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# Comparison of stress predictions for ship onboard equipment shock response using dynamic design analysis method and transient finite element method

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## ABSTRACT

Modelling the structural response to underwater explosions (UNDEX) remains one of the most demanding fields of simulation engineering due to the limited feasibility of experimental validation. To address this, the present study compares two approaches for assessing the shock resistance of naval structures: the dynamic design analysis method (DDAM) and transient non-linear finite element method (FEM) simulations. The objective was to evaluate the accuracy and applicability of DDAM as a preliminary design tool by benchmarking it against detailed FEM results. Original DDAM formulations were consistently reformulated into the SI unit system to enable a direct comparison. For the investigated tank–frame structure, the maximum effective stress predicted by DDAM was 258.3 MPa, whereas FEM analysis yielded 292.1 MPa, corresponding to a relative difference of 11.6%. These findings confirm that DDAM provides a conservative yet reasonably accurate estimate of global shock response within engineering tolerance, while FEM captures localized peaks and non-linear effects at critical joints. Thus, DDAM proves to be highly efficient for rapid preliminary assessments, whereas FEM ensures reliability in detailed evaluation of complex structural behaviour under UNDEX loading.

**Keywords:** underwater explosion, dynamic design analysis method, finite element method, shock response, naval structures.

## INTRODUCTION

Shock resistance of shipboard equipment is essential, as naval vessels often face underwater explosions generating high-intensity shock waves. These waves induce severe dynamic loads on the ship's structure and onboard systems. To maintain operational capability and safety, equipment must be designed and qualified to withstand such shocks.

Shock qualification standards for naval ships often reference the Shock Resistance Analysis of Equipment for Surface Ships (STANAG 4142), which remains classified. However, under general circumstances, other publicly available standards are commonly applied. These include the Lloyd's Register Naval Ship Rules (LRNSR) [1] and regulations issued by Det Norske Veritas (DNV) [2], which provide guidelines for increasing allowable material stresses as a function of strain rate.

Complementary methodologies for naval vessels are also outlined in the Naval Sea Systems Command (NAVSEA 0908-LP-000-3010A) framework [3], which includes elastic-plastic shock design values and the application of the Dynamic Design Analysis Method (DDAM), specifically tailored to meet the requirements of the U.S. Navy.

To evaluate the structural integrity and functionality of onboard equipment under underwater explosion (UNDEX) loading, several analytical and numerical approaches are employed [4–7]. These differ in terms of fidelity, computational cost, and applicability to specific equipment types. The primary approaches include:

- quasi-static finite element analysis (FEA) [8, 9], which is suitable for relatively rigid equipment,
- design response spectra (DRS) and the dynamic design analysis method (DDAM) [10, 11], both widely applied in evaluating dynamic shock responses,
- dynamic or nonlinear structural FEA [12], applicable to complex systems exhibiting significant nonlinearity,
- advanced numerical methods [13], such as fluid structure interaction (FSI) techniques, which allow for detailed modelling of coupled domains using arbitrary Lagrangian-Eulerian (ALE) or structural arbitrary Lagrangian-Eulerian (S-ALE) based solvers.

Each of these approaches offers unique advantages and limitations, and in practice, they are often used in combination to achieve reliable and comprehensive shock qualification results.

The primary objective of this study is to evaluate the shock resistance of ship structures, with particular emphasis on the stress distribution and load-bearing capacity of structural elements such as frames and supports. While existing manuals and classification society standards provide general guidelines for shock qualification, they often lack detailed methodologies for optimizing shock attenuation, especially in terms of damper selection and design. This study addresses that gap by presenting and comparing the results of two numerical methods, offering practical insight for engineers involved in designing and assessing naval equipment. By integrating advanced computational techniques with international standards, the findings contribute to a more comprehensive approach to mitigating shock effects and enhancing overall ship safety and performance. The choice of the two methods analysed is based on balancing computational accuracy and efficiency.

Fluid-structure interaction approaches are widely recognized as the most accurate tools for simulating shock responses due to their ability to capture coupled domain behaviour with high fidelity. However, they are also extremely demanding in terms of computational time and hardware resources, making them less practical for fast, iterative analyses during early design stages.

The classical transient FEM remains the most commonly used approach for dynamic analyses in ship shock qualification. Nevertheless, when applied to solid 3D models that include material nonlinearities and contact interactions, its computational performance significantly deteriorates. One way to alleviate this issue is by replacing solid elements with shell formulations, which considerably reduce the degrees of freedom, although at the cost of increased modelling effort.

An alternative, less popular but historically well-established method is the DDAM, which, despite being implemented in most commercial FEM solvers, is rarely employed in modern engineering practice. The likely reason for this is not its lack of capability, but the complexity of its application rules and result interpretation. A key advantage of DDAM lies in its frequency-domain formulation, which allows for rapid computations as opposed to the time-consuming transient simulations of classical FEM.

Demonstrating agreement between DDAM predictions and transient FEM results under realistic boundary and loading conditions would offer a novel contribution to the field of shock response analyses. It would validate DDAM as a reliable alternative for early-stage evaluation of naval equipment, providing a practical methodology that combines speed and sufficient accuracy—especially valuable in scenarios where rapid iteration and conservative estimations are needed.

## BACKGROUND

In recent years, studies on stresses and shock response of naval equipment have focused on three complementary directions. First, rapid spectral–modal methods are being developed for equipment mounted to the hull and foundations, allowing the estimation of forces and stresses without costly time-domain simulations, based on the results of modal analysis and design spectra [14]. Second, unsteady calculations with full fluid structure interaction and ALE methods are carried out, which serve to reproduce the wave-structure interaction and to update design spectra on the basis of data from ship trials [15].Third, results are published from large-scale shock platform tests with real equipment, e.g., naval engines, which provide reliable data for model validation and for selecting damping parameters and mounting methods [16].

Currently, spectral-modal methods have reached a high level of maturity and are widely applied in engineering practice, but their limitation remains their approximate nature and the inability to capture nonlinear local effects.

FSI/FEM simulations are developing rapidly and aim to increase the accuracy of wave–structure interaction modelling, yet they still face the barrier of very high computational costs and challenges in validation. Shock platform trials, on the other hand, provide unique experimental data, but their major drawback lies in their high cost and limited repeatability, which hinders the systematic updating of design spectra.

Of particular note is the dynamic structural mechanics analysis system (DYSMAS) [17], which has been continuously developed since the 1990s solely for the purpose of UNDEX analysis. The DYSMAS is currently the most advanced computational tool for analysing naval structures subjected to underwater explosions. Its main advantage lies in its specialization – the built-in fluid–structure interaction (Euler-Lagrange coupling) solver has been tailored specifically to capture shock wave propagation, cavitation, and gas bubble pulsation. As a result, DYSMAS simulations are more stable, computationally efficient, and physically accurate than those performed with general-purpose codes such as LS-DYNA. The system enables a detailed assessment of both the global hull response and local effects in critical structural components. However, its main drawbacks are its limited accessibility – DYSMAS is a NATO reference code, strictly controlled by the German and U.S. Navies – and the fact that, despite its efficiency, UNDEX simulations remain extremely demanding in terms of computational resources, often requiring high-performance computing facilities.

A clear trend in the literature is the combination of these approaches into hybrid computational–experimental schemes. Fast spectral–modal methods continue to be used at the early design stages, but they are complemented with local FSI/FEM simulations in critical structural regions. Data from shock platform trials are, in turn, employed for calibration and validation of design spectra, allowing the gradual reduction of numerical model uncertainties.

The direction of development is therefore the integration of methods within shared databases and unified design procedures, in order to preserve computational efficiency while better capturing nonlinear local effects and realistic operating conditions.

In light of these works, DDAM remains a computationally efficient tool at the preliminary stage, while FSI/FEM simulations provide references for verifying local stress maxima and nonlinear effects. DDAM is a modal method used to qualify the strength of supporting structures and equipment subjected to underwater explosions. Shock spectra are defined based on modal parameters and empirical data from shock tests.

Unlike direct shock analysis, DDAM uses the shock response spectrum (SRS) theory and the structure's own properties. Structural information provides participating modes, natural frequencies, and other modal properties for DDAM analyses. Finally, is computed via modal superposition [11]. DDAM has been widely used in naval engineering since the early 1950's, when it was applied by the American and British navies to evaluate shock responses of embarked equipment. Today, it remains a global reference for rapid and efficient shock design [12].

In DDAM, on-board equipment or structures are discretized in equivalent mass – spring systems (finite element meshes) subjected to a shock response spectrum, with the objective of calculating the equipment/structure in terms of displacements, velocity, and acceleration but also to determine the stress state inside the structures. DDAM method has been implemented in several finite element software such as NASTRAN and LS-DYNA.

