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Experimental and Computational Study of the Response of Composite Structures to Extreme Loading

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EXPERIMENTAL AND COMPUTATIONAL STUDY OF THE RESPONSE OF
COMPOSITE STRUCTURES TO EXTREME LOADING

BY

ERIN GAUCH

A DISSERTATION SUBMITTED IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE DEGREE OF

DOCTOR OF PHILOSOPHY

IN

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UNIVERSITY OF RHODE ISLAND

2016

DOCTOR OF PHILOSOPHY DISSERTATION

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ABSTRACT

Fiber reinforced composite materials offer a variety of advantages in marine applications. They are corrosion resistant, require minimal maintenance, and offer a high strength to weight ratio. Additionally, they can be used to create complex geometries, can be tailored for optimal mechanical performance and are often inexpensive to produce and work with. Because of these advantageous properties they have been employed in a variety of settings, both military and commercial.

When employed in harsh environments, from the battlefield to the marine oil field, structures built from composite laminates may be subjected to dynamic events such as underwater explosive loading, both close-in and far-field, as well as overwhelming hydrostatic forces which could lead to implosion. In order to protect against these dynamic events composite structures are often over designed and the weight savings that composites offer goes unrealized. The confident use of composites in harsh marine environments requires the ability to predict their response to an array of severe loading conditions. The goal of this study is a better understanding of the response of composites to extreme loadings and computational tools and methods to predict these events.

First, the effects of preload on the response of flat composite plates to underwater explosive loading were investigated via computational simulations. Three preload conditions were investigated: directly applied compression, indirectly applied compression, and directly applied tension. Preload effects were assessed through comparison of material damage, delamination evolution and center point displacement. The primary effect of the preload is seen in the time required for the plate to recover from

the displaced shape. Little effect was observed on the amount of damage and delamination.

The second focus of this study was on the computational simulation of the implosion of composite cylinders composed of differing materials, Carbon/Epoxy and E-glass/Polyester. Simulations were built using the Dynamic System Mechanics Advanced Simulation software suit developed by the Naval Surface Warfare Center, Indian Head. Predicted dynamic pressures in the surrounding fluid were compared with experimental results from previous studies. Damage evolution in the simulations was also compared with experimental observations. It was found that the material model employed was not capable of predicting the damage evolution in the cylinders, however, pressure predictions for the initial collapse phase provided a reasonable correlation with measured data.

The third phase of this study was an experimental investigation of the response of composite cylinders with and without polyurea coatings to near field underwater explosive loading. Two coating thickness were investigated (100% and 200% of composite thickness) and each cylinder configuration was subjected to explosive loading at two different charge standoffs, 2.54 cm and 5.08 cm. The responses of the non-charge side of the cylinders were compared as well as damage sustained by the cylinders. It was found that the coatings had a slight effect on the response of the cylinders but significantly reduced the sustained damage.

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Through many late nights and holiday weekends spent studying and working my family has been at my side every step of the way. I would like to thank my husband William Gauch for going above and beyond any reasonable expectation to support my academic pursuits. He has made this all possible. My children, Laura, Claire and David have kept me laughing throughout and provide more support, encouragement and perspective than I could ask for. My parents, David and Lydia Bergeron, have managed to push and pull me through the hardest moments by having an unshakeable faith in me and an endless font of love and encouragement.

I feel truly privileged to have had the opportunity to work with my fellow graduate students from the dynamic photo-mechanics laboratory over the years. Professor Shukla has assembled and maintained an incredibly talented and supportive team. I have made lasting friends and learned so much. In particular I would like to thank my good friend Dr. James LeBlanc who blazed the trail and helped guide and encourage me through this path. I would also like to thank Michael Pinto for his help and patience in

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DEDICATION

This dissertation is dedicated to my mother, Lydia Bergeron, and my late grandmother-in-law, Genevieve Mathison; both remarkable role models.

PREFACE

This dissertation is prepared using the manuscript format.

Chapter 1 presents a computational study of the effects of preload on the response and damage of thin, flat composite plates to underwater explosive loading. Three types of in-plane preload are investigated: (1) directly applied compression, (2) indirectly applied compression, and (3) directly applied tension. Preload effects are assessed through comparison of material damage, delamination evolution and center point displacement. The primary effect of the preload is seen in the time required for the plate to recover from the displaced shape. Little effect was observed on the amount of damage and delamination. This chapter has been published in *Computers and Structures*.

Chapter 2 investigates the use of the Dynamic System Mechanics Advanced Simulation fluid-structure interaction code to predict the dynamic pressure and damage evolution during the implosion of Carbon/Epoxy and E-Glass/Polyester cylinders. Finite element models were built and simulations were conducted to model experimental test results from previous studies. For both cylinders considered the simulations failed to accurately capture the damage evolution in the collapsing structure, however, the simulations provided a reasonable envelope of pressures in the local fluid for the underpressure region. This chapter has been prepared for submission to *Computers and Structures*.

Chapter 3 presents the experimental study of the response of composite cylinders with and without polyurea coatings to near field underwater explosive loading. Cylinders were prepared as-is and with thin (100% composite thickness) and thick (200% composite thickness) spray cast polyurea coatings. Each cylinder configuration was

subjected to near field loading at charge standoffs of 2.54 cm and 5.08 cm. Deflection of the center line and center point on the non-charge side of the cylinders were used to evaluate the effects of the coatings as well as observations of the damage sustained by the cylinders. It was found that the polyurea coatings had a slight effect on the response of the non-charge side of the cylinder, however, damage was dramatically reduced. This chapter is prepared for submission to Experimental Mechanics.

Chapter 4 provides an overview of the conclusions drawn from this work as well as proposals for future work.

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CHAPTER 1

RESPONSE OF PRELOADED THIN COMPOSITE PANELS SUBJECTED TO UNDERWATER EXPLOSIVE LOADING

By

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Abstract

The effect of preloading on thin composite plates subjected to underwater explosive loading has been studied through computational simulations. In this study the effects of three types of in plane preloading are considered: (1) directly applied compression, (2) indirectly applied compression, and (3) directly applied tension. The effects of the preloading conditions are assessed using the plate center point deflection, material damage, and delamination evolution. The results show that for thin composite plates subjected to underwater shock loading conditions there is a minimal effect of preload on the response of the plates or the amount of damage and delamination sustained.

Keywords: Composite Materials, Composite Damage, Underwater Explosion, Preload, Buckling

1. Introduction

Composite materials offer several advantages in design including a high strength to weight ratio and an ability to be tailored to a specific application. For this reason composites have found use in a large number of industries, including consumer, aerospace, and military applications. There is a current desire within the United States Navy (USN) to make use of composite materials in the production of vehicle hulls and structural bodies in order to exploit the advantageous properties of these materials. In order for this desire to be realized there must be a clear understanding of how composite materials perform in environments such naval structures are exposed to including underwater explosive (UNDEX) shock loading. As naval applications are expanded to include submersible vehicles the structures are subjected to hydrostatic preloads due to depth pressure. These preloads are likely to influence the response of the structure to any other types of loading conditions. This study serves as a first step in characterizing the

effects of preload on glass reinforced composite materials subject to UNDEX loading.

To avoid overdesign and realize the full potential of this material class it is necessary to be able to predict, with confidence, the effects such preloading conditions will have on survivability.

The response of materials subjected to shock loading has been studied over a wide range of loading rates. The effect of shock loading on stainless steel plates subjected to underwater impulsive loads has been presented by Espinosa et al. [1]. Nurick et al. [2,3] have studied the effects of boundary conditions on plates subjected to blast loading and identified distinct failure modes depending on the magnitude of the impulse and standoff. The response of E-Glass and Carbon based composite laminates under shock and explosive loading (including the effects of heat generation during combustion) has been presented by Tekalur et al. [4]. Mouritz [5] studied the effectiveness of adding a light weight, through thickness stitching material to increase the damage resistance of composites. LeBlanc et al. [6] have studied the effects of shock loading on three-dimensional woven composite materials. Recently, there has been an increased interest in the study of the effect of shock loading on sandwich structures. These studies include the effects of shock and impact loading conditions (Jackson et al. [7], Schubel et al. [8], Arora et al. [9]).

Robb, Arnold and Marshall [10] conducted a series of experiments investigating the effect of preload conditions on chop strand mat composite panels. The preload conditions investigated included uniaxial and biaxial tensile and compressive preloads as well as combined tensile/compressive loading. Although the damage area of the composite was not significantly affected for most loading cases Robb, et al., found that

the loading condition did affect the shape of the damage. Whittingham, et al., conducted a study of prestrained 8-ply, quasi isotropic carbon/epoxy laminate panels [11]. In order to assess their study Whittingham, et al., determined impact force, absorbed energy and penetration/perforation depth for each panel. It appears that the preload conditions have very little effect on the observed peak impact force. Zhang, et al. [12], studied the effects of compressive preload on damage area and post-impact compressive strength of quasi-isotropic carbon/epoxy laminate panels loaded at and above their critical buckling loads. In this study it was found that there was no significant correlation between level of preload and damage area post-impact. Herzberg and Weller, [13], conducted a similar study of post-buckled 2 mm thick woven glass composite panels which were subjected to various impact energies. Preload did have an effect on the critical velocity of impact; increasing preloads corresponded with decreasing critical velocities. This relationship was particularly pronounced for the specimens impacted on their convex faces. Wiedenman and Dharan [14] describe their study on compressively loaded E-Glass composite plates (1.6mm-6.4mm in thickness) subjected to ballistic penetration. This study found that the ballistic limit was reduced with increasing preload. Wang and Shukla [15] studied the dynamic response of sandwich composite constructions with initial compressive preloads subjected to blast loading. This study showed that as the in-plane compressive loading was increased, there was a corresponding increase of face sheet damage, out of plane deflections, in-plane strains, and an overall reduction in the blast resistance of the panels. In studies of air-blasted aluminum plates Veldman et al. [16, 17] found little effect of pre-pressurization on specimens subject to elastic or plastic deformation. At blast pressures high enough to induce failure in the plates the magnitude

of the preload affected the threshold of failure onset and the extent of the corresponding damage.

