in particular with the result of the initial impulse near the point of impact and the subsequent fall due to gravity. The structural response after this fall and further from the impact point is harder to assess. The calculations show strong correlations to the measurements across the ship for both velocities and displacements. On the other hand there are systematic errors: there are short time scale oscillations in the simulations that do not exist in the measurements, and there are phase errors between measurements and simulations.

These comparisons raise an important question: how good is good enough? Some analyses use statistical metrics such as the Russell correlation index, together with a threshold based on historical or anecdotal evidence. However without directly connecting the oscillations that are observed on a ship to the probability of component failure it is difficult to be sure that these comparisons are good enough for their desired purpose. With respect to the component testing one must conclude *that without a clear and convincing argument to the contrary*, it is unclear whether the simulations are good enough to mitigate the risks now taken on by FSST. We believe that a principal goal of the Navy's future work on developing simulational capabilities should be to develop the necessary data and arguments to resolve this issue. We will return to this very important point subsequently.

What are the uncertainties in the simulations that could be the cause of the discrepancies? The largest uncertainties are:

1. Damping and dissipation. No reasonable model of damping and dissipation for ship oscillations exists. To not introduce additional uncertainty, the DYSMAS simulations of the Lütjens test assumes no damp-

ing. Perhaps this is the reason for the short time scale oscillations in the simulations that do not exist in the measurements.

2. Another uncertainty is in the finite element model itself. Choices are made about how parts of the ship should be precisely modeled. Unresolved components must be dealt with in the model in some way. We were told in our briefings that the choice of model can be made only with substantial experience; how accurate the models are, and where the dominant error occurs is currently unknown. Most strikingly there are no systematic metrics for estimating the errors in the finite element models that are commonly used.

3. Finally, there is the question of transferring energy from the resolved scales to the unresolved scales in the problem, such as joints and welds. This energy transfer is like dissipation in that it results in energy that is lost from the simulational degrees of freedom; but it is unlike dissipation in that the energy is not dissipated to heat and can in principle couple back to the simulated degrees of freedom later on.

An appendix to this report offers quantitative estimates about the relative importance of these effects, and suggestions about how they might be dealt with.

6.4 Findings on the Lütjens Trial

The Lütjens trial was an enormous success. This was one of the first (if not the first) full scale code validation test comparing simulations and experiments, and it clearly demonstrated the strengths and weaknesses of the simulation capabilities. The Lütjens test was not done with the question of FSST replacement in mind, and for that reason there was no component testing involved in the trial: Lütjens was *not* a full ship shock trial. No electrical or systems failures were monitored and thus from this test we cannot assess the ability of the simulations to indicate potential failures in a shock trial.

6.5 Findings on General M&S Capabilities

The historical Navy M&S capability – DDAM – is a reasonable, low cost method for benchmarking components. The acceleration spectrum is based on sound physical arguments, though the data that underlies it is more than 50 years old. Current and developing M&S capabilities give significant opportunities for updating DDAM.

Beyond DDAM, M&S can be divided into liquid and structural capabilities. Liquid capabilities are important for understanding the precise pressure impact on a ship given an explosion. This capability is critical for matching experiments with simulations in an actual trial.

Modern structural M&S capabilities offer the potential for moving substantially beyond DDAM, and

1. Connecting shock spectra to current ships and current threats. As outlined above the DDAM spectrum assumes the component experiences the same blast profile everywhere on the ship. This is not true and could be corrected.
2. Flagging components that might fail because the component tests makes incorrect assumptions about their foundational impedance.
3. Potentially identifying failure modes of large pieces of equipment; or at least providing a more principled way to simulate large pieces of

equipment. As stated earlier, DDAM's assumptions are simply incorrect when equipment is as large as the vibrational modes of the ship. In principle structural M&S offers the potential of much greater accuracy and reliability for large components than DDAM.

There is however widespread skepticism in the Navy over the use of M&S for qualification. We believe that this is mainly because of the lack of proper validation. The Lütjens test was in important initial step in this direction but more needs to be done. Indeed, at present there have been no explicit comparisons between a full ship shock trial and M&S. It is therefore completely unclear whether M&S can predict the failure modes of an FSST.

6.6 How Good is Good Enough?

This question is of critical importance. The answer must be substantive enough to address criticisms and concerns about whether simulations can validate tests. To this end, it is not sufficient to simply observe that the simulations roughly agree with the measurements and reproduce the correct trends. Instead it is necessary to argue that the comparison between measurements and simulation is such that the simulations are expected to give (or not give) the information required to predict ship survivability and component failure. Metrics should not be based on statistical quantities unconnected to failure modes (e.g., the Russell correlation index), coupled with anecdotal accounts of the level that is acceptable.

We believe that the proper addressing of this question is important enough that the entire validation program for M&S capability needs to be built so that the question of metrics is answerable. We will return to this issue below.

7 WHAT SHOULD BE THE GOAL OF A SUCCESSFUL NAVY What M&S PROGRAM?

Before proceeding further, we believe it is worthwhile to take a step back and ask what the Navy stands to gain with a successful M&S program. Understanding the full benefits is important because it will help establish a business model for M&S.

With respect to FSST, there are two sets of goals for the Navy M&S program.

1. The M&S program could substantially decrease the probability that FSST leads to expensive retrofits. This could be done for example by improving the correspondence between the component testing procedures and operational ships.
2. A more ambitious goal would be to completely replace FSST with M&S.

While we agree that both of these goals are quite reasonable, we think that the opportunities for using M&S go well beyond this. In particular a validated M&S capability will lead to opportunities to both

1. Assess threats realistically.
2. Control costs, by using M&S as part of the design process,

Indeed, we think there are legitimate arguments that a validated M&S capability could lead to significant improvements in the Navy's shock hardening program as a whole.

