Shock characterization of fiberglass composite laminates: numerical and experimental comparison

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Abstract. When subjected to a no contact underwater explosions (UNDEX), naval composite structures show highly nonlinear deformations. In this paper, fiberglass composite laminates are characterized dynamically. Experimentally, modal analyses are carried out to determine the modal parameters of the specimens, while dedicated shock tests are performed using the MIL S 901 D Medium Weight Shock Machine to measure their shock deformations. Numerically, finite element model is built up, running both modal and implicit dynamic analyses to predict the structural response of different E-Glass laminates. In the end, results obtained by calculations are compared with experimental data, validating the model.

Keywords. shock, composites, experimental, numerical, design

1. Introduction

Naval ship structures should be designed to withstand to no contact underwater explosions (UNDEX), according to military requirements. Composite structures in naval applications have been object of study for their low weight properties and absence of contribution to the ship magnetic signature. However, they are also required to show excellent properties of shock resistance. Fiber reinforced glass laminates can show a strain rate dependence that depends by the impact velocity and the manufacturing method, as reported in the work by Barrè et al. [3] and Welsh and Harding [16]. LeBlanc and Shukla [8] studied experimentally and numerically the effects of UNDEX to some E-glass composites, assuming that the material inputs are determined from quasi-static tests data. In any case, the dynamic transient response to UNDEX of fiberglass laminates is not clearly defined.

In this paper, MIL S 901 D Medium Weight Shock Machine (MWSM) [11] [12] is used to verify the shock response of naval ship construction materials. This machine reproduces the shock effects on board and its behaviour is easily reproducible using a

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simple mathematical mass-spring-mass model, as reported by Clements [5]. A dedicated structure, specially designed for these tests to induce large deflections, is used to support the fiberglass specimens. Before verifying shock response, the materials mechanical properties are characterized using Experimental Modal Analysis (EMA), following the guidelines reported in [7]. Modal analyses and shock tests are performed for specimens of different materials, thicknesses and weights.

A finite element (FE) model is built applying the same modelling strategies used in ref. [9], and fed by the composite parameters achieved by EMA results, adding the MWSM mechanism defined by Clements [5]. In the end, the comparison with experimental data validates the model. In this paper, for the sake of shortness only calculation and test results of laminates of a single manufacturer are shown.

2. Modal analysis

2.1. Experimental Modal Analysis

The fiberglass laminates specimens object of study are beams of square cross section with nominal length and breadth (2500x250 mm), different thicknesses (20, 40 mm), weights per unit area (reported as G1, G2 g/m^2 to preserve industrial privacy) and materials produced by four different companies, identified by a letter only (A, B, C and D) for confidentiality reason.

The EMA is related to the study of free vibration of the fiberglass laminate beams, simply supported almost at ends, with a free span of 2250 mm in between supports. The hammer-roving method is used, considering 13 measurements points along the beam axis and achieving the related Frequency Response Functions (FRFs). The driving points used as reference for the FRFs measurements are points no. 6 and 7 (see Fig. 1). The goal of these tests is to estimate the modal parameters (natural frequencies and damping) and shapes of given specimens. Me Scope Ves commercial software [10] is used for the determination of the natural frequencies and related Eigenvalues, performing the quick curve fitting function available in the software environment.