There are a limited number of publications detailing the prediction of buckling and the effects of preload on the shock response of composites. Bao et al. [18] compiled numerous analytical solutions for the buckling prediction of orthotropic rectangular plates from literature and developed corresponding finite element models for comparison. It was found that finite element models can accurately predict the buckling behavior of orthotropic plates as compared to analytical for thin plates but divergence between the solutions occurs as the plates become thicker. Mikkor et al. [19] computationally investigated the effect of tensile preloads on carbon/epoxy panels subjected to impact loading with comparison made between preloaded and non-preloaded configurations. In the study it was found that there is only a small effect of preload on the material damage extents up to a certain critical impact velocity. Although both plates sustained catastrophic failure above a critical impact velocity, the non-preloaded plates displayed an increase in damage area prior to failure, while the preloaded plates did not display increased material damage prior to catastrophic failure. Khalili et al. [20] developed analytical solutions for the prediction of composite plate impact response under initial tensile loading. This study showed that as tensile preloads are increased, there is a corresponding increase in impact forces but a reduction in center point deflection.

2. Composite Material Model

The composite material used in the simulations presented in this paper is representative of Cyply 1002, a cured epoxy composite reinforced with continuous E-Glass fibers. This material was utilized in the experimental work [21] which serves as

the foundation for the current study. The material is of a bi-axial construction (cross ply) with fiber orientations of 0° and 90° . The simulated plates are 26.54cm (10.45 in.) in diameter and have a total thickness of 3.3 mm (0.130 in.). They are comprised of thirteen individual layers of alternating 0 and 90 degree plies with a per ply thickness of 0.254 mm (0.01 in.). The model of the composite plate consists of seven layers of solid brick elements of constant stress formulation. Each layer, with one exception, represents a 90° and a 0° ply and is 0.508 mm (0.02 in.) in thickness with a single element in the through thickness direction. The central layer represents a single 0° ply which is 0.254 mm (0.01 in.) thick. The material properties are listed in Table 1 and were previously determined through testing per ASTM specifications.

The elements comprising the composite plate are 0.508 mm (0.02 in.) in the through thickness direction and 0.254 mm (0.01 in.) in the in-plane directions. The choice of 0.508 mm (0.02 in.) for the through thickness direction corresponds to the thickness of a combined $0/90$ degree ply, from which the in-plane element lengths were derived such that an aspect ratio of 5:1 was obtained in keeping with modeling best practices for element quality. An outer ring of nodes, 3.68 cm (1.45 in.) in thickness, on the first and last ply are constrained in the normal direction to simulate the clamped condition of the tested specimens in Reference [21]. This boundary condition restricts out-of-plane motion of the composite plate while allowing in-plane motion between layers. Through thickness bolt holes are included in the modeled plate to maintain geometric consistency with the validated model of Reference [21], however, no bolts are modeled and the bolt holes remain unconstrained for this study. The unsupported area of the modeled plate is 22.8 cm (9 in.).

The material model utilized is the Mat_Composite_Failure_Option_Model (MAT_059, Option=solid) provided by the explicit finite element code LS-DYNA. This material definition is capable of modeling failure due to inplane compression / tension and shear with different failure parameters for tension and compression. Once a given failure criterion has been met the ability of the material to carry a load in that direction is eliminated. An element is only deleted once it has failed in all directions.

Interlaminar failure, or delamination, plays a significant role in energy absorption and degradation of a composite's stiffness during impact and must be accounted for in simulation. The approach taken in this study, and previously verified [21], is to use a surface-to-surface tie break contact in the implementation of the finite element code. This contact definition ties the nodes between plies together rather than making them equivalent. This inhibits relative sliding until the normal or shear stress at any given node exceeds a defined failure value. Once this value is exceeded the node becomes free to slide and the contact reverts to a standard definition. This allows the slave node to separate from the master surface but does not allow it to pass through. In the current model the choice of a delamination criterion was taken to be 34.4 MPa (5000 lb/in²) for both tensile and shear stresses. This value represents one-half of the tensile strength of the pure epoxy resin. The degradation by ½ of the tensile strength accounts for voids and interfacial defects / flaws between the layers of fibers during the manufacturing of the material. This value was determined by prior parametric studies as well as discussions with other experts in this field.

3. Loading Conditions

3.1 Preload Configurations

This study examines the effects of three distinct types of preloading conditions: (1) direct compressive preload, (2) indirect compressive preload, and (3) direct tensile preload. The direct compressive and tensile preload cases represent loading configurations in which the structure remains loaded (follower load) even while undergoing deformation. An example would be a submerged structure that is continually acted upon by depth pressure while undergoing deformation due to additional applied loads (ie shock / impact). The indirect loading case represents a configuration where the structure is compressed by a loading mechanism which applies a prescribed displacement but is otherwise unattached to the structure. Therefore if the test article undergoes a flexural deformation it can separate from the loading fixture, and effectively release the preload.

Table 1 - Cyply 1002 cross ply – mechanical properties from [15]

	N/m ² (lb/in ²)
Tensile Modulus (0°)	22.7e9 (3.3e6)
Tensile Modulus (90°)	20.0e9 (2.9e6)
Tensile Strength (0°)	482e6 (70e3)
Tensile Strength (90°)	482e6 (70e3)
Compressive Strength (0°)	689e6 (100e3)
Compressive Strength (90°)	689e6 (100e3)

The preloads which are presented in the following work are functions of the compressive buckling load of the plates. As the plates are made from a composite material with orthotropic material properties there is a limited amount of literature that present closed form solutions for the buckling load. Those that are available present a

wide range of solution methods which yield differing results. Therefore, in order to determine the appropriate buckling load of the plates, a computational methodology was utilized. LS-DYNA has the capability to predict buckling loads for given load configurations (boundary and loading conditions) by applying the desired load configuration but at a level lower than the expected buckling load. The code then implicitly solves this load state and determines lagrangian multipliers which yield the buckling load as a function of the applied load. This approach was validated by comparing the results of the buckling analysis with closed form solutions present in the literature, [22-23], for isotropic plates, both round and rectangular. The difference between the closed form solutions and the analyses were found to be small (<5%) and allowed confidence that LS-DYNA was capable of accurately predicting the buckling load for the current loading configurations. Further, a study by Orifici, et al. [24], compared experimentally obtained buckling loads and post-buckling stiffness properties of fiber reinforced composite panels with those from several finite element codes, including LS-DYNA. The comparison showed good correlation between the predicted and actual buckling load and an acceptable correlation with post-buckling stiffness. Therefore, this method was deemed acceptable for the composite plate. The radially compressive load which initiates buckling is 55,360 N/m (320 lb/in). Using this buckling load as a basis, simulations were run at both lower and higher preload values to determine the effects on the plate response. Table 2 lists preload values which are presented in this work, both in terms of load per unit edge length and as a percentage of the compressive buckling load. Load values above the critical buckling load are studied to determine the effect of shock loading on structures that have buckled under load and

are no longer in their baseline shape. For the present study this corresponds to deviations from planarity. The tensile preloads are based on the compressive buckling load simply for comparative purposes, although it is noted buckling would not be present under these conditions.

Table 2 – Plate Preload Values

Edge Load N/m (lb/in)	% Buckling Load
0 (0)	0
5,536 (32)	10
11,072 (64)	20
16,608 (96)	30
22,144 (128)	40
27,680 (160)	50
55,360 (320)	100
83,040 (480)	150
110,720 (640)	200

The preloads were applied to the plates utilizing two different methods. For the case of the direct (follower) loading in compression and tension, a pressure loading was applied to the outer edge of the plate. This is possible because the plate is modeled using solid elements and therefore has an outer surface. By applying the load as a pressure it ensures that the load will act on the plate edge even while undergoing flexural deformation during the shock loading. For the case of the indirect (non-follower) preloading a ring of thermal material was added to surround the outer boundary of the plate and a contact defined between the two parts. Utilizing a thermal material model for the ring allowed it to be thermally contracted, effectively applying a radial preload to the plate. No friction is defined for this contact to avoid inducing a shear load. The correct preload value for the thermal contraction was determined by running a parametric study in which the stress state and radial deformation induced by the thermal ring was compared to the direct pressure load results until 1:1 correlation was obtained for each

preload value. This ensures that prior to the arrival of the shock front each plate is under equal conditions. The thermal ring was held fixed after the preload application to prevent further contraction and a resulting follower load on the plate.

3.2 Preload Results

The stress state in the composite plate that results from the application of the preload is of important consideration. These stresses represent the plate configuration prior to the arrival of the shock front and to which any stresses resulting from the shock response itself will be added or subtracted through effective superposition. The stress state resulting from the direct compressive preload just below the buckling load is shown in Figure 1. The stress states vary slightly in terms of vertical and horizontal components because the plate is slightly stiffer in the X direction due to the additional ply in this direction. It is seen that globally the peak stress is approximately 20.7 MPa (3,000 psi), with very localized higher stresses on the order of 34.4 MPa (5,000 psi) around the bolt holes, which act as geometric stress raisers. The tensile and compressive strengths of the material are 482 and 689 MPa (70 and 100 ksi) respectively. Therefore, at the onset of buckling, the stresses in the plate are significantly below the strength of the plate. This can be attributed to the “thin” nature of the plate, as it has an unsupported diameter to thickness ratios of 70:1 (228 mm/3.3 mm). With this observation of the stress at the onset of buckling, it is clear that the plate can sustain significant additional stresses resulting from the shock loading to be imposed. Furthermore, the stresses resulting from the preload would represent only a small portion of the failure stress.