7.1 Threat Evaluation

For threat evaluation, a validated M&S capability would provide a mechanism for connecting current threats to shock hardening: various threat scenarios could be simulated and the ability of the threat scenarios to cause ship damage could be assessed. Currently an FSST only allows a limited number of explosions in the vicinity of a ship. In contrast there essentially are no limits to the number of simulations that could be run, and hence a wide variety of different possibilities could be considered. These simulations could be used to predict the vulnerabilities of ships.

When new threats emerge, M&S could be used as part of the process to connect these threats to vulnerabilities.

The need for better threat assessment were underscored during our briefings by A. Maggioncalda, the Life Fire Testing and Evaluation Manager for the DDG1000. She argued that there is a critical need for understanding "the distribution of forces seen at different component locations", and "the probability of various spaces experiencing ... forces in excess of component level tolerances." This information, she argued, would lead to the ability to " concentrate funding on areas that provide the best chance for UNDEX survivability against realistic threats."

Indeed, within the development of the DDG1000 there are serious efforts underway to include threat assessment as part of the design process. Figure 17 shows a picture of the various scenarios that are being considered by the M&S component of the DDG1000 team. This clearly indicates the types of analysis that are possible. The difficulty that still exists is that the M&S capability is still not validated to the level that there is sufficient confidence in the conclusions to take them as seriously as they could be in the design process. We will return to this point below.

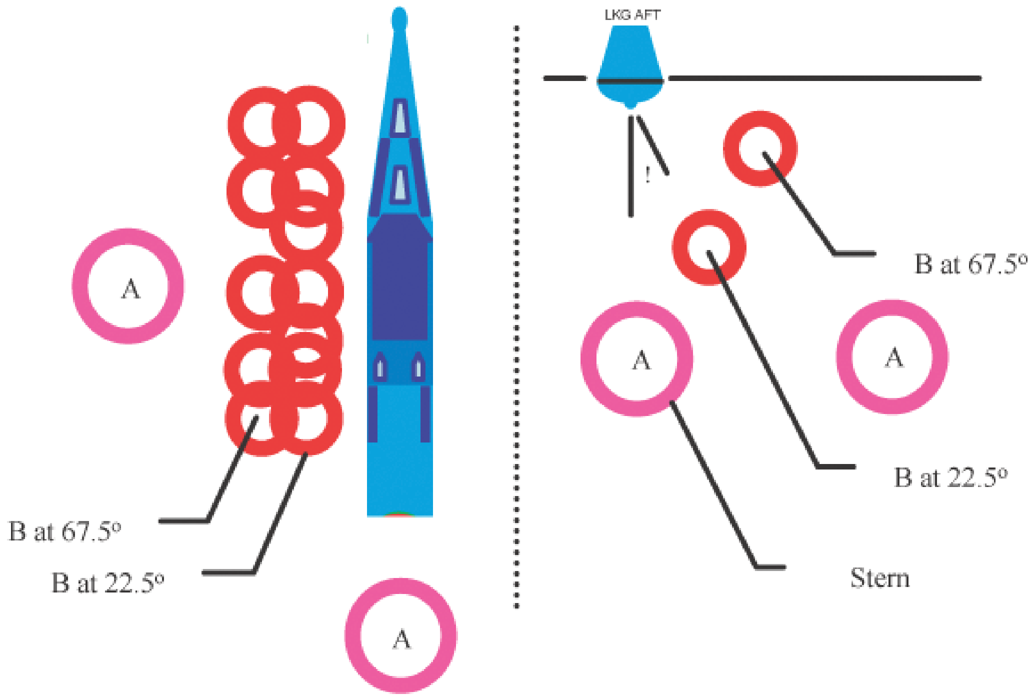

Figure 17: Set of simulational tests carried out in the design phases for the DDG1000. Circles represent the different explosion scenarios around the ship. (Courtesy of Tom Moyer)

7.2 Cost Reduction

In our briefings, the high cost of component testing and FSST was often emphasized. Indeed, for the DDG1000, estimates for the cost of FSST are $35 – $50M, while component testing is estimated at $220M.

On the other hand consider the costs of the development of the DDG1000 ship class. The total cost of a single ship is currently estimated upwards of $3500M, and the lifetime operating costs for a single ship is about $1000M. The DDG1000 class is expected to consist of 7 ships. Including the nonrecurring development costs (~ $7000M) the total cost of the ship class is about $43, 000M.

Hence, although the costs are high, the cost of FSST is about 0.1% of the total budget while component testing is about 0.5%. Thus, it is reasonable to wonder whether that considering M&S as a mechanism for cost savings through FSST replacement could vastly underestimate the savings that could be incurred. Could a validated M&S capability lead to savings in the purchase of a ship? In the operation of a ship? In maintaining the schedule for ship acquisition? Could all of this occur without leading to any increase in the survivability risk of a ship to an UNDEX event?

A significant savings here would be of order 10% of the entire budget, the cost of a single ship. Although we are in no position to examine whether such a cost savings is indeed possible, there are several basic reasons for imagining that significant cost savings might be possible.

1. A validated M&S capability could be used for ship design. Information that is now obtained from FSST could be obtained much earlier in the design process. Additionally a validated M&S capability would allow for both the ship and the component hardening to be designed for a fixed level of risk. With a full ship M&S capability it might be reasonable to re-examine the paradigm for equipment that works anywhere on a ship. For example permanently mounted critical equipment could be located to reduce their necessary shock hardening (and hence, expense).

2. M&S could identify design issues earlier, and hence speed up the ship acquisition process.

3. A validated M&S capability could help to evaluate and improve the component testing procedures that are currently in use. For the reasons outlined above we feel the scientific basis of component testing procedures needs to be reexamined. It is unknown whether current testing procedures lead to components which are over-hardened or under-

hardened with respect to the design requirement. Are the different testing procedures consistent with each other?

4. Finally, we feel that M&S combined with the monitoring of ship vibrations could help identify where replacement parts are needed during normal operation. We will discuss this concept in more detail below.