As input, the acceleration shock response spectrum is usually obtained from on-board experimental tests and/or operational data records performed at different locations of a pattern vessel. The shock spectrum serves as a critical tool in naval design, allowing engineers to:

- predict dynamic responses of components across different modal frequencies,
- determine the required structural reinforcement based on modal characteristics,
- compare different configurations and select materials that meet shock resistance criteria,
- ensure compliance with DDAM standards, preventing underestimation of critical design accelerations.

The shock spectrum embodies the core principles of DDAM and serves as a practical framework for evaluating and optimizing the shock resilience of naval structures. The effective modal mass and support structure are determined through preliminary modal analyses.

DDAM is derived from the broader shock testing framework (Fig. 1) outlined in MIL-DTL-901 [18], which establishes the fundamental

**Fig. 1.** Typical MIL-DTL-901E floating shock platform setup [19, 20]

requirements for high-impact shock testing of shipboard equipment. However, MIL-DTL-901 primarily focuses on physical shock testing procedures and does not provide a direct computational method for assessing dynamic responses analytically. Instead, numerical shock analyses and design criteria are governed by NAVSEA standards.

The primary reference for DDAM implementation is [21], which provides guidelines for evaluating shock resistance through modal analyses. Its updated version, T9070-AJ-DPC-120/3010 [3], refines these guidelines but does not explicitly define the formulas for the fundamental shock parameters (Eq. 1 and Eq. 2).

To determine these dynamic parameters, engineers refer to DDS 072-1 [18], which contains the official formulas for design velocity and acceleration values ($V_0$ and $A_0$). However, this document is classified and not publicly accessible. As a result, engineering applications often rely on alternative references such as NR 1396 [22], which provides practical formulations derived from prior validated methodologies.

Additional literature further supplements these foundational documents by exploring the ship shock response, underwater explosion effects, and computational modelling techniques. Notable references included by Reid [23], which compiles findings from the Underwater Explosions Research Department (UERD) at the Naval Surface Warfare Centre, including ship shock trials and numerical modelling approaches. Didoszak [9] presents modern computational techniques for shock failure evaluation, while Tasdelen [8] compares different numerical strategies for evaluating the effects of underwater explosions on shipboard structures.

The DDAM methodology and its numerical implementation are summarized in Figure 2. The diagram illustrates the hierarchy of empirical standards (MIL and NAVSEA), formal procedures (DDS), and numerical implementations (e.g., LS-DYNA DDAM module) that together form the analytical backbone of the DDAM approach.

## OBJECT AND OUTCOME OF THE STUDY

The marine tank (Fig. 3) mounted on a support frame is a crucial component in various types

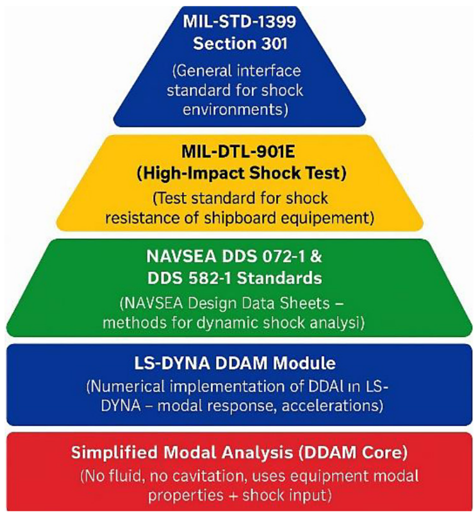


**Fig. 2.** Summary of the DDAM framework – from MIL-DTL-901 and MIL-STD-1399 to numerical implementation in LS-DYNA

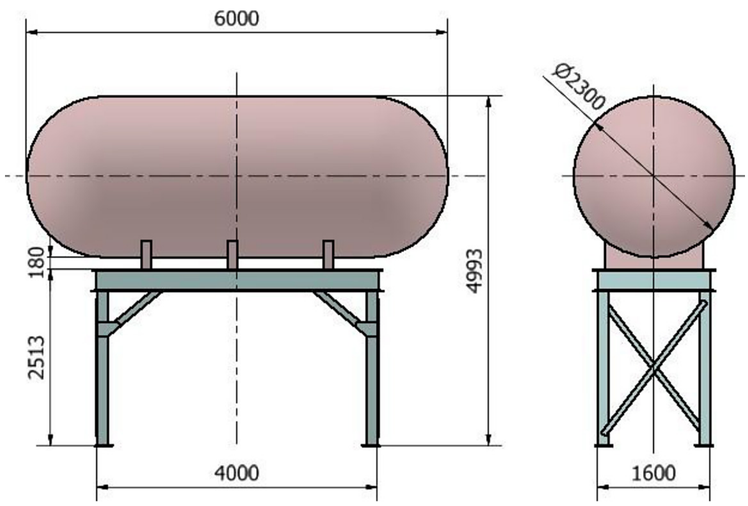


**Fig. 3.** Marine tank CAD model mounted on a support frame for shock resistance analysis under UNDEX conditions (dimensions in millimetres)

of vessels, serving purposes such as storing fuel, water, or other technical media. Its resilience to dynamic loads, such as impulsive accelerations, is particularly significant in the context of maritime operations, where forces generated by underwater explosions, shock loads may occur.

## Finite element model description

### *Dimensions*

The FE model of the marine tank system was developed in LS-DYNA using shell elements and for the support frame. The average element size was 25 mm, refined in the support connections. The tank dimensions of 6000 mm in length and 2300 mm in diameter, with a wall thickness of 30 mm is suitable for storing liquid or pressurized gases. The frame structure consisting of a lattice design with cross-bracings, constructed from 8 mm thick structural sections, which enhances rigidity and minimizes localized deformation under dynamic loads.

### *Materials*

The FEM model incorporates real material properties of the tank and frame steel (Fig. 4), including stress-strain behaviour and damping capacity, critical for accuracy and reliability.

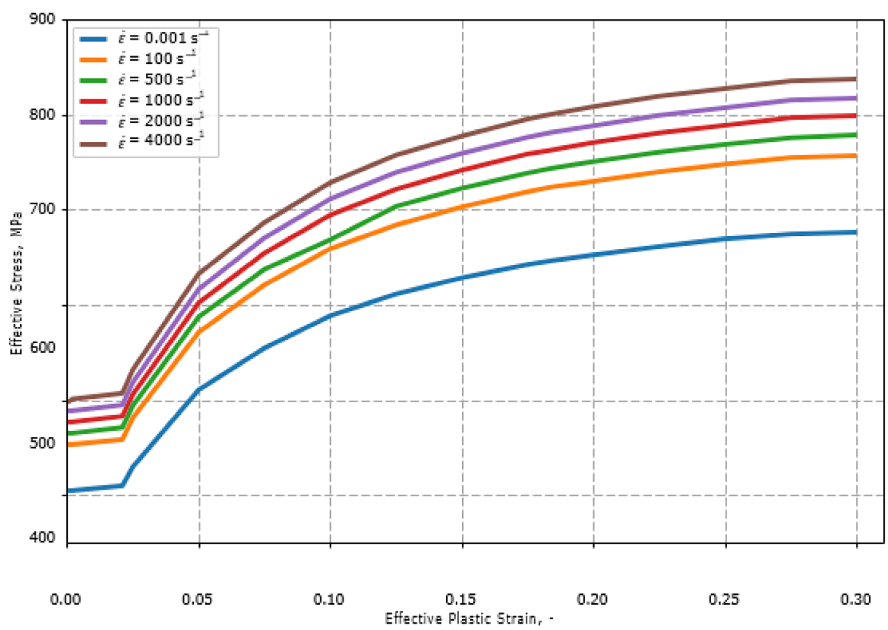


**Fig. 4.** Plastic behaviour $\sigma_{true} - \varepsilon_{plasticity}$

For the transient FEM analyses, a high-strength structural steel with a tensile strength of $R_m$ = 647 MPa and yield strength of $R_e$ = 405 MPa was adopted. To account for the dynamic nature of the loading conditions, a rate-dependent plasticity model based on piecewise linear flow stress curves was used (*MAT PIECEWISE LINEAR PLASTICITY). The material response to different plastic strain rates was introduced through a tabulated input (*DEFINE_TABLE), allowing for a more flexible and physically representative description of strain rate effects. Representative data for this material, incorporating different plastic strain rates, are shown in Figure 4 and were used in a demonstrative context to ensure the consistency of results under rapid loading.

The use of tabulated data in constitutive modelling, eliminates the need to identify empirical constants required by traditional rate-dependent models such as Cowper-Symonds or Johnson-Cook. When reliable experimental or standard tabular data for flow stress at various strain rates is available, modern FEM techniques recommend direct implementation of such datasets. This approach improves both the physical realism and numerical robustness of simulations, particularly under dynamic loading conditions.