Figure 1 – Stress State at the Onset of Buckling (Units of MPa)

4. Conical Shock Tube

The experimental results which act as the validation for the computational model were obtained through the use of a conical shock tube (CST) facility located at the Naval Undersea Warfare Center, Division Newport. Although the tube is not utilized in the current study a brief overview is provided for background. The shock tube is a horizontally mounted, water filled tube with a conical internal shape, Figure 2. The tube geometry represents a solid angle segment of the pressure field that results from the detonation of a spherical, explosive charge. In an open water environment the pressure wave expands from the charge location as a spherical wave. In the shock tube the rigid wall acts to confine the expansion of the pressure wave in a manner that simulates a conical sector of the pressure field.

Figure 2 – Conical Shock Tube Schematic (not to scale)

The internal cone angle of the tube is 2.6 degrees. The tube is 5.25 m (207 in.) long from the charge location to the location of the test specimen and internally contains 98.4 L (26 gal.) of water at atmospheric pressure. The pressure shock wave is initiated by the detonation of an explosive charge at the breech end of the tube (left side of figure) which then proceeds down the length of the tube. Peak shock pressures from 10.3 MPa (1500 lb/in²) to 20.6 MPa (3000 lb/in²) can be obtained depending on the amount of explosive charge. A typical pressure profile obtained from the use of the tube is shown in Figure 3. This figure illustrates the rapid pressure increase associated with the shock front followed by the exponential decay of the wave. This profile was obtained using a M6 Blasting Cap – 1.32g (.00292 lb) TNT Equivalency and is measured 0.508 m (20 in.) from the impact face of the test specimen. The length of the tube is sufficient so that plane wave conditions are nearly established at the test specimen.

Figure 3 - Typical Shock Tube Pressure Profile

A mounting fixture has been designed so the test specimens are air backed with fully clamped edges. The specimens are 26.54 cm (10.45 in.) in overall diameter with a 22.86 cm (9 in.) unsupported middle section. The mounting arrangement that was used in the experiments is shown in Figure 4.

Figure 4 – Experimental Test Setup

5. Finite Element Model

5.1 Overview

This study makes use of, and further extends, the computational models that were previously developed and correlated to experimental test data in Reference [21]. The complete finite element model is shown in Figure 5. The model consists of the internal fluid of the shock tube and the composite test sample. No numerical damping has been applied to the model. The fluid within the tube is considered in the simulation so as to capture the fluid structure interaction (FSI) at the interface of the fluid and test plate. Only the first 1.01 m (40 in.) of the fluid extending from the test sample towards the charge location are modeled. This was deemed to be acceptable for 2 reasons: (1) the fluid is loaded with the pressure profile measured 50.8 cm (20 in.) from the test sample and (2) a non-reflecting boundary layer is applied at the charge side boundary of the fluid domain. The fluid is modeled with solid elements and a null material definition. The fluid mesh uses a global element size of 6.35 mm (0.25 in.). The use of the null material allows for the fluid to be defined with an equation of state (EOS) definition. The interface between the fluid and the composite plate is modeled through a tied-surface-to-surface contact definition (LS-DYNA keyword *Contact_Tied_Surface_To_Surface). The contact surfaces are defined and the nodes are tied together. This method ensures accurate load transfer between the materials without the need for node equivalencies at the fluid surface.

Figure 5 - CST Finite Element Model

The pressure load is applied as a plane wave at the location of the test pressure transducer. The pressure profile used in this study corresponds to those measured during testing; a typical profile is shown in Figure 3. The simulations for this study, as well as that mentioned above, were all performed using the LS-DYNA finite element code produced by Livermore Software Technology Corporation (LSTC), Version 971 Release 3.1, run in double precision mode.

Beyond the application of the preload in the current study, there is one difference in the boundary conditions between the current model and the model previously validated. In the prior model the inner surface of the bolt holes were held fixed to simulate the bolts that were used during the test. In the preload simulations for this study these bolt holes are not constrained. This is chosen to simulate conditions that would be present during a laboratory preload test in which the plate would be radially compressed to varying levels and as such any bolt holes would align differently for each preload value. Additionally, any constraints or representative bolts would bear a portion of the

preload and prevent even preloading of the plate. Therefore, the bolt holes are left in the plate geometry however it is assumed that the bolts themselves are not put in place.

5.2 Validation

For the case of non-preloaded plates this model and methodology has been shown [21] to be able to accurately capture both the transient response of the plate as well as the final post-mortem damage state of the composite. Figure 6 shows a comparison of the simulation to strain gage data obtained during testing on the back face of the panels. Furthermore, the comparison of final damage levels is shown in Figure 7. These comparisons show that the model is able to simulate the test results for the case of the shock loaded plate with no initial preload, and as such allows the model to be extended to include the preloads with a high level of confidence in the results.

Figure 6 – Strain Gage Correlation

Figure 7- (a) Material Damage during Test, (b) Material Damage from Simulation

6. Results

For each of the three preload conditions that are studied in this work the preload effect is assessed using transient and post mortem measurements. Specifically, comparisons are made between the transient center point displacement, and the evolution of both in-plane material damage and through thickness delamination.

6.1 Direct Compression (Follower Load)

The center point displacements of the back face of the plates for each direct compression preload value are shown in Figure 8. Time zero in this figure corresponds to the initialization of the pressure wave 0.5 m in front of the plate. In this figure it is important to point out that for the cases of 100% and 200% buckling load there is an initial displacement prior to the arrival of the shock front. These initial displacements indicate the onset of buckling and show that the code is accurately predicting the buckling behavior. For both cases the plates buckle away from the fluid resulting in a concave wetted surface as viewed from the fluid. From the displacement profiles for the

plates preloaded up to and including the buckling load it is seen that there is no preload influence on the peak center point displacement, with all plates displacing approximately 17.8 mm (0.7 in.) out of plane. There is a difference however in the rate at which the plates attempt to recover to their un-deformed shape. As the preload increases the plates are slower to recover. A further observation is that none of the preloaded plates are able to fully recover to their initial planar (flat) geometry. The zero preload plate is able to recover but after a long duration of time beyond the range of the current plot. The preloaded plates are not able to do so because as the plate is deforming in a flexural manner the preload is simultaneously acting to reduce the outer diameter of the plate and thus restricting the ability to return to its initial state. This is further emphasized for the plates which are preloaded to 40, 50, and 100 % buckling load respectively in that these plates suffer complete failure (see later discussion) during the rebound phase of the deformation due to the resisting action of the preload. For the case of the plate preloaded to double the buckling load the plate does not suffer failure but does have a large permanent deflection more than double the value of the initial buckling induced displacement. This indicates there is likely a large stress relief once the plate is sufficiently buckled prior to the arrival of the shock front.

Figure 8 – Center Point Out of Plane Displacement – Direct Compressive Preload

Figure 9 compares the damage evolution for directly applied (follower) compressive preload levels of 0, 20, 40, 50, 100 and 200% buckling load. It is noted that although the plate geometry is radially symmetric, the material is not. The fibers are running in both the vertical and horizontal directions (0 and 90 degrees respectively), with the 0 degree fibers passing in line with the top and bottom holes. In the horizontal direction there is no hole and there is no corresponding stress concentration. For all preload values the damage initiates from the top and bottom holes in the form of longitudinal cracks propagating towards the center of the plates. In the 40 % load case the development of circumferential cracks is present, which is not seen in the lower preload values. These circumferential cracks are not seen in the plate loaded to the buckling load but the radial cracks are much more pronounced. For the case of the plate preloaded to double the buckling load there is much more material damage, primarily in

the form of cracking, and propagation from the bolt holes. At load levels beginning at 40 % buckling it is observed that there is catastrophic plate damage with total plate failure. The onset of total plate failure occurs later in time while the plate is returning to its original planar shape and not during the initial plate flexure. This can be attributed to a reduction in plate outer diameter due to the nature of the follower load. While the plate is undergoing flexure due to the initial shock loading it tends to draw material towards the center, thus reducing the outer diameter of the plate. Since the pressure is directly applied, it serves to hinder the plate's ability to return to its original planar shape and thus the stress state is increased during the rebound phase of the deformation. Therefore, at lower levels the plate is able to absorb some of this additional compression load but is not able to do so at the higher preload values. In general it is shown that there is not a significant effect of preload on the damage that the plate sustains during initial deformation, but there is a large effect on the plate's ability to return to its initial configuration.

Figure 9- Element Damage – Direct Compressive Preload

The evolution of the delamination damage for the directly applied compressive preload is shown in Figure 10. From this figure it can be seen that there is a minimal effect of compressive preload on the amount of delamination present. It is also observed that the delamination primarily occurs during the initial flexure of the plate and not during the recovery phase. For the preload values up to and including the buckling load the delamination propagates from the edges of the plate towards the center, with nearly equivalent extents as measured radially from the plate edge. It is noted that for the case of the plate that was subjected to double the buckling load the delamination is less than that of the plates which were planar at the time of arrival of the incident shock. Based on these observations there is minimal effect of preload on the amount of delamination damage. These results observed for directly applied compressive preloads are consistent with those outlined in the literature review. The damage and delamination areas were not significantly affected by the level of preload.

Figure 10 - Delamination – Direct Compressive Preload

6.2 Indirect Compression (Non-Follower Load)

In addition to the directly applied compressive preload which acts as a follower load during plate deformation, the case of a non-follower preload is also considered. As previously discussed in detail this approach employs a ring of thermal material which is contracted to impart a compressive preload to the plate. Once the ring is contracted to obtain the desired preload the nodes are fixed, preventing any further contraction while the plate undergoes deformation due to the shock loading. As a result of this, the plate is free to separate from the thermal ring, effectively releasing the preload during deformation.

Figure 11 illustrates how the plate is free to separate from the thermal ring during deformation. In the first image it is seen that the plate is in contact with the ring prior to the arrival of the shock front, however once the wave loads the plate and causes flexural deformation, the plate quickly pulls away from the ring, and effectively releases the preload. As the plate returns to its initial configuration it again comes into contact with the ring. This loading case is meant to highlight loading conditions where preloads are applied by mechanical means in which the loading fixture is not able to respond as quickly, if at all, to deformation modes which act to draw material away from the fixture.