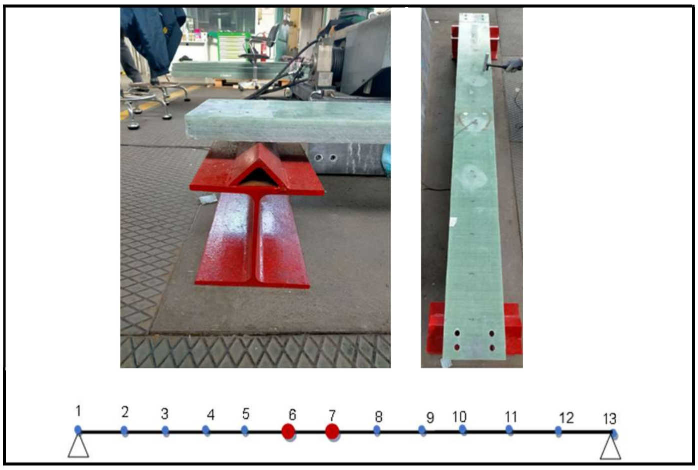


Figure 1. Experimental Modal Analysis set-up and measurement points

2.2. Analytical formulation and numerical method

Flexural natural frequencies obtained by experimental modal analyses are used to determine the axial Young Modulus of fiberglass laminates. By the analytical formula provided by the beam theory for simply supported beams [6], it is possible to calculate the longitudinal Young Modulus for each mode E_n as:

$$E_n = \frac{4\pi^2}{{A_n}^2} \frac{f_n \rho S l^4}{J} \tag{1}$$

where ρ is the material density, J is the inertia modulus of the beam, S its cross-section area and L its free span. A_n values are different for each vibration mode and are available in literature [6]. A mean value of longitudinal Young modulus E_L is assessed for each specimen, dividing the sum of the axial Young modulus for each mode E_n for the number of modes n.

$$E_L = \frac{\sum_n E_n}{n} \tag{2}$$

Then, using this mean Young modulus it is possible to recalculate the natural frequencies f_n for each mode (from Eq. (1)) and comparing results with experimental data.

In the numerical FE model, the material is characterized considering its orthotropic behaviour: the same E_L value assessed previously is used for the transversal Young modulus E_T, to reproduce the biaxial properties of the laminates. Shear modulus G and Poisson ratio ν are obtained by quasi-static tests performed according to ASTM standards [2]. The beam is discretized using 4-nodes multi-layered shell elements having 5 x 5 mm size and MITC formulation. Supports are placed at 125 mm from beam ends (see Fig. 2). The FE modal analysis is run using the Enriched Subspace Iteration Method, available in ADINA™ software, to estimate natural frequencies and modal shapes [1][4].

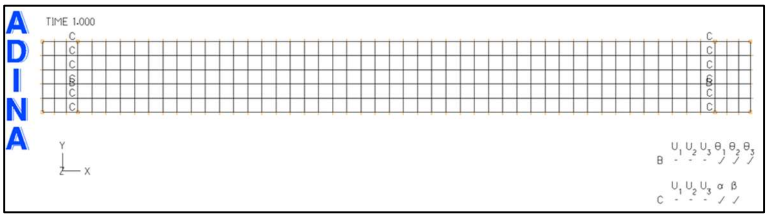


Figure 2. Modal Analysis - Finite Element setting

2.3. Numerical, analytical and experimental comparison

In the following graphs, the flexural natural frequencies up to the 4[th] modal shape, measured by EMA and calculated both analytically and numerically, are reported in some examples for specimens of manufacturer “A”, see Fig. 3. The first modal frequency is also important to properly set the time step size for subsequent dynamic calculations.

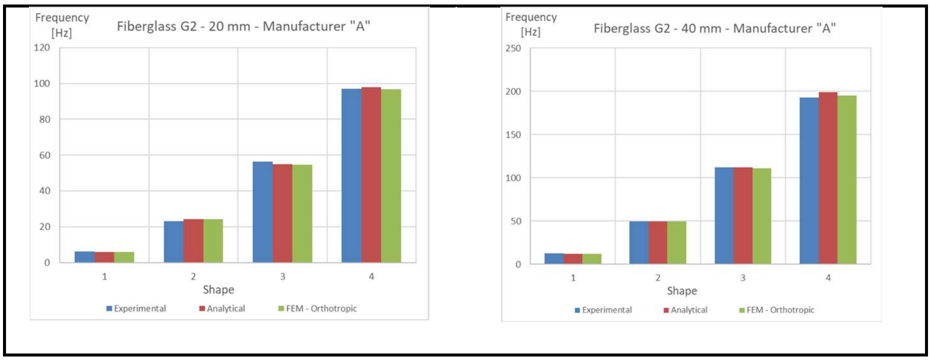


Figure 3. Flexural natural frequencies – Fiberglass G2 – 20, 40 mm thickness – Manufacturer "A"