### *Boundary conditions*

The tank was rigidly attached to the frame using CONTACT_TIED_SURFACE_TO_ SURFACE. The frame supports were constrained to the foundation with BOUNDARY_SPC constraints in all translational and rotational degrees of freedom, representing welded connections to the base structure.

In contrast, the DDAM methodology inherently assumes simplified material behaviour. When the "elasto-plastic" option (MATTYP=2) is selected, LS-DYNA internally applies an idealized bilinear material model with default values for the yield strength and elastic modulus. These are not derived from any user-defined *MAT card and do not consider strain rate sensitivity.

### *Kinematic loading conditions*

The study examines the structural behaviour under kinematic loading conditions. The maximum accelerations applied to the system are 12.5 g. These loads are applied as boundary conditions to the legs of the frame supporting the tank. The gravitational effects and inertia forces generated by the tank and its contents are included to simulate realistic operational conditions.

To capture realistic loading scenarios, kinematic excitations were defined as time-dependent functions. The simulation time is set to allow for several cycles of excitation to achieve peak stress and deformation responses, reflecting the real-world performance of the structure under impulsive loads. The analyses include a single shock impulse.

The core analyses use the quasi-static FEM method, a robust approach for modelling dynamic interactions between loads and structures. Additionally, the DDAM method was evaluated to provide a comparative perspective, analysing their advantages and limitations in terms of computational efficiency and result accuracy. This multi-method approach enhances the understanding of the structural performance of the marine tank and frame, validating their applicability for high-stress maritime environments and providing a foundation for further optimisation.

The evaluation assesses the shock resistance of the foundation and load-bearing frame, with simulations carried out in accordance with the standards outlined in Det Norske Veritas (DNV) and Lloyd's Register Naval Ship Rules (LRNSR).

## DDAM ANALYSIS

Based on the DDAM methodology, different analyses variants can be applied depending on the ship type, material behaviour, and loading conditions (Fig. 5). The key parameters considered in DDAM analyses are:
- shock spectrum, NRL-1396 standard navy defined shock spectrum and user defined spectrum based on custom conditions,
- ship type, submarine, surface ship,
- mounting type, hull-mounted equipment, deck-mounted equipment, shell plating-mounted equipment,
- material type, elastic (linear material behaviour), elasto-plastic (considering plastic deformations),
- load direction, vertical, athwartship (side-to-side motion), fore and aft (longitudinal motion).

These variants allow for customization of the DDAM analyses based on the specific operational environment of the naval vessel and the type of structural components being evaluated. The selection of the appropriate parameters ensures that

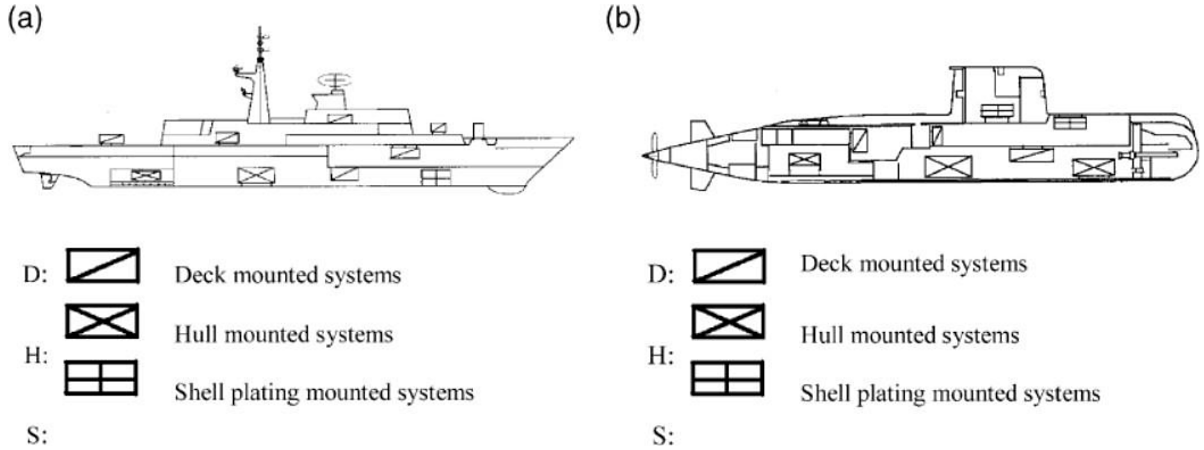


**Fig. 5.** Mounted systems of equipment and foundation [22]: a) surface ship, b) submarine

the shock qualification process accounts for realistic loading conditions and structural constraints.

## Methodology behind DDAM equations

The DDAM equations provide an efficient and validated approach for computing structural stresses caused by shock loading. By leveraging modal decomposition and response spectra, they eliminate the need for full transient simulations while maintaining sufficient accuracy for naval system qualification.

DDAM combines empirical data from historical ship shock tests with theoretical modal analyses. The core idea is to assess how different structural modes respond to a given shock spectrum, allowing estimation of shock-induced stresses and deformations without time-consuming simulations. A key parameter in this method is the modal acceleration, $A_0$, which quantifies how a given mode experiences acceleration under shock loading.

For example, for submarines and hull-mounted systems, the empirical equations [22] are:

- Modal acceleration $A_0$ – derived from empirical data, ensures that acceleration is correctly scaled for various structural masses based on real-world shock test results:

$$A_0 = 10.4 \cdot \frac{480+\overline{w}_a}{20+\overline{w}_a} \cdot g, \text{where } g = 32.174, \text{ ft/s}^2 \quad (1)$$

where: $A_0$ – modal acceleration, g,
$V_0$ – modal velocity, in/s.
$\overline{w}_a = w_a/1000$ – effective modal weight, kips,
$w_a = m_{eff} \cdot g$ – effective modal weight in pounds.

- Modal velocity $V_0$ – provides an estimate of how kinetic energy is distributed in a given mode:

$$V_0 = 20 \cdot \frac{480+\overline{w}_a}{100+\overline{w}_a}, \text{ in/s}^2 \quad (2)$$

- Shock design acceleration $D_a$ – represents the expected acceleration a structure must endure under shock loading. This value reflects modal characteristics and is essential for structural evaluation. The actual acceleration demand imposed on the structure is calculated as:

$$D_a = \min(A_0 \cdot g,\ V_0 \cdot \omega) \quad (3)$$

This ensures that both acceleration-dominated and velocity-dominated responses are considered. Note that in SI-based versions of DDAM, is not multiplied by $g$ (Eq. (6)).

- Shock design value *SDV* – the structural design is ultimately based on the worst-case acceleration demand:

$$SDV = \max(D_a, 6g) \quad (4)$$

The $6g$ threshold guarantees a baseline level of robustness, consistent with naval shock test standards. The SDV is the benchmark acceleration used in structural design and verification. In practice, it is often equal to $D_a$:

$$SDV = D_a \quad (5)$$

The SDV serves as a reference for verifying stress levels and ensuring compliance with standards such as DNV or MIL-STD and can be sum up as:

- defines the required acceleration capacity of a structure,
- provides a standardized reference for engineering assessments,
- both are essential for ensuring naval and offshore structures withstand shock loads effectively.

The DDAM methodology is rooted in extensive shock testing by the U.S. Navy, where

acceleration, velocity, and deformation were recorded during controlled experiments on real ships. The constants in the equations (e.g., 480, 20, 100, 101.99, 0.508) were statistically derived to generalize results across diverse naval structures and is based on three key principles:
- empirical validation, equations are calibrated using real-world shock response data,
- modal response approximation, structural behaviour is modelled using a superposition of natural modes, simplifying transient analysis,
- standardized safety factors, a minimum requirement of 6g ensures resilience even in lightweight systems.

## Reference equations and conversion to SI units

The original DDAM reference equations Eq. (1) and Eq. (2) are given in imperial units and below is to the SI unit system.

To convert the equations to SI units, the following adjustments are made: replace g with 9.81 m/s$^2$, and replace the constants 10.4 and 20 with their scaled SI equivalents with 1 in/sec = 0.0254 s as 10.4·9.81 = 101.99 m/s$^2$ and 20 in/sec = 20·0.0254 = 0.508 m/s, The converted equations in SI units now look like:
- modal acceleration

$$A_0 = 101.99 \cdot \frac{480+\overline{w}_a}{20+\overline{w}_a},\ \mathrm{m/s^2} \tag{6}$$

- modal velocity

$$V_0 = 20 \cdot \frac{480+\overline{w}_a}{100+\overline{w}_a},\ \mathrm{m/s} \tag{7}$$

with $w_a = w_a/1000$, and $w_a = m_{eff} \cdot g$, using $g$ = 9.81 m/s$^2$.