Figure 11 - Gap between Composite Plate and Loading Ring During Peak Deflection

For the indirect compressive preload, the center point displacement time histories are shown in Figure 12. In this figure it is once again seen that for the case of a preload 200% of the buckling load, there is an initial out of plane displacement corresponding to the buckled shape, and is oriented away from the fluid. For this loading type (non-follower) it is seen that for all preloads, up to and including the buckling load, the peak center point deflection is the same. Furthermore the recovery time and speed is the same regardless of preload value. As previously discussed, for this preloading case the plate is able to separate from the preloading ring and once it does the preload is essentially removed. Therefore it is expected that each of the plates would respond in a similar manner once separation from the preloading fixture occurs. This highlights a consideration that must be made when designing laboratory experiments of this nature. Mainly that although a preload is initially applied to the plates, if the fixture allows for releasing of the load then the true effect of preload may not be realized.

Figure 12 - Center Point Out of Plane Displacement – Indirect Compressive Preload

Figure 13 compares the damage evolution for indirectly applied (non-follower) compressive preload levels of 0, 20, 40, 50, 100 and 200 % buckling. As was seen in the directly applied preload case the damage initiates from the holes at the top and bottom of the plate where the fibers correspond to the radial direction. However it is observed that the damage states are nearly equivalent for all of the preload values including the pre-buckled plates. This is expected as once the plates separate from the preloading ring and release the preload, the plates should deform similarly. It is also noted that none of these plates suffer from large scale failure during the recovery phase of the deformation. For the case of the follower preload conditions this plate failure was attributed to a reduced outer plate diameter caused by constant compression of the preload during deformation. In the current non-follower preload condition, there is no such restriction on the ability of

the plate to return to its shape prior to the arrival of the shock front. This further highlights the differences in structural response between a plate that is preloaded with a follower load as opposed to a non-follower load when the primary deflection mechanism is flexure.

Figure 13 - Element Damage – Indirect Compressive Preload

The evolution of the delamination damage for the indirectly applied compressive preload is shown in Figure 14. Similar to the case of the direct preload there is a negligible effect of preload on the extent of the delamination. The delamination primarily evolves during the initial flexure of the plate with no additional evolution during the recovery phase of the motion. For the preload values up to and including the buckling load the delamination propagates from the edges of the plate towards the center, with nearly equivalent extents as measured radially from the plate edge. As was observed for the case of the direct preload, the plate that was initially buckled (200 % buckling load) suffers significantly less delamination than the plates which were planar at

the time of arrival of the incident shock. Based on these observations there is minimal effect of preload, up to buckling, on the amount of delamination damage.

Figure 14 - Delamination – Indirect Compressive Preload

6.3 Direct Tension (Follower Load)

The center point displacement of the back face of the plates for each tensile preload value is shown in Figure 15. As opposed to the previous two compression preload cases none of the plates sustain initial displacements prior to the arrival of the shock fronts in the form of buckling. It is observed from the figure that there is a clear influence of tensile preloads on the peak transient deflections during shock loading. As the preload value is increased the peak deflection is decreased. The baseline panel (zero preload) has a peak displacement of 17.8 mm (0.7 in.), and then as the preload increases the deflection decreases to a value of 15.2 mm (0.6 in.) for the 150% tensile preload. This trend is expected, as under tensile loading the plate effectively becomes “stiffer” with the tensile loading resisting the ability of the plate to deform in a flexural mode in response to the shock loading by drawing material in from the edges. Similarly the rate

at which the plate tends to recover is affected by the preload level. Using the zero preload as a baseline, the recovery time is decreased (occurs faster) as the preload is increased. This implies that the restoring forces become larger with increasing tensile preload.

Figure 15 - Center Point Out of Plane Displacement – Direct Tensile Preload

The evolution of damage for the composite plates preloaded in tension is shown in Figure 16. From this figure it is shown that as the tensile preload is incrementally increased, there is a corresponding decrease in material damage. For the baseline (zero preload) and 20 % preload the damage is similar to the compressive preload cases in that it evolves as radial cracking initiating at the top and bottom through holes. However, at preloads above the 20 % level, there is no material damage present due to the initial shock load. This observation indicates that the higher tensile preloads are offering a level of protection to the plate that was not observed in the compressive preloading. It is noted

that the plate suffers complete failure later in time for the 150 % tensile preload. This indicates that the protective effects of the tensile preloads are only present up to a certain point and then they become destructive; the higher preload and applied shock load superimpose and overwhelm the material.

Figure 16 - Element Damage – Direct Tensile Preload

The delamination damage for the tensile preloading condition is shown in Figure 17. As seen in this figure there is a reduction in the amount of delamination damage when a tensile preload is applied as compared to the plate with no preload. The delamination levels do tend to be consistent for the plates with tensile loading, regardless of the preload. These delamination results are in agreement with the material damage results in that there appears to be some level of protection offered by a tensile preload (up to a critical level at which plate failure occurs) as compared to a compressive preload.

Figure 17 – Delamination – Direct Tensile Preload

7. Summary and Conclusions

The effect of preloading conditions on thin composite plates (70:1 D:t ratio) subjected to underwater explosive loading conditions has been studied through computational simulations. The simulations presented in the study are an extension of previous work in which the simulation methodology was validated against test data. The model has previously been shown to be able to simulate both the transient response, as compared to strain gages, and the damage evolution and final state as compared to post mortem observations of the test article. Therefore the model is considered validated for the particular cases that are studied in the current effort. In this study the effects of three types of planar preloading are considered: (1) directly applied compression, (2) indirectly applied compression, and (3) directly applied tension. The preloads are chosen to be a function of the compressive buckling load of the plate. The effects of the preloading conditions are assessed using the plate center point deflection, material damage, and delamination evolution.

The effects of compressive preloads have been evaluated using both direct and indirect load application. In both cases the preload was found to have little effect on the peak displacement achieved under shock loading. However there is a difference in the rate of recovery for the two conditions. In the case of direct loading, as the preload was increased the rate of recovery was slowed down, meaning it took longer to recover. For indirect loading the plates recovered at the same rate regardless of preload. Additionally, for both types of preload there appears to be little effect on the level and type of damage that develops during the initial plate response. There are differences in the evolution of damage later in the event, primarily during the recovery phase. When the preload acts as a follower load, the plates sustain more damage as the preload is increased, eventually resulting in complete failure at higher levels. This is likely due to additional compression occurring while the plate deforms out of plane. This does not occur for the indirect preload as the ring used to apply the preload is held fixed and the plate is free to return to its initial preload configuration. For both cases the extent of delamination was nearly equivalent and is sustained during the initial flexure of the plates.

In addition to the compressive preloads, the effects of a tensile preload have been assessed. Tensile preloading was found to reduce the peak center point deflection during shock loading. As the preload is increased the plates have an increased resistance to deformation, essentially they become stiffer as the tensile load is increased. A similar effect is seen in the damage evolution of the plates in that there appears to be a protective effect as the preload is increased. As a larger preload is applied the amount of damage

sustained during the shock response is decreased until a critical value is exceeded and the protective effect is negated.

Overall, it has been shown that for thin composite plates subjected to underwater shock loading conditions there is a minimal effect of preload on the response of the plates. The primary effect is seen in the rate of recovery of the displaced shape. There is minimal effect on the amount of damage and delamination sustained; however there does appear to be a slight protective effect from moderate tensile preloading.

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Chapter 2

DAMAGE EVOLUTION AND DYNAMIC PRESSURE PREDICTIONS IN THE IMPLOSION OF CARBON/EPOXY AND E-GLASS/POLYESTER COMPOSITE CYLINDERS: A COMPUTATIONAL STUDY

By

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Abstract

The exposure of maritime structures to overwhelming, uncompensated hydrostatic pressure can lead to implosion of the structure and potentially harmful effects for nearby systems. The ability to predict these effects is important in the confident use of structures that may be vulnerable to implosion. In this study the use of computational analysis to predict the dynamic pressure and damage evolution during the implosion of Carbon/Epoxy and E-Glass/Polyester cylinders has been studied through the use of the Dynamic System Mechanics Advanced Simulation fully coupled fluid-structure interaction finite element code. Finite element models were built and simulations were conducted to model experimental test results from previous studies. For both cylinders considered the simulations failed to accurately capture the damage evolution in the collapsing structure, however, the simulations provided a reasonable envelope of pressures in the local fluid for the underpressure region.

1. Introduction

Implodable volumes are defined as structures which are gas filled or have an internal vacuum exposed to large, uncompensated, external hydrostatic pressures. Many thin-walled maritime structures can be categorized as implodable volumes; i.e. unmanned underwater vehicles (UUVs), sensors, and deep ocean submersibles. Instability caused by material defects, intense hydrostatic pressure or transient loading (such as underwater explosive (UNDEX) loading) could initiate an implosion event and produce large pressure waves in the surrounding fluid. When deployed in the vicinity of a structure or vessel these implodable volumes pose a potential threat. There has been increasing interest in being able to define and predict this threat and employ smart design solutions to combat it.

Past implosion research efforts have included experimental testing of glass, metallic and composite implodable volumes [1]-[6]. Related computational efforts have focused on the implosion of glass and metallic volumes [1], [3]- [5]. Turner and Ambrico [1] have published work investigating the basic physics of the implosion of cylindrical aluminum tubes. They conducted experiments with implosion initiated by both hydrostatic pressure and a mechanical depressor. Near field pressure profiles were measured radially from the center of the cylinders at three points along the lengths. They found that during the initial collapse there was a decrease in the surrounding pressure. When the central portions of the cylinders were fully collapsed along the width they observed a dramatic peak in pressure. This was followed by a lower magnitude pressure pulse that continued until the cylinders were fully collapsed. In addition to their experiments Turner and Ambrico [1] conducted detailed fluid-structure interaction (FSI) simulations using the Dynamic System Mechanics Advanced Simulation (DYSMAS) finite element code developed by the Naval Surface Warfare Center, Division Indian Head. Simulation results provided an excellent correlation with test results in regards to the near field pressure profiles and the collapsed shapes of the cylinders. The detailed simulations, validated against test data, were used to illuminate the implosion process from initiation through total collapse.