3. Shock analysis

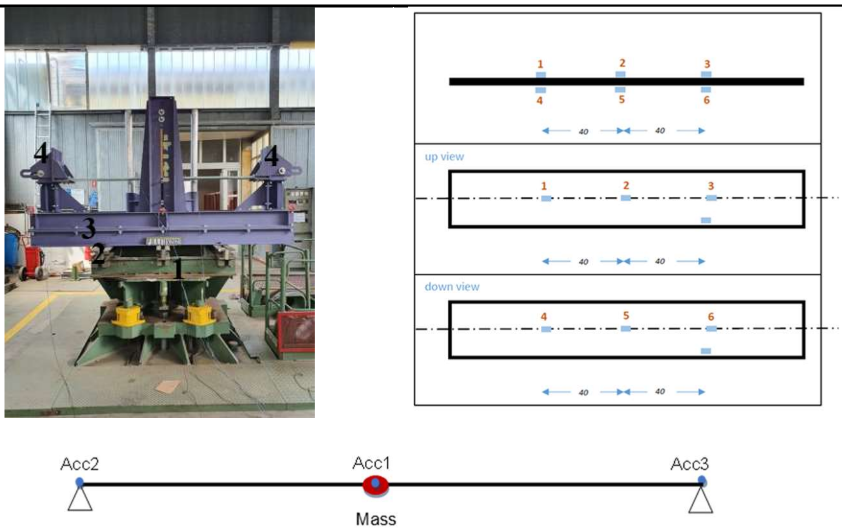
3.1. Shock tests setting and measurements

The shock tests are carried out using a MIL S 901 D medium weight shock machine. The specimens are connected to the machinery by means of a dedicated structure specially designed for these tests to induce large deflections. The boundary conditions are built to avoid unwanted stress concentrations at beam ends. Therefore, the fiberglass beams are constrained so as they are free to rotate and translate at the ends where they are bolted to purposely designed rollers. A calibrated mass of 60 kg, free to translate in the vertical direction, is rigidly connected to the centre of the specimen by means of two metal plates bolted together in a sandwich configuration. When the thickness of the beams increases an added mass is set upon the beam in the middle to increase the total deflection.

To measure the structural response, six linear strain gauges are set in the longitudinal direction along the beam axis. They are divided symmetrically: three on the top face and three on the bottom face of the specimen. On the top face, strain gauge no. 1 is set on the left side (40 cm apart from midspan), no. 2 at midspan and no. 3 on the right side (40 cm apart from midspan). On the bottom face, the same configuration is applied for strain gauges no. 4-5-6. To evaluate the input velocity, useful for subsequent dynamic calculation, no. 4 accelerometers are installed: no. 1 on the mass, no. 2-3 on the rollers and no. 4 on the MIL S 901D Medium Weight Shock Machine anvil table. See Fig. 4.

In the end, a high-speed camera is suitably placed to record the tests and to measure the maximum deflection of the beam: to achieve this result a graduated scale is drawn on the centre column of the supporting structure.

For each specimen, no. 6 shock tests were carried out, increasing gradually the hammer height from 60 cm to 170 cm. The shock tests are numbered in increasing order from test number 1, that corresponds to 60 cm hammer height to shock test number 6 that refers to 170 cm hammer height.



1. MIL S 901 D medium weight shock machine anvil table
2. Standard mounting fixtures
3. Dedicated support structure specially designed for the tests
4. Rollers

Figure 4. MIL S 91 D Shock test set-up (left side) – Strain gauge setting (right side) – Accelerometer setting (bottom side)