8 THE MAIN ISSUE: VALIDATION OF STRUCTURAL RESPONSE

The main issue that needs to be addressed before such ideas can be realized is that *the* M&S *procedures for structural response are not validated at the level required for reliable prediction*. Although there is qualitative (and in some cases even semi-quantitative) resemblance between pretest simulations and measurements, it is not understood whether the agreement is sufficient for the simulations to either validate or cast doubt on the conditions under which component tests were carried out.

With respect to the liquid response, the Lütjens trial showed that predictions of the propagation of the explosion through the liquid are much more reliable. Simulations accurately capture the impulse imparted by the blast to the ship, and the motion of the ship during the initial impulse is reasonably captured.

8.1 Validation Metrics

To establish whether the structural predictions are sufficient, we need metrics for validation. We discussed above that the failure mode of a component depends on

(a) the peak acceleration, if the characteristic frequency of the component is larger than 6 khz;

(b) the product $V_{kickoff}\omega$, where ω is either the frequency of the component or the oscillation frequency of the part of the ship where the component is located; or

(c) in the case of resonance, $NV_{kickoff}\omega$, where ω is the resonant frequency of the component and the forcing frequency of the part of the ship where the component is located, and N is the number of periods for the ship oscillation to damp out.

As discussed above, component testing procedures are designed for a fixed $V_{kickoff}$, with some imprecise allowances made for the mounting locations on the ship. Hence the effectiveness of component testing procedures depends on whether the frequencies and damping coefficients of ship oscillations are consistent with those assumed by the component testing procedures.

The validation metric must directly address the issues that M&S *is trying to address.* Given that a major role of FSST is to mitigate the risk of the component testing procedures, the computer codes must be able to accurately predict both the modal frequencies and the damping coefficients of the modes that are excited during a ship shock. The accuracy must be sufficient to reproduce accelerations and the possibilities of resonant interactions across the ship. The frequencies and dampings need to be predicted accurately enough that the tolerances of the component testing procedures are sufficient to guarantee that critical components do not fail during the test due to inappropriate environment assumed by the component testing procedures.

The issue here is both that the component testing might use different frequencies and dampings of mounts than those encountered on a ship, as well as the fact that the requirement of fixed kick off velocity is derived from the low frequency limit of the DDAM response spectrum discussed above; real threats on a real current ship might lead to a very different spectrum for the component testing procedures to match.

We should remark here that the Verification and Validation procedures required for the present purposes differ substantially from those used by the

Department of Energy for nuclear weapons assessment. The complexity of controlling margins and uncertainties for the yield of a weapon is a far simpler problem than guaranteeing the same for every Grade A component on a ship. We feel that the DOE approach, while valuable for their application, has limited applicability in the present case for FSST. Instead we will recommend below

9 MULTI-PURPOSE CONTINUOUS MONITORING INSTRUMENTATION FOR SHIPS

The validation metric of accurately predicting normal mode frequencies and damping coefficients requires that these quantities be accurately measured. Although the results of an FSST does depend on modal frequencies and dampings, there are much easier and direct methods for measuring these quantities than carrying out a shock trial. In particular, we believe that the best way for doing this is to measure ship vibrations directly, during normal operation of a ship. The utility of such measurements is manyfold, ranging from:

1. Monitoring ship response in full ship shock trials (FSSTs);
2. Monitoring the condition of the ship and its subsystems to inform the ship's commander of the readiness of all critical subsystems prior to entering hostilities (or at any other time);
3. Monitoring for condition-based maintenance;
4. Monitoring the structural and component vibration environment and response;
5. Validation of modeling and simulation (M&S) codes through the data obtained in items 1 and 4; and
6. Providing a detailed record of a significant vibration/shock event that occurs during ship operation, which is especially important if it exceeds the levels reached in FSSTs.

By law, a FSST must be done on the first ship in a class. Results are then used to introduce design improvements into subsequent ships in the class, as appropriate, or introduce retrofits into ships already completed, if the FSST reveals problems that are deemed important enough to warrant retrofits. Lessons learned from FSSTs can also be used to improve the design tools used for all future ship designs. According to briefings to JASON, the monitoring instruments are installed specifically for a FSST and then removed after it.

There are many ways, as listed above, the Navy could benefit by installing monitoring instruments when ships are built and leaving them on the ships. For example, the first ship in a class typically undergoes sea trials for a year before the FSST. During that time a determination of the response spectrum from ordinary activities could enable a validation of M&S tools and better predictions of the outcome of the FSST. Continuous monitoring enables the conditions of various ship subsystems to be determined at a moments notice for condition-based maintenance decisions as well as to enable the ships commander to determine if his ship is ready to undertake specific missions, including engaging enemy forces. Furthermore, the normal modes of the ship can be determined routinely using planned low-energy experiments or even using routine excitation as occurs during normal operations. Specific tests, such as dropping a known weight from a known height in particular locations, or the use of alternative testing procedures such as air guns, could be repeated to determine changed relative modal amplitudes and damping as loading changes occur, e.g., as equipment is added or moved in the ship, liquids in tanks change their levels, magazines are loaded and unloaded, and as fasteners and shock-mounts age. To the extent that linear analysis is valid, design and transient response codes can be validated.

If the instruments remain on the ship after the FSST and a significant shock event occurs, it will be possible to use the data from that event to improve design tools and operational safety above and beyond what FSSTs can provide. This is because a ship operating in hostile waters may be exposed to levels of shock that one is unwilling to expose either the whole ship during an FSST, or ships components during component testing. The instrumentation and its monitoring system could provide the desired data in a fully automated and physically secure way as is the case with aircraft that crash.

9.1 How it Might be Done

Significant ship shock events are unpredictable under operational conditions. As a consequence, collecting data on them requires an instrumentation suite on the ship that is in continuous operation. Such continuous monitoring can also provide information for condition-based maintenance assuming appropriate sensors are emplaced within subsystems as well as on structural members.