There has been increasing interest in leveraging the advantageous properties of composite materials in the maritime community. Composites offer a high strength to weight ratio along with outstanding corrosion resistance and reduced maintenance requirements. However, the mechanisms of composite failure are quite different from those of ductile metals. Damage in composites can be described by three major failure

modes including delamination, matrix cracking and fiber breakage. Further, composites of various lay-ups and constituent materials will behave differently under similar loading conditions. This study investigates the use of computational methods to predict the failure by implosion of thin-walled, laminated composite cylinders and the resulting dynamic pressure in the surrounding fluid.

2. Experimental Procedure

The focus of the current study is primarily computational simulations, however a brief description of the experiments used for model comparisons is provided here. A more detailed account of the experimental procedures used by Pinto and Shukla and their results can be found in [7] & [8].

2.1. Materials

The composite materials modeled in this study include filament wound Carbon/Epoxy and E-Glass/Polyester laminates used in Pinto and Shukla's experimental studies of composite cylinder implosion [7] & [8].

The Carbon/Epoxy laminate has a $[\pm 15/90/\pm 45/\pm 15]$ layup as measured with respect to the longitudinal axis of the cylinder, beginning at the inner surface, and was manufactured using the filament-winding process by Rock West Composites of West Jordan, UT. The cylinders have a nominal wall thickness of 1.39 mm, an inner diameter of 76.4 mm, and an unsupported length of 279.4 mm ($L/D = 3.7$). The fiber volume ratio for the material is 60%. Material properties as provided by the manufacturer can be found in Table 1.

Table 1. Carbon/Epoxy Ply Level Tensile Material Properties

Orientation	Strength (MPa)	Modulus (GPa)
90°	81	9.2
0°	2572	137

The second laminate considered is an E-Glass/Polyester manufactured by Nor'Easter Yachts, Inc., of Milford, CT. The E-Glass/Polyester cylinders, also produced via the filament winding process, have a $[\pm 55/0]$ layup. The cylinder has a nominal thickness of 2.08 mm, outer diameter of 61.1 mm and an unsupported length of 381 mm ($L/D = 6.2$). Tensile material properties, as provided by the manufacturer, can be found in Table 2.

Table 2. E-Glass/PE Ply Level Tensile Material Properties

Orientation	Strength (MPa)	Modulus (GPa)
90°	40	13.25
0°	1020	46.04

The composite cylinders were capped on each end with aluminum end caps and were sealed using an epoxy compound to ensure water tightness.

2.2.Experiments

The implosion experiments were conducted in a large pressure vessel (2.1 m diameter) located in the Dynamic Photomechanics Laboratory at the University of Rhode Island, Figure 1. The composite cylinders were secured horizontally in the center of the tank via wire cables attached to pad eyes along the interior of the tank. The tank was flooded with filtered water, leaving a small air space at the top of the vessel. Once filling was complete the vessel was pressurized gradually by the introduction of nitrogen gas into the air pocket at the top. Pressurization continued until the specimen became unstable and collapsed. Recording devices were triggered post collapse providing approximately 1.5 seconds of data prior to the trigger.

Figure 1. Large diameter pressure vessel at the University of Rhode Island Dynamic Photomechanics Laboratory

Recorded data included dynamic pressure at several locations along the length of the composite cylinder as well as full-field 3-dimensional transient displacement of the cylinder surface via high-speed video and Digital Image Correlation (DIC). The dynamic pressures were measured with PCB Piezotronic PCB 138A05 dynamic pressure transducers offset approximately 45 mm from the surface of the cylinder, Figure 2, and were recorded by an Astro-med Dash® 8HF-HS data recorder.

Figure 2. Pressure locations along the axis of the tested specimens

In order to employ the DIC technique the test samples were painted with a high contrast speckle pattern. During the experiments two high-speed digital cameras (Photron SA1 , Photron USA, Inc.) were arrayed so as to obtain a stereoscopic view of the specimen through two windows in the pressure vessel. During the implosion event images of the collapsing cylinder were recorded at rates of 20,000 – 40,000 frames/second. High intensity lights were employed to provide adequate lighting for this rate of capture. The stereoscopic images were then analyzed using the VIC3D 2012 image correlation software produced by Correlated Solutions, Inc., Columbia, SC to obtain full-field, real-time displacements and velocities of the test samples. This technique has been adapted for underwater testing, details of which can be found in [9] and [10].

3. Computational Methods

Computational models of the experiments were built using the Dynamic System Mechanics Advanced Simulation (DYSMAS) fluid-structure-interaction (FSI) finite element (FE) code developed by Naval Surface Warfare Center (NSWC), Division Indian Head. The DYSMAS code consists of a Lagrangian structural solver, an Eulerian fluid solver and a coupler interface. The structural code is a Lagrangian explicit solver for non-linear dynamic analysis known as DYNA3D. It was developed by Lawrence Livermore National Laboratory and allows interaction with the Eulerian fluid code. The Eulerian solver used in this study is Gemini, a compressible, inviscid computational fluid dynamics code capable of modeling shock wave propagation through a fluid as well as more general fluid flow. All simulations were conducted with DYSMAS version 6.9.02.

3.1. Structural Models

3.1.1. Carbon/Epoxy

The DYNA3D structural model, Figure 3, is a half-symmetry model comprised of the composite cylinder, an aluminum end cap and three rigid indenters, to be discussed further in Section 3.4. A half symmetry model was employed due to the mode three collapse shape observed in experiments. The aluminum endcap is modeled with 3,900 Hughes-Liu shell elements as an isotropic elastic material. The indenters, used to initiate instability in the simulated structure, are modeled as a rigid material. The mid-surface of the composite cylinder is modeled with 8,400 shell elements of the Belytschko-Lin-Tsay formulation with 14 integration points through the thickness (2 per ply). There are 160 elements around the circumference of the cylinder. The composite cylinder and aluminum endcaps are meshed as a continuous body. All nodes along the symmetry plane

(Y-Plane) are constrained to prevent out of plane displacement (Y translations) or rotation (X/Z Rotations).

Figure 3. Illustration of the Carbon/Epoxy structural model including the composite cylinder, aluminum end cap, rigid indenters and symmetry plane

3.1.2. E-Glass/PE

During experiments it was observed that the E-Glass/PE cylinders collapsed in a mode 2 shape, therefore an eighth symmetry model was developed with 170 elements around the half circumference of the cylinder. As with the Carbon/Epoxy cylinder the E-Glass/Polyester cylinder was meshed as a continuous body with the aluminum end caps. Along the longitudinal symmetry plane a rigid wall was modeled to prevent the passage of the structure through that plane during collapse.

3.2.Material Model

The material model employed to simulate the composite cylinder is the DYNA3D Material Type 22, Fiber Composite with Damage model [11]. This is a ply-level material model that allows the user to specify material orientations for each ply through the use of a user defined integration rule. Ply level material properties are specified on the material control card in the input deck. The resulting model is approximately orthotropic elastic until failure. A user defined integration scheme incorporating two integration points through the thickness of each ply was used.

The material model described above relies on the interactive Chang-Chang [12] failure criteria for prediction of material damage in the fiber and matrix directions independently. Once damage is predicted in both directions the affected element is eroded over 100 time steps. The stiffness is degraded linearly until the element is no longer able to sustain any load at which point it is removed from the analysis. It was found that this material model over-predicted matrix failure in both composite materials studied. In order to combat this tendency the matrix strength parameter in the material model was set to be equal to the fiber-direction strength effectively limiting the failure prediction to the fiber mode only. This provided a reasonable prediction of initial failure; however, the failure model and degradation scheme employed in the material model was fundamentally flawed and did not provide an accurate representation of material degradation which will be discussed below.

3.3.Fluid Models

The Gemini fluid model, Figure 4, consists of approximately 20 million fluid cells. The extremes of the fluid grid are modeled with non-reflecting boundary conditions with

the exception of the symmetry plane where a wall condition is imposed. The fluid interior to the composite cylinder and exterior to the cylinder extending from the cylinder surface to the pressure sensor locations was modeled with a refined fluid mesh. The mesh gradually becomes less refined as it moves away from the cylinder for the sake of computational efficiency.

Figure 4. Eighth symmetry view of the Eulerian fluid grid showing regions of refined and coarser mesh densities and the approximate location of the cylinder body

The air interior to the cylinder was modeled using the reversible, adiabatic γ law equation of state. The water surrounding the cylinder was modeled with the Tilotson equation of state [13]. This equation of state was developed to model hypervelocity impacts of metals but has found use in the modeling of both cavitating and compressed fluids [14].

3.4.Collapse Initiation

In the previously described experimental studies, [7] & [8], the hydrostatic pressure in the tank was gradually increased until the composite cylinders collapsed. Past work [15]-[16] has shown that the initiation of instability in an implosion event can be greatly influenced by material or geometric imperfections such as out-of-roundness, material voids, or thickness variations. This can result in experimental implosion pressures at nearly half of values predicted by classical buckling analysis or implicit finite element eigenvalue buckling analysis [15].

The numerical models of the composite cylinders employed in this study can be thought of as ideal in both geometry and material, thickness and out-of-roundness variations were not accounted for. In order to properly correlate the experimental pressure data and FSI predictions it is necessary that the hydrostatic pressure in the modeled fluid be equivalent to that in the pressure vessel at the initiation of collapse. In order to achieve collapse at the appropriate pressure the modeled cylinder is perturbed slightly by the indenters shown in Figure 3 after being loaded with the experimentally observed collapse pressure. This can be considered the inducement of an instability causing geometric defect.