These equations are now fully converted into SI units for use in DDAM analyses and reflect empirical data from ship shock tests and serve as a foundation for DDAM stress assessment. For example (CASE I) for a system with an effective modal mass $m_{\mathrm{eff}}$ = 50 kg and assumed ω = 100 rad/s, the calculations proceed as follows

$$w_a = m_{eff} \cdot g = 50 \cdot 9.81 = 4905\ \mathrm{N} \tag{8}$$

$$\overline{w}_a = \frac{w_a}{1000} = \frac{4905}{1000} = 4.905\ \mathrm{kN} \tag{9}$$

$$A_0 = 101.99 \cdot \frac{480+\overline{w}_a}{20+\overline{w}_a} = 101.99 \cdot \frac{480+4.905}{20+4.905} \approx$$
$$\approx 101.99 \cdot \frac{484.905}{24.905} \approx 1982.4\ \mathrm{m/s^2} \tag{10}$$

$$V_0 = 0.508 \cdot \frac{480+\overline{w}_a}{100+\overline{w}_a} = 0.508 \cdot \frac{480+4.905}{20+4.905} \approx$$
$$\approx 0.508 \cdot \frac{484.905}{104.905} \approx 2.34\ \mathrm{m/s} \tag{11}$$

The calculated values of $A_0$ = 1982.4 m/s$^2$ and $V_0$ = 2.34 m/s are specific to the given effective modal mass and system configuration. These results highlight the importance of accurate effective modal mass determination and unit conversion for ensuring compliance with DDAM standards.

The next step in DDAM involves determining the dynamic design acceleration $D_a$ and the shock design value *SDV*. Taking $A_0$ and $V_0$ multiplying by $\omega$ (the circular frequency in radians per second). $D_a$ is the lesser of these two values

$$D_a = \min(A_0 \cdot g,\ V_0 \cdot \omega) \tag{12}$$

where $\omega_1$ = 100 rad/s, then

$$D_a = \min(1982.4,\ 2.34 \cdot 100) = \min(1982.4,\ 234.0) = 234.0\ \mathrm{m/s} \tag{13}$$

Taking the maximum of $D_a$ and 6$g$ (minimum acceleration value in SI as 6·9.1 = 58.86 m/s$^2$) we set the shock design value

$$SDV = max(D_a,\ 6g) = max(234 \cdot 0{,}58 \cdot 86) = 234.0\ \mathrm{m/s} \tag{14}$$

In this case, the system's natural frequency is not explicitly provided. When the natural frequency is unknown in practical DDAM applications for naval structures, it is common to assume a standard value of $\omega$≈100 rad/s. This assumption is based on the typical frequency range of ship structures subjected to shock loads, where an approximate fundamental frequency of 15.9 Hz is often used, leading to

$$\omega = 2\pi f \approx 2\pi \cdot 15.9 \approx 100\ \mathrm{rad/s} \tag{15}$$

This standardization ensures that the computed dynamic design acceleration $D_a$ remains within the expected range for ship-mounted equipment, as outlined in DDAM methodologies.

For structures where the natural frequency is explicitly determined (beams case), the actual computed frequency is used instead of this assumed value. For a steel beam (CASE II) with dimensions (length width height) $L$ = 1 m × $B$ = 0.1 m × $H$ = 0.1 m was fixed at one end with Young's modulus $E$ = 2.110$^5$ MPa, Poisson's ratio ν = 0.3 density, $\rho$ = 7850 kg/m$^3$, yield strength $R_e$ = 205 MPa, ultimate tensile strength $R_m$ = 345 MPa, the calculation starts from the moment of inertia:

$$I = \frac{b \cdot h^3}{12} = \frac{0.1 \cdot 0.1^3}{12} = 8.33\ \mathrm{m^4} \tag{16}$$

then cross-sectional area

$$A = b \cdot h = 0.1 \cdot 0.1 = 0.01\ \mathrm{m^2} \tag{17}$$

and the beam mass

$$m = \rho A L = 7850 \cdot 0.01 \cdot 1 = 78.5\ \mathrm{kg} \tag{18}$$

The first natural frequency for the beam with distributed mass is given by:

$$f_n = \frac{\alpha_n^2}{2\pi}\sqrt{\frac{EI}{mL^3}} \quad (19)$$

where $\alpha_1$ = 1.8755 then computed frequencies are

$$f_1 = 83.54 \text{ Hz} \quad (20)$$

For the beam taking an effective modal mass as $m_{eff} = m$ =78.5 kg, the calculations proceed as follows:

$$w_a = m_{eff} \cdot g = 78.5 \cdot 9.81 \approx 770 \text{ N} \quad (21)$$

$$\text{w}_a = \text{w}_a/1000 = 770/1000 = 0.770 \text{ kN} \quad (22)$$

$$A_0 = 101.99 \cdot \frac{480+\overline{w}_a}{20+\overline{w}_a} = 101.99 \cdot \frac{480+0.77}{20+4.905} \approx$$

$$\approx 101.99 \cdot \frac{484.905}{24.905} \approx 2360 \text{ m/s}^2 \quad (23)$$

$$V_0 = 0.508 \cdot \frac{480+\overline{w}_a}{100+\overline{w}_a} = 0.508 \cdot \frac{480+4.905}{20+4.905} \approx$$

$$\approx 0.508 \cdot \frac{484.905}{104.905} \approx 2.34 \text{ m/s} \quad (24)$$

Determining the dynamic design acceleration and the shock design value

$$D_a = \min(A_0 \cdot g,\ V_0 \cdot \omega) \quad (25)$$

(where $\omega_1 = 2\pi f_1$ is the circular frequency for the first mode) and take the maximum of $D_a$ and 6$g$ (minimum acceleration value in SI as 6 · 9.81 = 58.86 m/s$^2$ as the shock design value:

$$SDV = \max(D_a, 58.86) \quad (26)$$

The final shock design values is:

$$\omega_1 = 2\pi \text{f}_1 = 2\pi \cdot 83.54 \approx 524.85 \text{ rad/s} \quad (27)$$

$$D_a = \min(2360,\ 2.42 \cdot 524.85)=$$
$$\min(2360,\ 1270.25) = 1270.25 \text{ m/s}^2 \quad (28)$$

$$SDV = \max(1270.25,\ 58.86) = 1270.25 \text{ m/s}^2 \quad (29)$$

For a surface ship and hull mounted system (CASE III) but the same beam as in case II we have also an effective modal mass as $m_{eff} = m$ = 78.5 kg but the calculations proceed as follows

$$w_a = m_{eff} \cdot g = 78.5 \cdot 9.81 \approx 770 \text{ N} \quad (30)$$

$$w_a = w_a/1000 = 770/1000 = 0.770 \text{ kN} \quad (31)$$

$$A_0 = 101.99 \cdot \frac{480+\overline{w}_a}{20+\overline{w}_a} = 101.99 \cdot \frac{480+0.77}{20+4.905} \approx$$

$$\approx 101.99 \cdot \frac{484.905}{24.905} \approx 2092 \text{ m/s}^2 \quad (32)$$

$$V_0 = 0.508 \cdot \frac{480+\overline{w}_a}{100+\overline{w}_a} = 0.508 \cdot \frac{480+4.905}{20+4.905} \approx$$

$$\approx 0.508 \cdot \frac{484.905}{104.905} \approx 2.65 \text{ m/s} \quad (33)$$

The dynamic design analysis method (DDAM) typically considers only the first mode of vibration when assessing shock response. This is based on the fundamental assumption that the first mode contributes the most to the overall response in most practical cases. Higher-order modes (e.g. $f_2$, $f_3$) have shorter wavelengths and are less likely to be significantly excited by shock loads. The primary reasons for this approach include:

- dominant energy distribution; the first mode generally captures the largest displacement and absorbs the most energy from the shock event,
- engineering standards; NAVSEA 0908-LP-000-3010 emphasize using the fundamental mode for shipboard shock analysis,
- computational efficiency; higher-order modes contribute less to the overall response, so including them provides diminishing returns for design safety.

However, in specialized cases where higher-order modes play a critical role (e.g., very flexible structures or local resonances), a full modal analyses may be warranted.

## Interpretation of results

The computed shock design values provide insight into the structural resilience under dynamic loading conditions. The minimum threshold of 58.86 m/s$^2$ represents the baseline acceleration resistance required by standard guidelines, ensuring that even in less critical scenarios, structures maintain a fundamental level of durability.

In CASE I, where $D_a$ = 234.0 m/s² (≈ 24 $g$), the calculated value significantly exceeds the minimum threshold, indicating that the structure must be designed to withstand considerable dynamic forces. This suggests the necessity for robust material selection and reinforcement to prevent potential failure under extreme shock conditions.

In CASE II, with $D_a$ = 1270.25 m/s² (≈ 127 $g$), the required structural resistance is even higher, implying a much greater exposure to dynamic forces. Such high acceleration values necessitate advanced engineering considerations, including optimized damping mechanisms and structural reinforcements, to ensure that the system remains operational and within acceptable safety margins.