Mass-produced, relatively low-cost accelerometers that survive for years and function at high-g loads when needed have been proven by air-bag activation systems for automobiles. Proof of long-life reliability of components, connectors, and wiring after high-g launch loads is also provided by space-qualified components and instruments. Data recorders based on flash memory (no moving components) that are shock-mounted, are powered with an uninterruptible power supply (UPS), record continuously on circular buffers and stop a certain time after an event can provide data that straddle the event, i.e., both before and after it. Such recorders need not be too expensive and, at the expense of some additional wiring, more than one can be provided for redundancy. Also, depending on the relative cost of wiring to recording, one can limit signal cable runs by distributing a few recorders throughout the ship. Further, high-bandwidth network data protocols exist today that would permit only a few fiber-optic cables to run through the ship, bringing multi-channel transducer data to the recording station(s).

For the sake of discussion, assume that data from 32 instrumentation points can be delivered to each recorder. The state-of-the-art for a data collection computer based on flash memory will easily handle the required data stream. Continuous instrumentation capability by ± 10 g accelerometers with ~ 1 kHz, or higher, bandwidth is readily available today for under \$15 for 2-axis devices, for example. 1000 g accelerometers are also available, but at greater cost. If the need is established and this becomes part of standard instrumentation for ships and perhaps other DoD platforms, suitable transducers that meet the full-scale dynamic range, environmental, and ruggedization requirements could be developed for ship monitoring at low (under \$100) marginal unit costs. Automobile airbag package sensors appear to be in that range [see, for example,

`http://www.gm-trucks.com/forums/index.php?showtopic=8990mode=threaded]`.

Assuming, again, that 32 instrumentation points per data recorder and 4 data recorders suffice, for a total of 128 channels over the ship, synchronizing data and recorders can easily be achieved by distributing a common clock signal via fiber-optic cable to all data recorders. Common (even redundant) fiber-optic buses could also be employed per recorder to avoid costly point-to-recorder star wiring networks.

Assuming a 8 kS/s sampling rate and 2 Bytes per sample per point, each data recorder would receive 512 kB/s. Storing 30 min (1800 s) of data would then require less than 1 GB of memory per recorder (per 32 instrumentation points). Inexpensive ($\sim$ \$120) USB flash memories are available today with a 4 GB capacity. Thus, half-hour data intervals could be retrieved, archived, and analyzed to obtain the ships structural normal modes in routine operation. If there is no reason to archive data recorded either intentionally or inadvertently, removable flash memories can be over-written. If an event occurs (as determined by one of several passive threshold sensors), the data recorders would stop recording after an appropriate length of time and save the data to their external flash drive(s).

The data recorders UPSs would be kept fully charged by the ships power until the latter cuts off, which would be one indication to stop over-writing the memory after a predetermined time interval. The system would employ both high-load (perhaps ± 1000 g) and low-load (e.g., ± 10 g) accelerometers, depending on location, to capture both serious shock events and low-level (e.g., for condition-based maintenance) monitoring at selected locations.

In addition to sensors for vibration and shock events, it might be valuable to have the equivalent of an airplane black box, in which information from the ships normal instruments is also stored and saved after an event, together with the information from the shock sensors. Such a data suite would provide a more-complete picture of what the ship was doing up to and immediately following a significant event.

Transducers and recorders could be engineered and tested to survive for many years of routine data collection on a ship and could be tested and, if necessary, replaced, as part of the ships routine maintenance.

The complete sensor suite requirements would be determined by the range of intended purposes. Including the sensors required to provide continuous monitoring of the health of the ships structure, major subsystems, selected component mounts, etc., that would be read out at maintenance time, as presently done for many modern automobiles, would add a very small fraction to the cost of a ship. However, it could conceivably reduce the cost of routine maintenance and lead to design improvements. Catching an incipient electrical failure in a weapon system before a ship goes into battle could conceivably save a ship. If suitably designed, it may also be quite feasible and cost-effective to retrofit existing ships that must operate in harms way with such a monitoring and data-logging system. The installation might be simple and inexpensive enough to be done during a routine re-fitting port call.

When the above was originally suggested, we were unaware that the ONR has had issued calls for SBIR proposals for ship instrumentation, especially as part of the Navys Condition-Based Maintenance initiative, and that some of these have been funded as far as the Phase II stage of SBIR programs. We hope that valuable results will accrue and encourage the Navy to move forward with significant installations on ships as proof tests.

10 THE USE OF FULL SHIP SHOCK TRIALS FOR VALIDATION

In addition to continuous monitoring of ships, it is critical to use FSST's– both of the recent past, and those of the future– to assess how well M&S can perform in predicting critical failures. Figure 18 shows the shock trials that have occurred since the Navy started carrying them out in the 1940's: *there has never been a single shock trial where* M&S *has tried to predict, or a posteriori explain the failures that occurred.* Without such studies, it is impossible to determine the extent to which the failures during shock trials are predictable at the present time, when the Navy codes are properly validated. We see several possibilities:

1. Critical failures in FSST's could be caused by equipment that was improperly qualified in component tests, due to inappropriate testing environments. This is exactly the type of failure that the M&S capability should be able to predict.

2. Critical failures could occur because of mistakes in workmanship.

3. Critical failures could occur because of systems level issues that go beyond the individual components themselves.

Without some understanding of which of these possibilities actually occurs during an FSST, it is impossible to assess how well M&S will mitigate the risks of eliminating FSST. If most of the failures turned out to be in the second category, the fix would be to maintain higher standards for inspection of workmanship. If it turned out to be in the third category, it would be necessary to understand the failures and devise methods for predicting them.

The DDG 53 and DDG 81 FSST summary reports provided to JASON made clear that:

Figure 18: List of shock trials that have occurred since the Navy started the program. (Courtesy of Mike Winnette)

- *There are important lessons learned from the FSST.* There are both systems-level and component-level failures observed in the FSST. These failures were neither mitigated nor predicted by prior component testing or simulation efforts. In some cases, the failure or degradation observed led to a recommendation of design review for a mission-critical shipboard system.