The method employed to complete the FSI simulations was also featured in the work of Turner and Ambrico [1] in their study of aluminum cylinders. Prior to integration with the fluid grid the structural model is loaded in a quasi-static manner until the pressure applied to each element is equivalent to the experimentally observed collapse pressure. Once the proper hydrostatic loading has been achieved the indenters begin to accelerate radially to 60 cm/s. When the cylinder begins to buckle and pull away from the indenters

a file is written to record the nodal displacements, velocities and stress states in the structure. The cylinder, in the nascent unstable state, is then integrated into the fluid grid and the simulation is continued. Initiating instability prior to integration with the fluid grid is done for the sake of computational efficiency.

4. Results and Discussion

4.1. Carbon/Epoxy

The simulation predictions of local dynamic pressures at gauge 3 (see Figure 2) plotted with recorded test data at the same location can be seen in Figure 5. It is important to note that a 20 kHz low-pass butterworth filter has been applied to the simulation output. The unfiltered pressure predictions can be seen in Figure 6. As elements were deleted within the simulation the velocities predicted at the surface of the cylinder became discontinuous and caused “ringing,” high frequency pressure waves to be shed into the fluid grid, Figure 7. Removal of these high-frequency pressure oscillations allows for a degree of comparison to be made with the recorded test data.

Figure 5. Predicted vs recorded dynamic pressures, predicted pressure signal filtered at 20kHz

Figure 6. Predicted vs recorded dynamic pressures, unfiltered

Figure 7. x-y plane view of the pressure contours in the fluid highlighting a high pressure wave

Region 1 in Figure 5 represents the initial underpressure developed in the local fluid as the walls of the cylinder begin to collapse in the experiments. As the inward, radial velocity of the cylinder increases, along with that of the surrounding fluid, the local pressure begins to decrease. At point “A”, ~2.5 msec, it was observed that cracks began to form along the lobes of the buckled test specimen. This also coincides with the initiation of damage in the modeled cylinder, although the character is quite different. In Figure 8 it can be seen that damage initiates in the valleys of the buckled cylinder rather than along the lobes as observed during testing. Post-mortem observations of the test specimens revealed that the damage to the composite cylinders was dominated largely by 3 longitudinal cracks along the lobes of the cylinder. These cracks initiated in the outer layer of the laminate and were oriented along the outer-most fiber direction (15°) through the wall thickness. A more detailed description of the damage can be found in [7].

Figure 8. Progression of the collapse of the structure and resulting fluid pressure contours

Region 2 in Figure 5 corresponds with a period of increased collapse rate and more rapidly decreasing pressure. During this phase cavitation bubbles were observed in the high-speed video captured during testing. As the bubbles collapsed, point “B”, a pressure wave was generated. The simulation provides a reasonable correlation with this region of the pressure curve although cavitation on the surface of the simulated cylinder is difficult to distinguish due to the high frequency oscillations in the fluid. Following this pressure spike the local pressures continue to decrease until there is contact between the cylinder walls. In the experimental test data this correlates to point "C" and point “D”. The simulation does not provide a correlation with these points in the curve. It can be seen from Figure 8 that by 6.8 msec the walls of the simulated cylinder have nearly vanished due to the progression of the damage. As a result there is no indication of wall contact in the simulation. It has been observed that using the current material model (MAT22) in DYNA3D results in a very rapid progression of damage. Once damage has been predicted in an element and it is deleted a stress concentration is produced in the neighboring elements causing those elements to be overwhelmed and removed from the simulation over the subsequent 100 time steps. This results in a very poor correlation with observed damage during testing.

In the experiments, as the walls of the cylinder make contact, the velocities of the cylinder surface and the intruding fluid are forced to zero. The sudden arrest in momentum results in a large amplitude pressure wave radiating into the surrounding fluid, point “F”. Because of the duration of the experiments it is likely that reflections of the underpressure wave from the tank walls affected the peak pressures recorded by the sensors. The test tank geometry allows for a 1.4 msec reflection-free window. It is likely

that the peak pressure observed in a free field environment would be higher. Direct comparisons should not be made between the predicted and observed peak pressure magnitudes. The experimental data shows an extended overpressure region with decreasing magnitude following initial wall contact. This region corresponds to collapse of the structure along the longitudinal axis following initial contact at the center point. In the simulations the structure of the cylinder is largely eroded by the time the peak pressure is observed. Without a structure to impede the intruding fluid a more efficient collapse is predicted leading to the large over pressure region following the initial peak.

The Russell Comprehensive Error measurement, [17], was employed to evaluate the level of correlation between the experimental test data and the simulations. This method quantifies the variation in magnitude and phase of two transient signals and provides three metrics by which to judge correlation; magnitude (RM), phase (RP) and overall/comprehensive (RC). The error measures are expressed as:

$$RM = \frac{1}{\pi} \cos^{-1} \left(\frac{\sum c_i m_i}{\sum c_i^2 \sum m_i^2} \right)$$

$$RP = \text{sign}(m) \log_{10}(1 + |m|)$$

$$m = \frac{\sum c_i^2 - \sum m_i^2}{\sqrt{\sum c_i^2 \sum m_i^2}}$$

$$RC = \sqrt{\frac{\pi}{4} (RM^2 + RP^2)}$$

where c_i and m_i represent the calculated (simulated) and measured responses, respectively.

A comprehensive error, RC, value of ≤ 0.15 indicates an excellent correlation between two transient signals. For $0.15 < RC \leq 0.28$ the correlation is considered acceptable and for $RC > 0.28$ the correlation is considered poor.

For the case of the Carbon/Epoxy cylinders the simulation provides a comprehensive error factor of 0.0964 (excellent) in the region of the pressure curves up to “point E” in Figure 5. Comparison of the phase only out to 8 msec shows an error measure of 0.2092, within the acceptable range. If magnitude were included it would be clear that the error would be unacceptable due to the physically unrealistic erosion of the structure but that cannot be determined quantitatively through direct comparison.

4.2.E-Glass/PE

As with the Carbon/Epoxy simulations all predicted pressure pulses from the study of the E-Glass/Polyester cylinders were subject to a 20kHz low-pass butterworth filter to remove physically unrealistic ringing caused by element deletion during the course of the simulation. Figure 9 plots the simulated pressure in the vicinity of the central gauge against the pressure recorded during test. Figure 10 depicts the progression of the collapse of the simulated cylinder.

Figure 9. Predicted vs recorded dynamic pressures for E-Glass/Polyester cylinder, predicted pressure signal filtered at 20kHz

Figure 10. Progression of structural collapse for E-Glass/Polyester cylinder

At point “A” of Figure 9 damage in the test sample was observed in the form of debonding and inter-fiber and matrix cracking. In the simulation, elements did not begin to fail until point “C,” 1.60msec, nearly a millisecond further into the collapse. In the experiments the initiation of damage coincided with an increase in the rate of collapse and more rapidly decreasing pressure. The absence of this immediate damage in the simulation led to the oscillations in the pressure field visible in Figure 9 as the simulated structure resists collapse. Once damage initiated in the simulated cylinder the collapse proceeded at an increased pace and a distinct underpressure region can be discerned.

At point “B”, approximately 1.25 msec, a distinct dimple can be seen just to the left of the cylinder center, Figure 10. This corresponds to experimental observations of a dimple on either side of the cylinder center which propagated toward the end caps along either side. At point “D” the walls of the test samples were seen to make initial contact. In the simulations the initial wall contact was predicted to occur at approximately 2.4msec, point “E.” This wall contact is followed by a distinct overpressure peak, point “G,” in both the test and simulated pressure data sets. Similar to the results outlined above measured peak pressures are likely reduced due to reflections of the underpressure wave from the tank wall and should not be used as a basis of direct comparison. This is followed by considerable oscillations in the simulated pressures, characterized by the expansion and subsequent collapse of the air, no longer contained within the structure.

The Russell Error method was again employed to judge the fitness of the correlation between the simulated and measured pressure data for the underpressure region, point “F”. The comprehensive error was found to be 0.3062, well outside of the acceptable range. The damage modes observed in the experimental testing were largely in the

matrix, including delamination and matrix cracking and debonding. These were very different from the modes observed in the Carbon/Epoxy implosion tests. As the material damage model was not able to accurately capture these modes and was limited to fiber mode damage the poor correlation is not unexpected.

5. Conclusions

Fluid-Structure-Interaction simulations were carried out to model the implosion process of Carbon/Epoxy and E-glass/Polyester laminated cylinders. Initiation of instability, collapse, and damage progression of the cylinders were simulated as well as dynamic pressure in the surrounding fluid. Simulation results were compared with test data and the following conclusions can be postulated:

- Damage evolution during the implosion event was not well predicted by the simulations. Both simulations, one of a Carbon/Epoxy cylinder and one of an E-Glass/Polyester cylinder, failed to capture the damage evolution of the test samples. The damage model employed by the MAT_COMPOSITE_DAMAGE material model in Dyna3D was not able to accurately represent failure in the studied materials and leads to unrealistic erosion of the composite structure.
- Instability in the predicted velocities of the structural model when elements begin to fail and are deleted result in physically unrealistic high-frequency pressure waves being shed into the fluid grid. Future work should investigate and include alternative element degradation schemes in order to minimize these instabilities.
- The dynamic pressure predictions for the Carbon/Epoxy cylinder provide an excellent correlation per the Russell Error Measure with the available test data, both temporally and in magnitude, for the underpressure region. As time

progresses phase correlation degrades due to the effects of incorrect structural damage dynamics and the resulting release of the entrained air.

- The dynamic pressure predictions for the E-glass/Polyester cylinder provided a poor correlation in the underpressure region. This was largely driven by poor damage prediction and an inability to capture the primary damage mechanisms of the material.