In CASE III, with $D_a$ = 2092.86 m/s² (≈ 210 $g$), the shock demand reaches its highest level among the analysed scenarios. This extreme value indicates that the structure is subjected to very intense dynamic loads, likely approaching or exceeding the limits of conventional design methods. It underscores the critical need for specialized engineering solutions, including enhanced material selection, energy-absorbing components, and rigorous qualification procedures, particularly for mission-critical naval systems exposed to underwater explosions.

Overall, the computed values emphasize the importance of evaluating dynamic responses accurately in structural design, ensuring that all components meet or exceed the prescribed safety requirements under shock conditions.

The computed shock design values provide insight into the structural resilience under dynamic loading conditions. The minimum threshold of 58.86 m/s² represents the baseline acceleration resistance required by standard guidelines, ensuring that even in less critical scenarios, structures maintain a fundamental level of durability.

While these acceleration values are significant, they are not exceptionally high compared to those experienced in extreme shock environments such as underwater explosions or severe impact loading. Military and naval structures are often designed to withstand accelerations exceeding 100*g*–200*g* in critical areas. Therefore, while these values indicate notable shock loads, they remain within a range that can be addressed with conventional structural reinforcement techniques.

### *Shock design spectrum*

The DDAM shock design spectrum presented in Figure 6 provides a graphical representation of the computed dynamic acceleration $D_a$ values across a range of natural frequencies. This spectrum is crucial for understanding the structural response of naval components subjected to underwater shock loads, allowing for a systematic assessment of the required design acceleration and can be set up individually in DDAM calculations against standard equations given by NRL-1396 standard.

The spectrum consists of three distinct regions, corresponding to different frequency ranges:

- low-frequency range (<15Hz) – in this region, the minimum design acceleration is constrained by 6*g* (58.86 m/s²), ensuring a baseline level of structural robustness even for components with low modal frequencies. This threshold, represented by the red dashed line, prevents designs from being under-conservative.
- mid-frequency range (15–100 Hz) – the acceleration increases linearly according to the equation:

$$D_a = V_0 \cdot \omega \tag{34}$$

  This relationship shows that dynamic acceleration is directly proportional to the modal velocity $V_0$ and the circular frequency $\omega$. In practical terms, components operating in this frequency range experience increasingly higher shock loads, requiring a corresponding increase in structural resilience.
- high-frequency range (>100 Hz) – the acceleration levels off, reaching its upper limit, indicated by the green dashed line. At these frequencies, modal acceleration dominates over velocity-dependent effects, meaning that

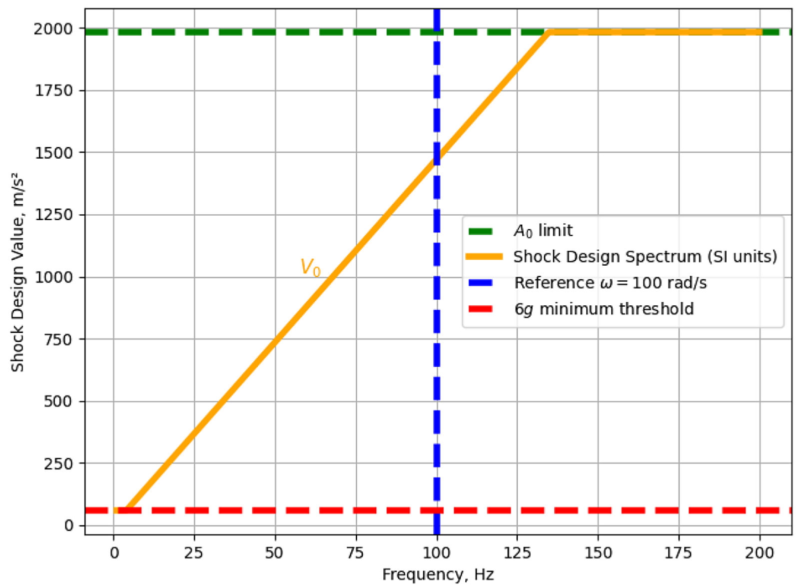


**Fig. 6.** DDAM shock design spectrum in SI units

further increases in frequency do not contribute to higher design accelerations. This plateau defines the maximum shock load a structure must endure.

The computed spectrum is directly linked to the analytical results obtained for specific shipboard components. The three primary cases analysed in this study are:

- CASE I $f_1 = 15.9$ Hz $\rightarrow \omega_1 = 100$ rad/s,
  $A_0 = 1982.4$ m/s²,
  $V_0 = 2.34$ m/s,
  $D_a = \min(1982.4, 234.0) = 234.0$ m/s²,
  $SDV = \max(234.0, 58.86) = 234.0$ m/s².

This case lies in the mid-frequency region, where  is velocity-controlled.

- CASE II $f_1 = 83.54$ Hz $\rightarrow \omega_1 = 2\pi f_1 = 524.85$ rad/s,
  $A_0 = 2360$ m/s²,
  $V_0 = 2.42$ m/s,
  $D_a = \min(2360, 1270.25) = 1270.25$ m/s²,
  $SDV = \max(D_a, 6g) = \max(1270.25, 58.86) =$ 1270.25 m/s².

This case is in the high-frequency range, where $A_0$ dominates.

- CASE III $f_1 = 83.54$ Hz $\rightarrow \omega_1 = 524.85$ rad/s,
  $A_0 = 2092$ m/s²,
  $V_0 = 2.65$ m/s,
  $D_a = \min(2092, 1391.86) = 1391.86$ m/s²,
  $SDV = \max(D_a, 6g) = \max(1391.86, 58.86) =$ 1391.86 m/s².

This case corresponds to a surface ship and hull-mounted system where the effective mass leads to an increased dynamic response. All examples align with the expected behaviour of the spectrum, confirming its validity in describing the structural response under shock loads.

### *Understanding control by $V_0 \cdot \omega$ and $V_0$*

The design acceleration $D_a$ in the DDAM is determined by the minimum of the two expressions:

$$D_a = \min(A_0, V_0 \cdot \omega) \tag{35}$$

This means that the governing factor for $D_a$ depends on the frequency range:

- in mid-frequency ranges, where $V_0 \cdot \omega < A_0$, the acceleration is velocity-controlled. As frequency increases, $D_a$ grows linearly with $\omega$.
- in high-frequency ranges, where $V_0 \cdot \omega > A_0$, the acceleration reaches a plateau at $A_0$. Here, modal acceleration dominates, and further frequency increases do not contribute to higher acceleration loads.

### *Velocity-controlled regime ($V_0 \cdot \omega$ dominates)*

When acceleration is controlled by $V_0 \cdot \omega$, it is essential to reduce modal velocity to mitigate shock effects. Practical strategies include:

- modifying system stiffness (e.g., by adding reinforcements).
- introducing damping elements (e.g., rubber pads, shock absorbers).
- reducing mass (e.g., minimizing inertia of certain structural elements).

In real applications, mid-frequency components are more sensitive to velocity-dependent effects, so their protection should focus on controlling $V_0$ rather than purely reducing acceleration.

### *Acceleration-controlled regime ($A_0$ dominates)*

When acceleration is limited by $A_0$, modal acceleration determines the structural response. In this case, the key strategies involve:

- optimizing material selection to enhance energy absorption (e.g., composites, damping materials).
- adjusting structural geometry (e.g., using strategically placed stiffeners).
- designing damping systems to reduce peak acceleration loads.

High-frequency components are particularly sensitive to peak acceleration values, making $A_0$ the limiting factor in their response.

### *Practical importance of SDV*

The $SDV$ is essential in structural engineering for several reasons:

- it provides a standardized acceleration value for stress analysis and structural verification,
- it ensures that designs meet safety and performance criteria under shock loads,
- it allows engineers to compare different structural components under a uniform loading framework,
- it aids in certification and compliance with regulatory requirements.

In summary, while $D_a$ describes the expected dynamic response, the $SDV$ is the practical design value used in engineering assessments to ensure structural integrity and compliance.

## STRESS IN TERMS OF DDAM – MODAL STRESS COMPUTATION

In classical finite element analyses (FEM), the motion of a structure is governed by the second-order differential equation

$$\mathbf{Ku}(t) + \mathbf{Cu}(t) + \mathbf{Mu}(t) = \mathbf{F}(t) \tag{36}$$

where: $\mathbf{K}$ – global stiffness matrix,
$\mathbf{C}$ – damping matrix,
$\mathbf{M}$ – global mass matrix,
$\mathbf{u}(t)$ – displacement vector as a function of time,
$\mathbf{F}(t)$ – external force vector.