- *A few component failures can have significant consequence.* A component shock-qualification procedure which ensures the survivability of 99% of the critical components still is not good enough to ensure a ship's continued operational capability in the aftermath of a nearby underwater explosion. Oftentimes, the restoration time is measured in hours.

- *The FSST test operators do not always trust MIL-S-901 certification.* Unexpected and unexplained component failures in both tests were linked to inadequate/inappropriate testing during shock-qualification. Furthermore, additional data collection was carried out in the vicinity of especially critical equipment, in order to understand better the shock environment and, eventually, design a more appropriate component test.

Despite this important insight, the FSST documentation was found to be lacking in critical areas. If the FSST procedure is ever to be eliminated or replaced, it is necessary to understand why components still fail in the FSST, despite component testing. Furthermore, additional information is still needed before one can make an accurate assessment as to what extent these failures might be mitigated by modeling and simulation.

The summary reports, which would ideally be turned over to M&S analysts upon completion, require a detailed description of those shock-qualified components (Grade A and B) which failed during the FSST. This description should include:

- The component test used to qualify the component (e.g. LWSM, MWSM, FSP, DDAM), as well as a pointer to that test report.

- The component's shipboard mounting configuration, as it compared to the component test mounting configuration.
- The component's location on the ship.
- Any data collected in the vicinity of the failed component.
- The potential and observed consequences of the component failure.

In the reports reviewed by JASON, we could not locate even an explicit list of the Grade A and B components which failed or degraded during the FSST. Furthermore, failures and anomalies reported were generally not explicitly linked to the location where they occurred. The above information would be extremely helpful in answering those questions posed by the sponsor of this study. For those tests recently conducted (e.g. DDG 53 and 81), it seems reasonable that this data could still be compiled. Such an effort would be of great service to the Navy, and its M&S analysts in particular.

The FSST reports we reviewed did, however, provide extensive sensor data collected during the test. An abbreviated examination of this data indicates that it is adequate in quantity and nature to support validation of the structural code (DYSMAS) in development, though more data is always helpful. In the future, structural simulations carried out in advance of the FSST or FSST alternative test will inevitably indicate components which were not properly shock-qualified, based on their location on the ship. For instance, a simulation may report that a particular deck has a fundamental mode of 10Hz, but the components to be placed on it were qualified in a test with a simulated deck that has a 14Hz fundamental mode. In all such instances, we suggest enhanced data collection in the vicinity of the component. In the event that the component fails during the FSST, this additional data can be used to help understand the fragility boundaries of that particular component. Whether or not the component fails, this data can be used to verify those M&S predictions which may have the most significant consequence.

11 AIR-GUNS FOR SHIP TESTS

Can FSST be replaced with alternative tests? We have suggested above the utility of combining a validated M&S capability with monitoring of ship vibrations. Whether these tests alone will be sufficient for M&S to mitigate the risks currently borne by FSST can only be determined by identifying the failure modes during an FSST, and understanding what fraction of the failure modes are predictable by M&S.

It seems entirely plausible that even under the perhaps over-optimistic scenario that M&S can predict most of the failure modes it would still be useful to subject ships to localized explosive sources that mimic actual explosions. For example, it might turn out that certain system failures observed in FSST occur because of the excitation of a low frequency mode of the ship that is best stimulated by a local explosion.

In this regard, we were asked whether air-guns, the standard source of sound pulses for underwater seismic surveying, could be a useful substitute for high explosives in ship tests. A typical air-gun used for seismic work has a working volume of 0.3 liter filled with air at 100 bar pressure. Models with larger volume and higher pressure are available. The standard model produces a pressure pulse with peak pressure $P = 0.5$ bar at distance $D = 100$ meters, with pulse duration $\tau = 2 \times 10^{-3}$ seconds and peak frequency f=500 Hertz. A comparable amount of energy is emitted later by oscillations of the air-bubble with frequency around 20 Hertz. For ship testing, only the prompt pressure-pulse is relevant.

The three parameters that characterize an air gun are D, the distance from source to target, V, the working volume of the air-gun, and P_0, the working pressure. The scaling laws for the pressure pulse at the target are

$$\text{Peak pressure,} \qquad P \sim P_0 V^{1/3} D^{-1}.$$

$$\text{Pulse duration,} \qquad \tau \sim P_0^{-1/2} V^{1/3}$$

$$\text{Impulse,} \qquad P\tau \sim P_0^{1/2} V^{2/3} D^{-1}.$$

$$\text{Peak frequency.} \qquad f \sim P_0^{1/2} V^{-1/3}.$$

For a meaningful test of a ship we would need to fire a large array of air-guns simultaneously, but the use of an array would not significantly increase P or decrease τ. The basic weakness of air-guns is the fact that they give too small P and too large τ. The best we can do to make P large is to put the air-guns close to the ship so that $P = P_0$. The best we can do to make τ small is to take $P_0 = 1000$ bars, $V = 0.3$ liter, which gives $\tau = 6 \times 10^{-4}$ sec. Since the velocity of sound in water is 1500 meters/sec, the pressure pulse would then have an effective wave-length of 1 meter, too large to deliver any damaging shock to the ship.

Compared with chemical explosives which have $P_0 \sim 4 \times 10^4$ bars, the standard air-gun has P_0 smaller by a factor 400, τ larger by a factor of 20. That is the basic reason why air-guns cannot realistically simulate the effects of explosives.

It would be possible to use arrays of air-guns to deliver impulses to the ship, and then observe the response of the ship structures with accelerometers. This might be a convenient way to explore the normal modes of vibration of the ship and its contents, and to measure the damping of the various modes. But the air-guns could only excite the modes within the range where their amplitudes are linear. Air-guns are too feeble to explore the non-linear range where serious damage to the ship and its contents may occur.