3.2. Mathematical model and finite element method

The numerical analysis is performed using ADINA™ software [1]. The mathematical model used to simulate the behaviour of the MSWM with a dead-weight load is that proposed by Clements [5], in which the whole configuration is regarded as mass-spring-mass system. The anvil is replaced by a mass Ma (2030 kg), that moves vertically with an initial velocity Va, when hit by the hammer. The support structure is represented by a mass Ms (1900 kg), connected to the anvil table by a spring, simulating the supporting channel stiffness K. In the Ms value, the standard mounting mass is included. The K value is chosen considering that the average MSWM natural frequency is 65.2 Hz. The MWSM natural frequency depends by the natural frequency of the installation supports. The standard value is reported in[5]. In this model, also damping and displacement limits of the anvil table travel (76 mm) are included. The damping value is set to 4% of the critical one, considering the amplitudes of caused motion maxima. In addition, the fiberglass specimen is connected to the support structure by means of rollers, characterized with their mass Mr (50 kg), free to rotate and to translate in the longitudinal (X) direction. The composite beam is rigidly connected to a mass, free to move in vertical direction (Z) with its velocity Vm, when hit by the anvil table impact. See Fig. 6.

In the FE environment, as mentioned, 5 x 5 mm MITC 4-nodes multilayer shell elements are used for the fiberglass specimen, characterized by mechanical features achieved by EMA and actual density properties measured experimentally. 3-D contact algorithms [1] are used to simulate the contact between the hitting mass and the composite beam and to reproduce the anvil table travel limits. The applied loading conditions are the initial velocities of the anvil table (Va) and of the mass (Vm), that correspond to the maximum initial velocities results of the impacts, obtained integrating the accelerometers measurements of the MSWM shock tests. Gravity is applied on the model as mass proportional load. A non-linear dynamic analysis is performed using implicit Bathe Method time-integration [1] [4], in which the time step selection depends by the results of EMA. In fact, the first natural frequencies are in the range of 5-6 Hz. Therefore, to obtain a certain accuracy in the results, time step (Δt) selection has been chosen $\Delta t=10^{-3}$, a value that corresponds to about 1/100 of natural period. As far as boundary conditions is concerned, the anvil mass and the mass connected to the specimen are constrained in longitudinal X direction and in transversal Y rotation, so they can move only in vertical direction.

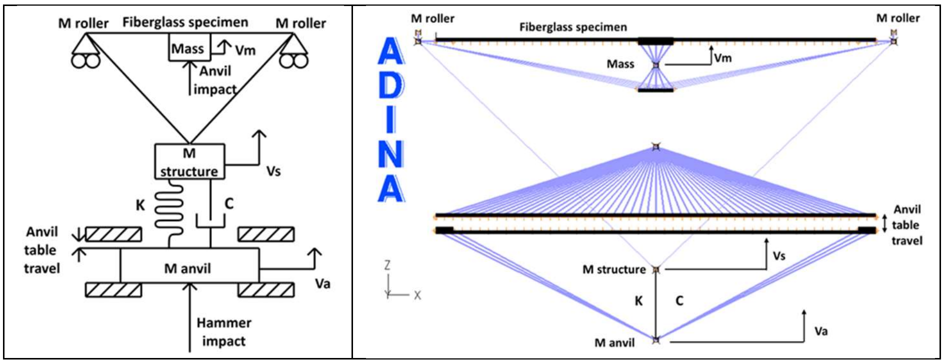


Figure 5. Shock test model and depiction in FE environment (XZ plane)

3.3. Results and comparison

From accelerometer measurements, maximum impact velocities are obtained, integrating the measured acceleration time histories. To obtain better accuracy, avoiding peak load velocities errors as reported in reference [5], peak anvil velocity numerical input Va is set to achieve the maximum velocity value of the structure Vs equal to the average of the peak velocity values obtained integrating the time histories of accelerometers no. 2-3 on the rollers (Fig. 4). Peak anvil and peak mass velocities are used as input velocity conditions for dynamic calculations.