The displacement vector $\mathbf{u}(t)$ is obtained by solving Eq. (36) using numerical time integration in dynamic simulations. Once displacements are known, the stress field is computed through the strain and constitutive relations

$$\boldsymbol{\sigma} = \mathbf{D}\boldsymbol{\varepsilon} \tag{37}$$

where: $\boldsymbol{\sigma}$ – stress vector,
$\mathbf{D}$ – material constitutive matrix (stiffness),
$\boldsymbol{\varepsilon}$ – strain vector.

The strain vector is obtained from displacements via the strain-displacement relation

$$\varepsilon = \mathbf{Bu} \tag{38}$$

where: $\mathbf{B}$ – strain-displacement transformation matrix.

In the DDAM procedure, stresses in the structure are not computed from global forces alone. Instead, the method relies on modal decomposition and spectrum-based response calculations. The procedure involves the following steps:

- modal analysis, the structure undergoes eigenvalue analysis to extract mode shapes $\Phi_j$, natural frequencies $\omega_j$, and participation factors $\Gamma_j$ in each direction (X, Y, Z).
- shock spectrum assignment, based on the selected standard (e.g., NRL 1396), a response spectrum $S_j$ is assigned to each mode.
- modal stress computation, for each node or element location $i$, the modal contribution to stress is calculated as

$$\sigma_i = \sum_{j=1}^{N} \Gamma_j \Phi_{ij} \cdot S_j \tag{39}$$

where: $\sigma_i$ – modal stress at node or element location $i$,
$\Gamma_j$ – participation factor of mode $j$,
$\Phi_{ij}$ – stress influence coefficient of mode $j$ at point $i$,
$S_j$ – spectral acceleration (shock input) assigned to mode $j$,
$N$ – number of considered eigenmodes.

- mode combination – to obtain total stress values, the modal contributions are combined using the square root of the sum of squares (SRSS) or more advanced methods. For correlated modes, the correlation matrix $C_{ij}$ is used

$$\sigma_{\max} = \sqrt{\sum_{i=1}^{N} \sum_{j=1}^{N} C_{ij} \sigma_i \sigma_j} \tag{40}$$

where: $\sigma_{max}$ – combined maximum stress from all modes,
$C_{ij}$ – modal correlation coefficient between modes $i$ and $j$,
$\sigma_i$, $\sigma_j$ – modal stresses due to modes $i$ and $j$,
$N$ – number of considered eigenmodes.

- directional evaluation, the above process is repeated separately for each principal direction of excitation (vertical, longitudinal, athwartship). The highest resulting value is used for stress qualification.
- stress mapping, the computed stress field is mapped to the structure as equivalent static stress, which can be visualized and compared to allowable limits.

### Derivation and role of modal equations in stress designation

To formulate and solve the equations of motion in modal space, it is necessary to first extract the mode shapes of the structure. This process consists of two main steps:

- solving the eigenvalue problem, the undamped free vibration equation is expressed as:

$$(\mathbf{K} - \omega^2\,\mathbf{M})\phi = \mathbf{0} \tag{41}$$

where: $\mathbf{K}$ – global stiffness matrix,
$\mathbf{M}$ – global mass matrix,
$\omega$ – natural circular frequency,
$\phi$ – eigenvector (mode shape).

Solving this problem yields $N$ eigenvalues $\omega^2$ and corresponding mode shapes which form the mode shape matrix

$$\boldsymbol{\Phi} = [\boldsymbol{\phi}_1, \boldsymbol{\phi}_2, \dots \boldsymbol{\phi}_N] \tag{42}$$

- transformation to modal space – with the mode shape matrix $\boldsymbol{\Phi}$ available, the original system of equations

$$\mathbf{M\ddot{u}} + \mathbf{Ku} = \mathbf{F} \tag{43}$$

can be transformed using the modal coordinate substitution

$$\mathbf{u}(t) = \boldsymbol{\Phi}\cdot\mathbf{q}(t) \tag{44}$$

where: $\mathbf{u}(t)$ – displacement vector as a function of time,
$\mathbf{\Phi}$ – mode shape matrix (eigenvectors),
$\mathbf{q}(t)$ – modal coordinates (time-dependent generalized displacements).

resulting in the modal equations of motion Eq. (14). In DDAM the dynamic equilibrium equation Eq. (8) is transformed using mode shapes as

$$\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\mathbf{\Phi}\ddot{\mathbf{q}} + \mathbf{\Phi}^{\mathrm{T}}\mathbf{K}\mathbf{\Phi}\mathbf{q} = \mathbf{\Phi}^{\mathrm{T}}\mathbf{F} \tag{45}$$

This reduced system enables efficient computation of the dynamic response in terms of modal coordinates $q(t)$ vector and is a fundamental step in DDAM and other spectrum-based dynamic analyses methods.

The spectral acceleration corresponding to mode $j$ is used to compute modal displacement

$$q_j = \frac{S_j}{\omega_j^2} \tag{46}$$

where: $q_j$ – modal displacement (generalized coordinate) for mode $j$,
$S_j$ – spectral acceleration assigned to mode $j$,
$\omega_j$ – natural circular frequency of mode $j$.

The modal stress at location $i$ due to mode $j$ is calculated as

$$\sigma_{ij} = \Phi_{ij} \cdot q_j \tag{47}$$

Alternatively, a computation algorithm combines all modal effects at point $i$ using the participation factor $\Gamma_j$ and spectrum $S_j$ as

$$\sigma_i = \sum_{j=1}^{N} \Gamma_j \Phi_{ij} \cdot S_j \tag{48}$$

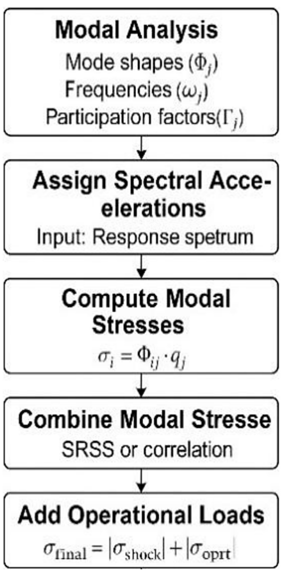


**Fig. 7.** Flowchart of modal stress computation in DDAM using LS-DYNA.

The total combined stress (e.g., von Mises) is evaluated by summing modal contributions with SRSS combination

$$\sigma_{\text{total}} = \sqrt{\sum_{i=1}^{N} \sigma_i^2} \tag{49}$$

or correlation method

$$\sigma_{max} = \sqrt{\sum_{i=1}^{N} \sum_{j=1}^{N} C_{ij}\sigma_i\sigma_j} \tag{50}$$

Finally, the shock-induced stress is added to operational stress (e.g., static load or pressure) using the equation

$$\sigma_{final} = |\sigma_{shock}| + |\sigma_{oprt}| \tag{51}$$

and can be illustrated as the flowchart

$$\boxed{\begin{array}{c} \sigma_i \longrightarrow \sigma_{\text{total}}\ (\text{SRSS}) \quad \text{or} \quad \sigma_{\max}\ (\text{correlation}) \\ \Downarrow \\ \boxed{\sigma_{\text{shock}}} \longrightarrow \sigma_{\text{final}} = |\sigma_{\text{shock}}| + |\sigma_{\text{oprt}}| \end{array}} \tag{52}$$

Figure 7 presents the entire DDAM stress evaluation procedure as a flowchart. This schematic summarizes the process from modal analyses through stress combination to final stress evaluation.

This algorithm offers an efficient and practical method for estimating shock-induced structural stresses by utilizing response spectrum inputs. It eliminates the need for full transient time-domain simulations, making it especially suitable for the qualification of naval and shipboard systems under dynamic shock conditions.

## STRESS IN SHIP ONBOARD EQUIPMENT

The dynamic design analysis method was employed to evaluate the shock resistance of naval equipment mounted on structural foundations subjected to underwater explosion (UNDEX) loading. The focus is on a cylindrical marine tank supported on a lattice steel frame, commonly found in naval and offshore systems (Fig. 8). For the analyses, it was assumed that the tank and frame assembly is mounted to a horizontal structural element of the ship, i.e., a deck-mounted system. The frame was subjected to a vertical acceleration applied vertically upward from below, corresponding to a peak value of approximately 125$g$. The structural strength of the system was evaluated under acceleration conditions compliant with requirements specified by, among others, DNV [2].