12 DEPARTMENT OF ENERGY ROLES

We were asked to comment on the potential role of the Department of Energy in achieving the goal of FSST replacement through improved M&S capability. We were briefed by Sandia National Laboratory about their M&S capabilities, with a focus on (1) QMU, quantifying margins and uncertainties; and (2) component fragility and qualification.

The DOE has made significant investment in structural analysis software, for qualifying parts that are used in nuclear weapons, and for simulating material response in response to explosions. They have significant capability, including the ability to geometrically model complex structures, with very large finite element or finite difference meshes. They have significant capability for optimizing these codes to run on high performance computing platforms.

As outlined in detail above, the modeling of the response of ships to explosions is an extremely complex problem, with much discipline significant knowledge required. Department of Energy capability could be fruitfully drawn upon to help sort out specific issues that are uncovered during the validation of current M&S capability (represented by the DYSMAS code). For example:

- Validation might uncover that current Navy simulations (with $\approx 10^6$ degrees of freedom) are insufficient to reproduce ship vibrations with the required accuracy, in which case the DOE has expertise, experience, and computing resources for implementing larger calculations effectively.
- Validation might demonstrate that the damping of ship modes is not even approximately described by a linear combination of the mass and stiffness matrices. DOE expertise could help sort out computational methodologies for dealing with more general damping matrices.

- Other specific problems could also arise where DOE expertise could be valuable.

However we do not think it is advisable to outsource *either* the M&S of Naval architectures, *or* the V&V (verification and validation) of the codes. The M&S of surface ships and large naval equipment include a number of aspects that are discipline specific. We believe it is unlikely that analysts with other expertise can contribute significantly to the creation or validation M&S program without substantial input from naval architects, scientists and engineers.

With respect to validation, the methods the DOE has developed are quite useful for their problems. However the full ship shock problem is so complicated – involving large numbers of Grade A components that must be qualified against failure in a ship-environment specific manner – that the DOE approach is not as useful and could actually divert attention from the real problem of determining how good a simulation is good enough.

On the other hand we emphasize that the DOE contains substantial technical expertise that can and should be drawn upon for specific technical challenges that arise during validation and code improvement.

13 SUMMARY AND RECOMMENDATIONS

The basic messages that emerge from this study is that there is substantial opportunity for the Navy to use M&S capability to better carry out its mission, both in designing and operating ships, and in improving the ability to certify that these ships are secure against ever changing threats. There is reason to believe that a validated M&S capability could lead to a situation where the risk currently mitigated by FSST could be entirely taken on by M&S.

However, the current Navy codes have not yet been validated to the level they need to be for these opportunities to be fully exploited; correspondingly there is much skepticism in the Navy over their use. Our specific recommendations are as follows:

1. We recommend validation of the Navy M&S predictions for elastic structural response (frequencies and damping).

2. We recommend that the Navy should instrument the lead ship to measure continuously the vibration modes and their associated dampings. Such tests should occur before FSST, in order to provide model validation before FSST predictions .

3. It needs to be determined how well present M&S capability can predict the failure modes of components in Full Ship Shock Trials. This can be done by (i) carrying out comparisons of simulations and observation of failure modes on future shock trials, and (ii) carrying out simulations on recent full ship shock trials. Successful prediction or understanding of the failure modes in the historical database is a substantial step forward in the code validation process.

4. Uncertainties in component testing procedures for testing to a given threat level must be better documented and understood. For example, the Navy's

validated M&S capability for liquid response should be used to determine whether the Keel Shock Factor is the right indicator of "similarity" between the shock induced by a hostile event and the impulse delivered in the component test program.

5. An analysis of the potential for the combination of continuous monitoring of ship vibrations with M&S to lead to cost savings in both the design of a ship to a fixed threat level, and the operation and maintenance of a ship should be carefully carried out and documented. This analysis should include the potential for cost savings in operations, and reducing the cost per ship through design improvements while still maintaining design margins for surviving realistic hazards.

6. DDAM should be updated using both experiments and M&S, and incorporating current ship requirements.

7. It is critical for the Navy to maintain its high quality of analysts in M&S.

A APPENDIX: Resolution, damping and energy transfer: some observations

In the report it was observed that the largest uncertainties in the structural simulations are:

1. Damping and dissipation. The DYSMAS simulations of the Lütjens test assume no damping. Perhaps this is the reason for the short time scale oscillations in the simulations that do not exist in the measurements.

2. Another uncertainty is in the finite element model itself. Unresolved components must be dealt with in the model in some way. However it is treated this leads to an error in the finite element model and the question is how large is this error.

3. Finally, there is the question of transferring energy from the resolved scales to the unresolved scales in the problem, such as joints and welds. This energy transfer is like dissipation in that it results in energy that is lost from the simulational degrees of freedom; but it is unlike dissipation in that the energy is not dissipated to heat and can in principle couple back to the simulated degrees of freedom later on.

Which of the uncertainties might be most significant for the simulations? As noted above, the calculations showed continuing oscillatory accelerations over ~ 10 cycles where the actual data did not. We ask if the origin of this discrepancy is inadequate resolution of the calculations, and if so how finely resolved the calculations must be to resolve the discrepancy. If not, then what is its origin, and how may it be resolved?

The finite element calculation of the Lütjens trial contains $\sim 10^6$ elements and nodes. A medium-sized ship may have about 6 deck levels over most of its length, about 6 compartments across its beam and perhaps 25 compartments along its length (these estimates come from assuming a keel-to-bridge height of 15 m, a beam of 18 m and a length of 125 m, with typical compartment dimensions of 2.5 m X 3 m X 5 m). The result is 900 compartments. If all edges have four adjoining compartments then there are three bulkhead or deck plates per compartment, or 2700 in all. Assuming half of the finite elements are used to resolve these plates (the other half being reserved for hatches, major structural members, or other features) then each plate is resolved into > 200 finite elements.