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CHAPTER 3
NEAR FIELD UNDERWATER EXPLOSION RESPONSE OF POLYUREA
COATED COMPOSITE CYLINDERS

By

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Abstract

Experiments were conducted on woven E-glass/epoxy roll wrapped cylinders in three configurations; base composite, and base composite with a thin (100% composite thickness) and thick (200% composite thickness) polyurea coating. Each cylinder configuration was subjected to near-field UNDEX loading in a large diameter test tank at charge standoff distances of 2.54 cm and 5.08 cm. The response of the cylinders on the non-loaded side was evaluated through Digital Image Correlation. Post-mortem damage comparisons were made to evaluate the effects of the applied coatings. Results show that the application of a polyurea coating is effective for significantly reducing damage in the cylinders. Center point displacements of the coated cylinders were reduced over the time period evaluated, however, consideration must be made of the weight penalty associated with adding mass to the structure.

1. Introduction

Composite materials have several characteristics which make them particularly appealing in marine environments such as high strength to weight ratios and superior resistance to corrosion. When structures composed of composite materials are fielded in a marine environment they may be subjected to harsh loading conditions such as UNDEX loading, both near and far field in addition to operational loads. Maximizing the benefit of these materials, particularly for minimum weight, requires a full understanding of the response of these materials to such loadings and the effects of any potential mitigators, such as blast resistant polymeric coatings, in order to avoid overly conservative designs.

Studies on the response of composites subjected to UNDEX have generally focused on far field loading in which the encroaching shock front is nearly planar and

there is no interaction between the UNDEX bubble and the structure. LeBlanc and Shukla [1,2] have studied the response of both flat and curved E-glass/epoxy composite plates to far field loading both experimentally and computationally and were able to accurately predict the response and damage evolution in the composite. In [3] Avachat and Zhou investigated the response of monolithic as well as sandwich structure composite cylinders to underwater impulsive loading imparted via a novel Underwater Shock Loading Simulator. They found that the inclusion of a foam core reduced damage to the cylinder as compared with a monolithic composite wall of similar mass. Further, decreasing foam core density resulted in a decrease in observed damage. Mouritz, et al., [4], conducted a study of the development of damage in a glass reinforced composite subjected to underwater explosive loading at increasing pressures both air backed and water backed. In the case of the water backed laminates no damage or degradation in strength was noted. In the air backed laminates delamination and matrix cracking led to a degradation of the residual strength of the composite.

Near-field loading is generally characterized by a spherical shock front impinging upon the structure as well as interaction of the UNDEX bubble and the target structure. This can lead to highly localized damage and response in the structure rather than the more global character of the far field loading. In LeBlanc, et al., [5], coated and non-coated flat E-glass/epoxy plates were subjected to near field UNDEX loading. Deflections and damage extents were compared across the plate configurations. It was found that the application of a polyurea coating reduced the overall response of the plate and significantly reduced damage to the composite. Brett, et al., [6,7], presented a study of steel cylinders subjected to near field UNDEX. They observed that at standoff

distances less than the UNDEX bubble radius the bubble was attracted to the cylinder and collapsed upon it resulting in a significant structural response.

Recently polyurea has found interest as a potential blast mitigating coating. It is an easy to apply polymer that becomes stiff at high rates of loading and is finding use as a post-design phase enhancement. Several studies have been conducted to determine polyurea's ability to reduce structural response to blast loading as well as reduce damage in materials. LeBlanc, et al., [8,9] studied the response of composite plates coated with polyurea to UNDEX loading. It was determined that both location and thickness of the coating were important considerations in efforts to reduce damage and deflection. When considering a weight penalty there is a coating thickness at which the polyurea becomes more advantageous in mitigating the out of plane response of the structure than simply increasing the base composite thickness. Tekalur, et al., [10] and Gardner, et al., [11] studied monolithic and sandwich composites, respectively, subjected to air blast loading. It was found that polyurea was able to mitigate damage and deflection in the monolithic plates. For the sandwich composites blast resistance was improved by placing the polyurea between the back face sheet and the foam core; performance was degraded when the polyurea was applied between the front face sheet and the foam core.

2. Materials

This investigation tested composite cylinders in a base configuration comprised solely of the composite material as well as the base composite with applied polymeric coatings.

Material details are outlined in the following two sections.

2.1 Composite

The composite cylinders were manufactured by ACP Composites, Inc. of Livermore, CA. The material is a cured, roll-wrapped E-glass/epoxy with a woven 0°/90° structure produced by Axiom Materials, Inc of Santa Ana, CA as AX-3112T. The composite cylinders have a wall thickness of 1.14 mm with 4 plies through the thickness and a laminate schedule of [0/45/45/0]. Resin content is 38% by weight and the areal weight is 0.49 kg/m² per ply. The material properties, as provided by the manufacturer, are listed in Table 1.

Table 1. Composite Material Properties

	Strength (MPa)	Modulus (GPa)	Test Method
Tensile	531	29	ASTM D638
Compressive	510	3.7	ASTM D695
Interlaminar Shear	60		ASTM D1002

2.2 Polyurea

A polyurea coating, Dragonshield-BC, was manufactured and applied via spray-cast by Specialty Products, Inc., of Lakewood, WA. This is a 2-part polymer which may be applied to a variety of surfaces. The coating was applied in two thicknesses, 100% and 200% of the composite thickness, to the outer surface of the cylinders and was cured at 160°F for 48hrs. As in the previous study by the authors [5] this configuration is intended to represent the post-design and manufacture application of the coating as reinforcement rather than an integral design aspect.

The polyurea coating was selected due to its strain rate dependent behavior and use as a blast mitigator in fielded systems [12]. A characterization of the polyurea

material was conducted at strain rates of 0.01s^{-1} to 100s^{-1} for both tensile and compressive loading in a previous study, [8]. Additionally, during the same study, strain rates of 2000s^{-1} in compression were achieved via a Split Hopkinson Pressure Bar (SHPB). It is assumed that the behavior of the polyurea is similar in tension for the equivalent strain rate. Figure 1 illustrates the stress-strain behavior of the Dragonshield-BC polyurea monolithic material over the range of tested strain rates. It is clear from Figure 1 that with increasing strain rate the response of the material becomes stiffer in both tension and compression, exhibiting a distinct plateau in tension.

Figure 1. Dragonshield-BC Polyurea Stress-Strain Behavior, [8]

3. Experimental Set-up

The following sections detail the experimental set-up for this investigation. A full account is given regarding the specimen geometry, test vessel, and data acquisition system and methods.

3.1 Specimen Geometry

The outside diameter of the base composite cylinder is 7.44 cm with a thickness of 1.14 mm. The total length of the cylinder is 40.64 cm with an unsupported length of 38.1 cm. Each end of the cylinder is fitted with an aluminum endcap protruding 12.7 mm into the length of the cylinder which seals against the inner diameter of the cylinder via a rubber o-ring to prevent water infiltration during experiments. The endcaps are held in place and the cylinder further sealed by the application of epoxy to the joints between the endcaps and cylinder. In addition to the base cylinder, cylinders were prepared with either a thick ($2.26 \text{ mm} \pm 0.5 \text{ mm}$) or thin ($1.19 \text{ mm} \pm 0.3 \text{ mm}$) outer coating of polyurea. Figure 2 provides a schematic of the cylinder construction.

Figure 2. Cylinder Construction

The areal weights and wall thicknesses of each cylinder configuration is given in Table 2, below.

Table 2. Cylinder Configuration Wall Thicknesses and Areal Weights

	Thickness (mm)	Areal Weight (kg/m ²)
Composite	1.14	1.96
Thin Coating	2.34	3.15
Thick Coating	3.04	3.90

3.2 Explosive Charge

The explosive used in this study is an RP-503 charge manufactured by Teledyne RSI, Inc. of Tracy, CA. It contains 454mg of RDX and 167mg of PETN. A characterization of the explosive was conducted in equivalent test conditions. The maximum bubble diameter was measured to be 21.7 cm. Figure 3 provides a plot of bubble diameter over time from detonation until the initial collapse of the bubble. Figure 4 provides the pressure profile in the water at three different radial distances from the charge center.

Figure 3. RP-503 Bubble Diameter – Growth and Collapse

Figure 4. RP-503 Characterization Pressure Profile

3.3 Test Tank

All experiments were conducted in the large diameter (2.1 m) water filled spherical test tank located in the University of Rhode Island Dynamic Photomechanics Laboratory (DPML). The test tank is rated to withstand pressures up to 6.89 MPa in addition to up to 4 gm of TNT. An array of windows along the horizontal axis of the test tank allow for full viewing and recording of experiments as well as illumination of the test article. The cylinder is mounted and held in the center of the tank via cables suspended from pad eyes located along the tank walls above and below the specimen. The cables include a ratcheting mechanism for adjusting the position of the specimen within the tank as well as tensioning of the cables to minimize rigid body motion of the test article during transient loading.

3.4 Pressure Sensors

Pressures in the vicinity of the cylinder were recorded using PCB 138A05 tourmaline dynamic pressure sensors produced by PCB Piezotronics, Inc. of Depew, NY. The sensors have a dynamic range of 34.5 MPa and a sensitivity of 1.45×10^4 mV/Pa with a 690 Pa resolution. The rise time of the sensors is $\leq 1.5 \mu\text{s}$. Pressure data was monitored and recorded at a sampling rate of 2 MHz using a Dash 8HF-HS data recorder produced by AstroNova, Inc. of West Warwick, RI.

3.5 High Speed Video and Digital Image Correlation

Three high speed video cameras (FastCam SA1, Photron USA, Inc., San Diego, CA) were used to capture video during experiments. One camera was mounted to align with the longitudinal axis of the cylinder, providing a side view of the UNDEX event. The remaining two cameras were arranged to provide a stereoscopic view of the cylinder on the opposite side of the explosive (front view). The two stereoscopic cameras were synced together to provide uniform timing between them. High intensity lights were used to provide the necessary light for high speed video capture. Frame rates of 36,000 fps or greater were used for both the side view and front view cameras.