The study uses DDAM alongside traditional FEM simulations for validation and cross-comparison. Using LS-DYNA, the DDAM workflow involves the following steps:

- perform modal analysis to obtain mode shapes, frequencies, and participation factors.
- define the acceleration shock spectrum based on empirical or standard curves (e.g., NRL 1396).
- calculate modal acceleration $A_0$ and modal velocity $V_0$ using an appropriate empirical formulas:

Ship type "surface":

Hull Mounted Systems:

Reference Equations:

− modal acceleration:

$$A_0 = 196.2 \cdot \frac{(37.5+\overline{w}_a)(12+\overline{w}_a)}{(6+\overline{w}_a)^2}, \mathrm{m/s^2} \tag{53}$$

− modal velocity:

$$V_0 = 1.524 \cdot \frac{12+\overline{w}_a}{6+\overline{w}_a}, \mathrm{m/s^2} \tag{54}$$

- determine the dynamic design acceleration $D_a$,
- establish the shock design value $SDV$,
- compute the dynamic force $F_{dyn}$ as

$$F_{dyn} = m_{eff} \cdot SDV \tag{55}$$

- apply the dynamic force as an equivalent dynamic load to the tank wall to asses stress

$$\sigma_{\mathrm{DDAM}} = \frac{F_{\mathrm{dyn}}}{A_{\mathrm{eff}}} = \frac{m_{\mathrm{eff}} \cdot SDV}{A_{\mathrm{eff}}} \tag{56}$$

where: $m_{eff}$ – the effective modal mass,
$A_{eff}$ – the effective load-bearing area.

The analysed object is located in ZONE 1 (Fig. 8) below the waterline, which is the most severely. The shock impulse is represented as a time-dependent acceleration function described by a double half-sine waveform. The mathematical expressions defining this waveform are presented in Figure 9. Predefined acceleration profiles for all shock zones are illustrated in Figure 10.

The next figures present the results obtained using the DDAM methodology (Fig. 11) and transient FEM simulations (Fig. 12), which were comparatively analysed to assess consistency in stress predictions and to evaluate the suitability of DDAM for preliminary design under underwater shock loading.

In FEM computation, compared "max" stresses were obtained as the time envelope of the von Mises

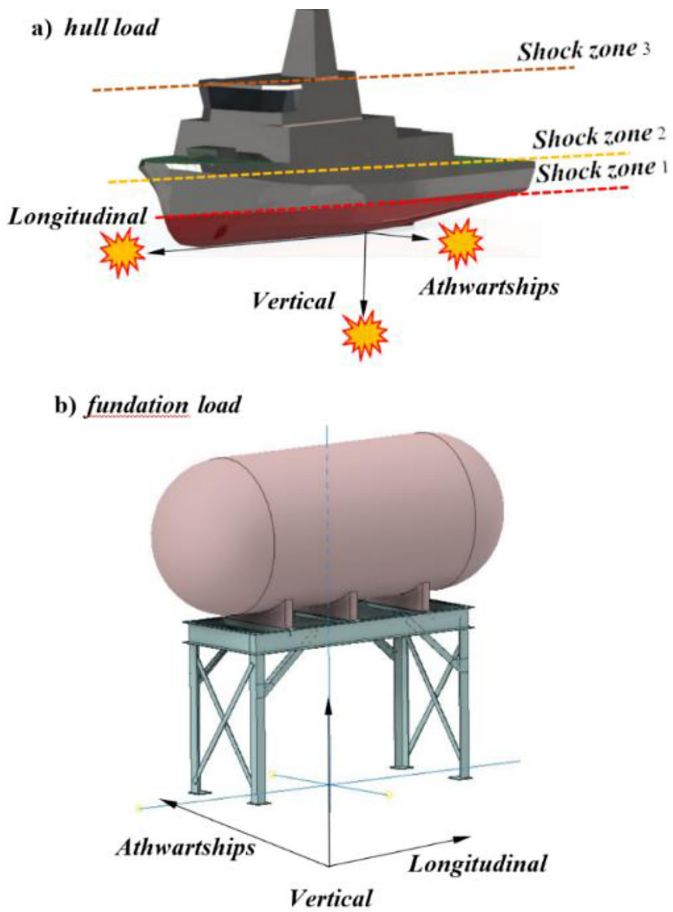


**Fig. 8.** Directions and zones of acceleration acting on the hull and the foundation

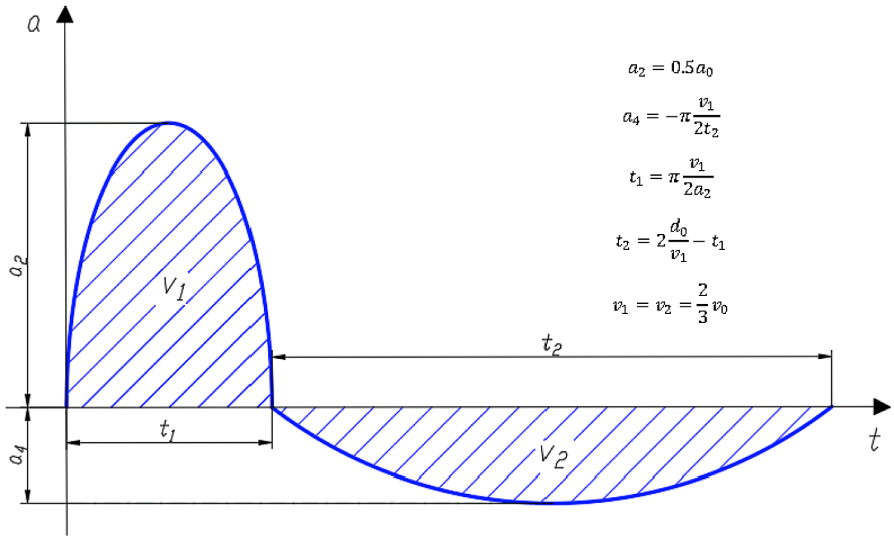


**Fig. 9.** Double half-sine shock pulse

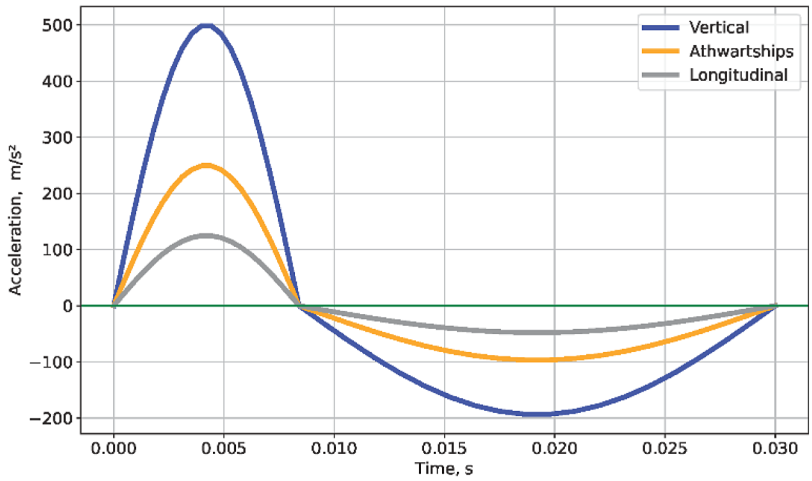


**Fig. 10.** Sinusoidal kinematic load profile for zone 1

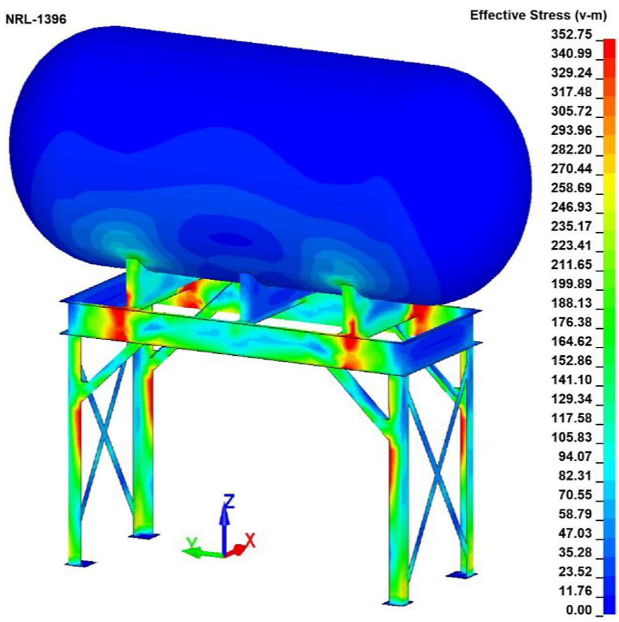


**Fig. 11**. Determination of stress according to the DDAM methodology

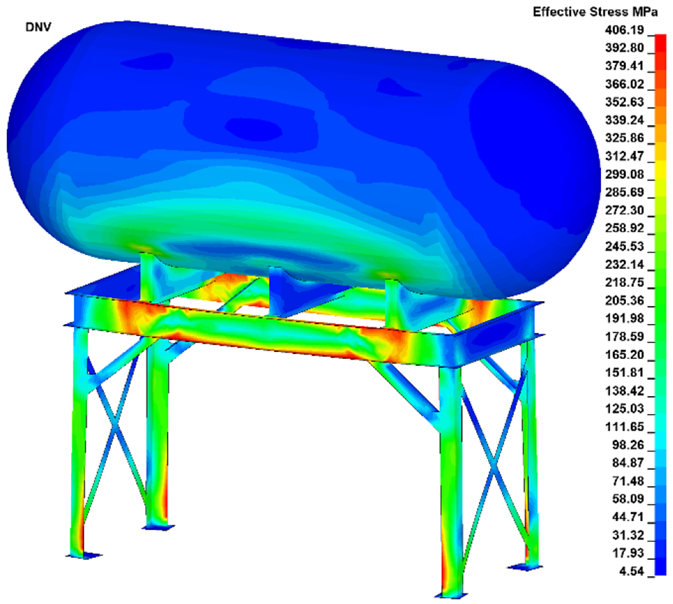


**Fig. 12.** Determination of stress according to explicit FEM methodology

effective stress over the whole shock window. Values were read as nodal-averaged stresses and the envelope over time was formed to capture the peak response. To avoid numerical outliers from single elements, we report the global absolute maximum for the whole structure (e.g., 406.19 MPa), and a representative maximum in the critical support zones defined as the 95th percentile within a region of interest around the tank–frame joints (typically 292 MPa to 320 MPa). This criterion suppresses single-element hot spots and yields engineeringly meaningful values for comparison.