We conclude that the resolution is sufficient to calculate, with reasonable fidelity, the response of each plate (at least, it is not limited by the number of elements available). Inadequate resolution is not likely to be the origin of the failure to calculate the high frequency part of the response, and there is no reason to expect increasing the number of elements to improve agreement.

It may be that the source of the problem is inadequate treatment of equipment mounts and riveted joints (which respond nonlinearly even at low loads, as shown by the familiar creaking sounds indicative of stick-slip friction). However, the information required to improve this is either unavailable (these are not well understood) or not feasibly incorporated in the calculation because of the large number of such features, each different and ill-characterized, in each ship. Hence a statistical treatment is necessary.

The codes use an artificial damping to represent the effects of modes too crudely resolved to treat accurately. However, this is not correct; coupling of energy to poorly resolved modes is reversible, rather than genuinely dissipative.

If coupling of the higher frequency resolved modes (those that remain at significant amplitude in the calculations but apparently not in the data) to poorly resolved modes is rapid, then energy will be drawn from these resolved modes and their amplitude will be small, as observed but not as found in the present calculations. We therefore suggest that this may be the case, and suggest a method for modeling this effect in the finite element calculations that conserves energy in a statistical sense but does not require an unphysically large microscopic damping into heat.

At each node couple a fraction of the kinetic energy (in each time interval, analogously to damping) to a variable representing high order modes. This is done by introducing an acceleration attributable to coupling to these modes:

$$\vec{a}_{ho} = -\frac{b\vec{v}}{m_{node}}, \tag{A-13}$$

where the damping parameter b, which must be estimated by comparing the results of finite element calculations to actual data, represents coupling to high order unresolved modes rather than actual friction. Rather than removing that energy from the calculation, add it to a variable representing the short-wavelength kinetic energy of the mass associated with that node:

$$\frac{dE_{node}}{dt} = bv^2 - \gamma E_{node}. \tag{A-14}$$

The second term allows for damping of these modes, estimated from the empirical damping rates of oscillations in structural materials. This would be in addition to the physical damping assumed for the explicitly calculated nodal velocity.

From E_{node} it is possible to define a pseudo-temperature:

$$\frac{3}{2}kT_{node} = E_{node} \tag{A-15}$$

and a pseudo-thermal velocity

$$v_{ps-th} = \sqrt{\frac{3kT_{node}}{m_{node}}}. \tag{A-16}$$

Add to the acceleration of each node another term representing coupling to the velocity drawn randomly from a pseudo-Maxwellian distribution at the pseudo-temperature:

$$\vec{a}_{coup} = \hat{n}\frac{bv_{ps-th}}{m_{node}}, \tag{A-17}$$

where the unit vector $\hat{n}$ is in a random direction, chosen isotropically. Both a_{ho} and a_{coup} are added to the accelerations found explicitly by the finite element calculation

The coupling rates must be estimated semi-empirically. If well chosen, this model can accurately represent the flow of energy into modes of too short wavelength to be calculated explicitly, or unable to be calculated explicitly because they depend on poorly understood or unknown quantities such as the properties of individual equipment mounts. It cannot represent the flow of such energy in space or in wave-vector space (among these unresolved modes), although it would be possible to assume sharing of the kinetic energy among the nodes, taking equilibrium of the pseudo-temperature, to represent this. Short-wavelength modes of elastic bodies typically have group velocities much greater than those of long-wavelength modes (though the situation may be more complex for complex structures), so rapid sharing is not necessarily valid.

Because the number of nodes is large, the resultant velocities and accelerations will represent the influence of many such random short-wavelength mode variables, and may average to a nearly constant value, without removing their kinetic energy from the calculation in an unphysical manner. If so, this appears to offer hope of improving the agreement between measurement and the calculated finite element model.

B APPENDIX: General Remarks About Hydrocode Simulation

While DDAM is primarily a tool for investigating shock response of individual components or systems of components, full ship response to underwater explosion (UNDEX) events is generally carried out with what is known as a *hydrocode*. Mair (1999) defines a hydrocode as a "computational continuum mechanics code that simulates the response of both solids and fluids under such highly dynamic conditions that shock wave propagation is a dominant feature." Hyrdocodes have the capability to predict the complex interaction of fluid, structure, and explosive energy, enabling the simulation of ship response to both distant and close-in underwater explosions.

Most hydrocodes are built upon the finite element method; hydrocodes are the subset of finite element codes which are applicable to the UNDEX problem. A general finite element simulation begins with a geometric model of the materials and structures of interest. The level of fidelity required in the geometric model is dictated by both the problem and the solution accuracy expected; in some cases, it is not known a priori (e.g., when is it ok to represent a component as a lumped mass?). The materials and structures in this geometric model are then discretized into area of volumetric elements – "finite elements." The set of elements, defined on their corners by nodes and on their surfaces by edges and faces, is known as the mesh. A hydrocode simulation of an UNDEX event may require as many as 1,000,000 elements in the mesh, as dictated by the time and spatial scales of interest. This mesh is then augmented with mathematical models of the material behaviors, material and structure interaction laws, and boundary conditions. Again, varying levels of fidelity may be required in these mathematical models, and the appropriate level may not be known a priori. Finally, the simulation is triggered with a set of initial conditions which, again, are a model of the true event of interest. During the simulation, element-level forces and displacements are calculated by maintaining continuity and equilibrium between adjacent elements. The gross response of the structure is the product of the these forces and displacements accumulating and interacting over time.

At this relative early stage of hydrocode development, the integrity of any given simulation is highly dependent upon the skill of the analyst who generates it. Numerous simplifying assumptions are required in order to convert a real event into a tractable mathematical model, especially in consideration of limitations on computational power. In general, standard meshes, material models, boundary conditions, etc. do not exist. The analyst must not only understand the mathematical framework of the hydrocode itself, but also the specific problem of interest. For this reason, a team of experts adept at simulating nuclear weapons behaviors would not be able to generate high-quality simulations of UNDEX events without significant assistance from the Navy (for example).