The maximum deflection of the fiberglass specimens is acquired by the images taken by the recorded videos, using the graduated scale located in the center of the support structure. The measurement is realized, considering that the runners are free to move in the vertical direction only. Therefore, a black straight line is drawn, connecting the beam ends, in order to define the reference point of the deflection: the measurement is taken from the reference line to the first metal plate connecting the beam and the mass. Some examples are reported in Fig.s 6-7, where the maximum deflection of the most stressed impact is shown (no. 6). Some examples of the axial maximum strain values calculated by ADINA^{TM} software for some of the most stressed impacts (no. 6) are shown in figures 8-9. The comparison of numerical and experimental deflection and measured and calculated axial strain for each strain gauge position is shown in Table 1, reporting the values of the most stressed tests for specimens of different materials, weight and thicknesses. The numerical values are in good agreement with experimental data, validating the model and showing that the Finite Element simulation can be used to predict the transient response of these fiberglass laminates to an underwater shock explosion scenario.

Figure 6. Impact no. 6 - Maximum deflection (380 mm) for G2, 20 mm, manufacturer “A” specimen (Image taken from a recorded video)

Figure 7. Impact no. 6 - Maximum deflection (180 mm) for G2, 40 mm, manufacturer “A” specimen (Image taken from a recorded video)

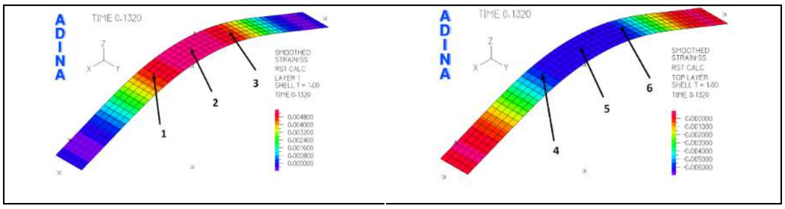


Figure 8. Numerical axial strain – Impact nr. 6 – G2– 20 mm – manufacturer “A” – Time step 0.132 s – Traction (left) and Compression (right)

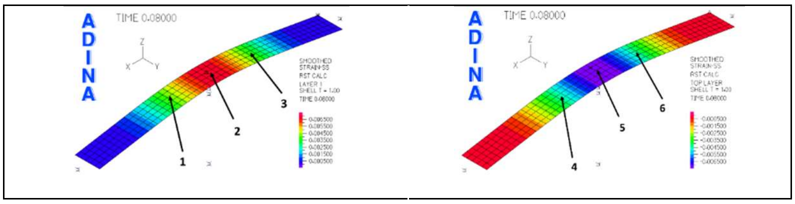


Figure 9. Numerical axial strain – Impact no. 6 –G2– 40 mm – manufacturer “A” – Time step 0.080 s – Traction (left) and Compression (right)

Table 1. Test 6 – Experimental and numerical comparison of deflections and strains for some specimens of manufacturer “A”

	Deflection [mm]			**Strain [10^3]**					
Specimen	G1 g/m^2 – 20 mm	G2 g/m^2 – 20 mm	G2 g/m^2 – 40 mm	G2 g/m^2 - 40 mm					
Strain gauge #		//		**1**	**2**	**3**	**4**	**5**	**6**
FEM	450	350	165	4.6	6.9	4.6	-4.9	-7.5	-4.9
Experimental	420	380	180	4.5	7.9	5.3	-4.8	-7.0	-5.3

4. Conclusions

The comparison between experimental data and numerical calculation can be considered very good, despite significant nonlinearities of the system. The idea to compare deflections by videos and strain deformations by strain gauges was useful to have a better understanding of the phenomenon, resulting in a good setting of the numerical method. As a summary, Experimental Modal Analysis has been carried out to obtain a first characterization of materials by determination of natural frequencies and to define the time step for the dynamic calculation. After this, a shock test has been performed using Medium Weight Shock Machine to verify the shock response of naval ship construction fiberglass laminates, while a Finite Element model is built to reproduce this Shock Machine behaviour. In the end, the comparison of experimental and numerical results was realized to validate the model and characterize the mechanical properties of composites. The choice to use mechanical properties of the materials achieved by EMA analysis to characterize materials in the dynamic numerical method seems to be acceptable to predict the shock transient response of the structures as shown in the numerical vs experimental comparison. This numerical method can be used not only to verify the shock response of plane laminates, but also to design more complex naval structures. Further tests are necessary and are indeed planned to study the composite damage mechanisms and its predictability using numerical methods.

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