In DDAM computation, directional spectral accelerations *SDV* $a_x$, $a_y$, $a_z$ were converted to equivalent inertial loads $F_{dyn} = m_{eff}\ a_i$, where ai corresponds to the *SDV* (Eq. (24)) in direction *i*. The effective modal masses $m_{effi}$ were obtained by including as many modes in direction *i* as needed to reach a cumulative modal mass participation of at least 90%, as required by MIL-STD-901D. Bearing stresses along the dominant load path were then evaluated as $\sigma_{\text{DDAM}} = F_{dyn}/A_{eff}$ and combined using the SRSS procedure. The combination across the three spatial directions is performed at the final stage of the procedure (see Eq. (18–19) and (Fig. 7)). Hence, FEM and DDAM are compared on the same physical quantity (stress). FEM captures localized peaks, while DDAM provides a conservative global estimate.

For a direct quantitative comparison, the maximum effective stress obtained via DDAM was $\sigma_{\text{DDAM}} = 258.3$ MPa, while the corresponding FEM analysis yielded $\sigma_{\text{FEM}} = 292.1$ MPa. The relative difference was calculated as:

$$\Delta = \frac{|\sigma_{\text{FEM}} - \sigma_{\text{DDAM}}|}{\sigma_{\text{FEM}}} \cdot 100\% = \\ = \frac{|292.1 - 258.3|}{292.1} \cdot 100\% \approx 11.6\% \tag{57}$$

Table 1 summarizes this comparison of stress values between the two approaches. It highlights that DDAM slightly underestimates the effective stress relative to FEM, but the discrepancy remains within 12%. This confirms the suitability of DDAM for preliminary engineering assessments, while FEM provides more detailed local insight

These findings confirm that DDAM provides a conservative but reasonably accurate estimate of the global shock response, while FEM captures localized effects such as stress amplification at joints, peak stress concentrations, non-linear material behaviour, and geometric discontinuities.

**Table 1.** Comparison of stress values obtained via DDAM and FEM

| Case | $\sigma_{\text{DDAM}}$ MPa | $\sigma_{\text{FEM}}$ MPa | Difference % |
|---|---|---|---|
| Tank-frame structure | 258.3 | 292.1 | 11.6 |

The difference between the two methodologies does not exceed 12%, which supports the feasibility of DDAM as a rapid preliminary design tool under underwater shock loading. In contrast, FEM remains indispensable for capturing detailed localised phenomena, particularly in critical support regions of the tank–frame interface. Thus, DDAM offers an efficient first-order estimation of the shock response, especially valuable during early design stages where rapid evaluation is necessary, whereas FEM ensures accurate assessment of complex local effects.

## CONCLUSIONS

In shock resistance analysis of naval structures, simplified and numerical methods serve complementary roles. DDAM offers rapid, conservative estimates valuable in early design, while FEM, as implemented in LS-DYNA or DYSMAS, captures detailed UNDEX phenomena, including cavitation and bubble pulsation, but requires substantial computational resources. In practice, combining both approaches proves most effective where DDAM enables fast preliminary assessments, and FEM provides accurate validation of localized non-linear effects, ensuring both efficiency and reliability in critical naval applications.

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## REFERENCES

1. Lloyd's Register. Rules and Regulations for the Classification of Naval Ships. Tech. rep. Updated version of naval classification rules. LR, 2022 Jan.
2. Det Norske Veritas. Rules and Regulations for the Classification of Naval Ships. Tech. rep. Updated regulations for naval ship classification. Norway: DNV, 2022 Jan.
3. Naval Sea Systems Command. Shock Design Criteria for Surface Ships. T9070-AJ-DPC-120/3010. NAVSEA Design Practices and Criteria Manual; Approved for public release. Naval Sea Systems Command. Washington, DC, 2017 Sep.
4. Grządziela A, Załęska-Fornal A, and Kluczyk M. The single degree of freedom simulation model of underwater explosion impact. Archives of Acoustics 2020; 45:341–348.
5. Flis L. Szacowanie odporności udarowej okrętu z wykorzystaniem symulacji komputerowej. Logistyka 2014.
6. Flis L. Full floating structure underwater explosion with pulsation and cavitation effect FEM simulation case. Bulletin of the Polish Academy of Sciences Technical Sciences 2025 :e155036. DOI: 10.24425/bpasts.2025. 155036. Available from: http://journals.pan.pl/ Content/135710/PDF/BPASTS-04999-EA.pdf
7. Kiciński R. Estimating the Size of a Crater after an Underwater Explosion. Advances in Science and Technology Research Journal 2023; 17:187–94. DOI: 10.12913/ 22998624/171503. Available from: https://doi.org/10.12913/22998624/171503
8. Tasdelen E et al. Shock Analysis of Onboard Equipment Submitted to Underwater Explosion. MA thesis. Université de Liège, Liège, Belgique, 2018.
9. Didoszak JM. Updated approach to shock failure assessment of shipboard equipment. PhD thesis. Monterey, CA; Naval Postgraduate School, 2019.
10. Huang Y and Cui Z. DDAM Analysis with LS-DYNA®. *Proceedings of the 12th LS-DYNA European Users' Conference*. 2019.
11. Koehler M, McCoy W, and Patel M. Dynamic Design Analysis Method to Evaluate Shipboard Shock in LS-DYNA®. *Proceedings of the 15th International LS-DYNA® Users' Conference*, 2018.
12. Szturomski B. The Impact of Non-contact Underwater Explosions on the Ship's Hull in FEM Approach. Academic Press AMW, 2021,
13. Flis L. Full floating structure underwater explosion with pulsation and cavitation effect FEM simulation case. *Proceedings of TKI 2024*. Early Access. 2025. Available from: https://tki.wat.edu.pl/media/streszczenia-tki- 2024/flis-tki2024.pdf [Accessed on: 2025 May 22],
14. Sigrist C and Broc K. A versatile method to calculate the response of equipment mounted on ship hulls. Finite Elements in Analysis and Design 2023; 217:104001. DOI: 10.1016/j.finel.2023.104001
15. Matos A et al. Computational modelling of underwater shock on naval structures – A review. Flow 2024; 4:100204. DOI: 10.1016/j.flow.2024.100204
16. Zhang Y et al. Research on the Shock Environment Characteristics of a Marine Diesel Engine Based on a Large Floating Shock Platform. Journal of Marine Science and Engineering 2023; 11:2163. DOI: 10.3390/ jmse11112163
17. Didoszak J and Kwon Y. Failure assessment of shipboard equipment subjected to underwater explosions. Multiscale and Multidisciplinary Modeling, Experiments and Design 2025; 8:17.
18. U.S. Navy. Shock Tests, H.I. (High-Impact)

Shipboard Machinery, Equipment, and Systems, Requirements for. Tech. rep. MIL-S-901D(Navy). Superseded by MIL-DTL-901E (2017). Washington, DC: U.S. Navy, 1989 Mar.

19. https://www.hitestlabs.com/mil-dtl-901-testing. [Accessed on: 2018 Jan 16].
20. https://www.youtube.com/watch?v=lQ4Rmj4pCOA&abchannel=LeonardoDRS [Accessed on: 2018 Jan 16].
21. Naval Sea Systems Command. Shock Design Criteria for Surface Ships. Tech. rep. 0908-LP-000-3010, Rev. 1. Superseded by T9070-AJ-DPC-120/3010 (2017). Washington, DC: Naval Sea Systems Command, 1995 Sep.
22. O'Hara G and Belsheim R. Interim Design Values for Shock Design of Shipboard Equipment. Tech. rep. National Technical Information Service, 1963.
23. Reid WD. The response of surface ships to underwater explosions. Tech. rep. Department of Defence Australia, 1996.