It is important to understand that not all hydrocode tools are alike, either. Numerous *classes* of hydrocodes exist, each with a unique set of inherent assumptions, capabilities, and limitations. Within each class of code there is also variability, but this level of variability should be thought of as second-order in evaluating a code for a particular problem (e.g UNDEX). Once the toolset is advanced for one code of the class, the fix or enhancement can easily be implemented in any other; limitations inherent in the class of code, however, generally can not be overcome. An overview and evaluation of the relevant classes of hydrocodes for the UNDEX problem is given below. These classes are distinguished primarily by the way in which material displacement is tracked by the mesh.

Lagrangian. In a Lagrangian code, the mesh remains fixed on the material. No material is allowed to flux across element boundaries – the mesh will distort to accommodate material distortions. Since individual element integration schemes require some level of geometric integrity to the element, Lagrangian codes generally break down under excessive material distortion. These integration schemes may either be implicit or explicit; explicit codes are usually more appropriate for dynamic problems. One current area of contention in Lagrangian finite element methods is the treatment of material failure, which has implications for both hull deformation predictions of cavitation. Due to the limitations on material distortions, Lagrangian methods are generally only applied to structural problems - not those including fluids. Current areas of development, including more complex nonlinear material behaviors and implicit dynamics, are unlikely to improve capabilities for the prediction of linear ship shock response. Nonetheless, these developments may aid in the prediction of individual component failures, given a known excitation function.

Eulerian. In an Eulerian code, the mesh remains fixed in space and material fluxes across the element boundaries. At each time step, there are two numerical steps: a Lagrangian calculation of forces/displacements, followed by a mesh remap (advection). This second advection step makes Eulerian codes in general more computationally intensive than Lagrangian codes. Because the material moves independent of the mesh, it is impossible to implement material models which include history-dependence, such as damage evolution or complex plasticity. The Eulerian method also requires properties to be constant across an element, so gradients in stress and/or displacement over a thickness can only be captured with a very fine mesh structure. For these reasons, the Eulerian method is not usually appropriate for simulating structural response. It is, however, excellent for simulating fluid motion with large material distortions.

Coupled Eulerian Lagrangian (CEL). In a Coupled Eulerian Lagrangian (CEL) code, the mesh is divided into domains which are pre-assigned to follow either the Lagrangian or Eulerian method. At the boundary, the Lagrangian domain overlaps the Eulerian domain. This method is especially useful for simulations of structures and fluids interacting, where the structure behavior is calculated by the Lagrangian method and the fluid behavior is calculated with the Eulerian method. Thus, CEL codes are very well-suited for the UNDEX problem. Development of the CEL code is usually an ad-hoc linkage of existing Lagrangian and Eulerian codes, and in this regard it is modular and adaptable. This construct also enables the CEL codes to advance with the state-of-the-art in both Lagrangian and Eulerian methods.

Arbitrary Lagrangian Eulerian (ALE). The Arbitrary Lagrangian Eulerian (ALE) method is an intermediary between the Lagrangian and Eulerian methods. The simulation begins with a regular, "unstructured" mesh. Like the Eulerian method, there are two numerical steps at each time step: Lagrangian motion calculation, and mesh remap. In this case, the mesh does not always remap to its original configuration. The remap might do nothing, leaving the mesh to remain as is (Lagrangian), or it may change to a different, more optimal configuration, based on prescribed weighting functions. Thus the remap occurs only when and where it is needed, and only to the degree required to establish a stable mesh. In this way, its computational expense lies somewhere between Lagrangian and Eulerian. ALE codes are a relatively new development, as compared to Lagrangian or Eulerian, and at this point not every code has capability to simulate more than one material. Furthermore, because of the allowance for mesh remapping, material models will not be able to have complex history-dependent internal variables.

Lagrangian + USA. While not a class of hydrocodes per se, this combination of numerical techniques has been applied numerous times in the simulation of ship response to an UNDEX event. In this method, the structure (ship) is modeled with a Lagrangian mesh; the surrounding fluid is modeled with boundary surface elements covering the structure's entire wet surface. The USA code is the predominant code for describing the boundary elements, implementing the analytical Doubly Asymptotic Approximation. This technique leverages off of the fact that the pressure pulse emanating from a far-field underwater explosion has been well characterized. Rather than simulate the entire underwater explosion, this technique allows one to simply model its primary effects as a boundary condition. This method inherently can not capture the effects of close-in explosions or bubble pulse loading. Furthermore, modification of the technique is required in order to capture cavitation effects due to hull acceleration. Nonetheless, for the simulation of the effects of a remote detonation, the Lagrangian + USA method offers an alternative with generous computational savings.

For practical reasons, the CEL, ALE, and Lagrangian+USA methods are the only ones which have thus far been used in the simulation of surface ship response to an UNDEX event. All of these instances have come within the last ten years, and still they are limited. The Navy has invested significant resources in a code of the CEL type: DYSMAS. While a Lagrangian+USA code may provide an adequate – and faster – solution for certain problems, the CEL-type allows flexibility to simulate both close-in and remote explosive events. Furthermore, as discussed earlier, it is modular and likely to advance with the state-of-the-art in both Eulerian and Lagrangian methods.

Recently, two commercial codes — MSC.Dytran and ABAQUS/Simulia – have introduced packages with capability for *all* of these methods (CEL, ALE, and Lagrangian+USA). These general packages allow a user to choose the tool which is best matched for a given problem. The standard interface reduces training time for new users. In the future, if the Navy finds that additional toolsets are needed, or that DYSMAS development can not keep up with advances in the state-of-the-art, there may be financial incentive to switch to one of these well-respected commercial codes. At the present, however, there is no apparent capability gap between the Navy's codes and available commercial codes. None of the codes – including DYSMAS – have been properly validated against the data of an FSST.

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