Each cylinder was prepared for Digital Image Correlation (DIC) data extraction in order to obtain full-field in- and out- of plane displacements of the cylinders during the test event. A coating of white paint was applied to each cylinder and a random pattern of black speckles was applied using flat black paint. Calibration of the DIC system, which includes the two stereoscopic front view cameras, for use in the large diameter test tank was accomplished by Gupta, et al., in [13]. Post processing of the front view high speed video to obtain full field displacements was accomplished using the VIC-3D software

package produced and maintained by Correlated Solutions, Inc., of Columbia, SC.

Displacements are obtained by comparison of pixel subsets of the random speckles between images as the cylinder deforms and the reference un-deformed state.

4. Experimental Methodology

For each experiment the cylinder under test was fixed within a wire support cage used to secure pressure sensors and the explosive at set distances from the cylinder surface.

Figure 5 illustrates the arrangement of the pressure sensors around the cylinders. Collars were affixed to the cylinder endcaps to which the wire cage and the support cabling were attached. The cylinder was then firmly secured in the center of the tank using the support cables and the alignment with the high speed video cameras was confirmed. Figure 6 provides a schematic of the test set-up.

Figure 5. Pressure Sensor Arrangement (not to scale)

Figure 6. Test Configuration, (a) Tank Schematic, (b) Cylinder in Support Cage

Each cylinder configuration (base composite, thick coating and thin coating) was tested at each of two charge stand-offs, 2.54cm and 5.08cm. A total of 16 experiments were conducted covering the range of test configurations. At least two experiments of each cylinder configuration/charge standoff combination were conducted. Representative cases are selected for comparison in the results and discussion.

The charge distance to the cylinder surface was maintained by fixing the charge within the support cage with monofilament line, see Figure 6 (b). The explosive lead wires were passed from the interior of the test tank to the exterior via a high pressure pass-through in the tank wall. Following placement of the set-up within the tank the hatch was secured and the tank flooded with water, leaving a small air pocket at the very top. All experiments were conducted at ambient pressure.

Once filling was complete the explosive lead wires were connected to a detonation box which supplied the amperage required to ignite the explosive. Once the explosion was observed by test personnel a trigger switch was activated. The trigger signal was sent to the high speed cameras which in return sent a positive TTL signal to

the Dash 8HF-HS data recorder. All recording devices utilized an end trigger configuration.

5. Results and Discussion

5.1 Bubble-Cylinder Interaction and Local Pressures

The near field nature of these experiments resulted in interesting interactions between the UNDEX bubble and the cylinders. All interactions were characterized by a splitting of the UNDEX bubble with one bubble forming in front (non-charge side) of the cylinder and the bulk of the UNDEX bubble remaining behind (charge side) the cylinder. Initially, as the shock from the explosive detonation passes the cylinder small cavitation bubbles form on the surface of the cylinder. This happens at 0.36 ± 0.08 msec for the 5.08 cm charge standoff and at 0.23 ± 0.05 msec for the 2.54 cm standoff. This is the result of the UNDEX shock wave interacting and passing by the cylinder and echoes the observations of Brett and Yiannakopolous [6]. As time progresses, the cavitation bubbles begin to coalesce. Following coalescence the cavitation bubbles collapse in front of the central region of the cylinder after about 1 msec. Figure 7 provides images of key developments observed during the bubble-structure interaction during an experiment conducted at a charge standoff of 2.54 cm on a cylinder with a thick coating applied. Similar features are observed in the experiments with a 5.08 cm charge standoff with difference in timing in accordance with the increased distance between structure and bubble center. No significant differences were noted in bubble interaction between uncoated and coated cylinders.

(a)

(b)

Figure 7. Bubble Growth and Interaction (a) Front View, (b) Side View

At around 5.0 msec for the 2.54 cm standoff a large bubble can be seen to form in the front of the cylinder. For the case of the 5.08 cm standoff the bubble forms around 5.5 msec from detonation. The formation of the front bubble coincides with bubble diameters of 18.76 cm and 19.20 cm for the case of the 2.54 cm and 5.08 cm standoffs respectively. Stack-up of the standoff and cylinder diameter show that the UNDEX bubble radius is approximately 3 cm (2.54 cm standoff) and 1 cm (5.08 cm standoff) shorter than the length of the standoff and cylinder diameter. This does not account for cylinder deflection which cannot be determined due to the bubble obscuring the cylinder in the high speed video. This result suggests bubble migration, whereby the center of the bubble is attracted toward the structure. Analysis of side view images shows a horizontal elongation of the bubble as it interacts with the structure and attachment of the bubble to the surface of the cylinder, Figure 8.

Figure 8. Bubble Attachment to Cylinder, side view

The large bubble which forms on the non-charge side of the cylinder collapses upon itself at approximately 12.7 msec. This provides a secondary loading of the cylinder. A third loading occurs with the collapse of the main UNDEX bubble approximately 4 msec following the collapse of the front bubble.

The pressure recorded on the non-charge side of the cylinder is shown in Figure 9. This pressure profile was recorded during the experiment from which the images presented above were taken. The incident shock pulse (5.60 MPa) is followed by the exponential decay in pressure typical of UNDEX loading. At 0.28 msec a second pressure peak (4.23 MPa) is recorded. This is the reflection of the incident shock from the surface of the cylinder. At 1.4 msec and 2.8 msec small pressure peaks can be seen which are the result of successive reflections of the shock wave from the walls of the test tank. From approximately 6.40 to 10.50 msec the front bubble encapsulates the pressure

sensor. At 10.50 msec the passage of the bubble edge past the sensor results in a small pressure increase. At 12.70 msec the pressure sensor records the peak resulting from the collapse of the front bubble which is quickly followed by the reflection from the surface of the cylinder. The magnitude of this pressure peak is 0.95 MPa, 18% of the initial shock recorded at the same location, and represents a significant secondary loading of the cylinder from the bubble collapse. Following the initial collapse the bubble expands and collapses for a second time at 15.12 msec. At 16.63 msec an additional increase in pressure is observed due to the collapse of the main UNDEX bubble behind the cylinder.

Figure 9. Pressure Profile, Non-Charge Side

Pressure recorded on the back (charge) side of the cylinder, to the right of the explosive in the same experiment is shown in Figure 10. This profile contains similar features to the one recorded on the front (non-charge) side of the cylinder. The large standoff between the sensor and the location of the bubble collapse preclude a meaningful comparison with the magnitude of the front bubble collapse pressure which occurred in close proximity to the sensor in front of the cylinder. While the volume of the main bubble is greater than the front bubble, the collapse occurs further away from the surface of the cylinder. It is possible that the bubble which forms in front of the cylinder poses a greater hazard to the cylinder.

Figure 10. Pressure Profile, Charge Side

5.2 Transient Cylinder Response

The response of the cylinders to the near field UNDEX loading will be described primarily by the radial displacement of the center point on the non-charge side of each cylinder. The radial displacements are determined via image analysis through DIC. Due to the bubble interaction with the cylinder described in the previous section the displacements of the cylinders could not be determined for the entirety of the loading events. Large scale cavitation on the surface of the cylinder and the formation of a bubble between the cylinder and the cameras prevent DIC analysis by obfuscation of the speckle pattern. Comparisons will be limited to the time period for which DIC results are available and may not include the peak displacements experienced by the cylinder during test.

5.2.1 Charge Standoff – 5.08 cm

The radial displacement of the cylinders exposed to an UNDEX at a 5.1 cm charge standoff is characterized by an initial global deformation in the positive radial direction (away from the charge and toward the cameras) followed by an inflection and dimpling in the center of the cylinder away from the camera view and toward the charge location as the cylinder rebounds. Figure 11, below, depicts the radial displacement of

line segments along the cylinder centers for all three cylinder configurations (uncoated, thin coated and thick coated) over time. At 0.5 msec the center point displacement for the coated cylinders is 2.5 mm in the positive direction. At this period in time the uncoated cylinder lags with a center point displacement of 1.9 mm in the positive direction. At 1.0 msec the uncoated cylinder has overtaken both the coated cylinders with a positive central displacement of 4.8 mm. The cylinder with the thin polyurea coating has a central displacement of 4.2 mm and the cylinder with the thick coating a 3.7 mm center point deflection. At 2.5msec all cylinders display a negative center point deflection of approximately 2 mm. Scattered cavitation on the surface of the cylinders then obscures portions of the speckle pattern on each cylinder and precludes a high confidence in directly comparing further displacement values.

Figure 11. Centerline Displacements for 5.08 cm Standoff

Full field displacement contours over the initial 2.75 msec of the experiments can be seen in Figure 12. The full field contours confirm the general shape suggested by the center line displacements presented in Figure 10 above. Comparisons with the uncoated

cylinder are difficult due to obscuration of the speckle pattern in that image set after 1.25 msec.

Figure 12. Full Field Radial Displacement Contours

In [9] LeBlanc, et al., introduced the areal weight ratio (AWR) as a means to account for the weight penalty associated with adding material, such as a coating, to an existing design. The AWR acts as a multiplier to quantify the added mass penalty associated with any additional material in terms of transient deflection. The AWR is given by Equation 1 as:

$$AWR = \frac{W_2}{W_1} \quad (1)$$

W_1 is the areal weight of the base material. In this case it is the areal weight of the composite from which the cylinder is constructed, as given in Table 1. W_2 is the areal weight of the base material plus any added material or coating. The AWR for each cylinder in this study is given in Table 3.

Table 3. Areal Weight Ratio

	AWR
Base Cylinder	1
Thin Coating	1.61
Thick Coating	1.99

Figure 13 (a) provides a comparison of center point deflection of the three cylinder configurations at 1.0 msec as determined through DIC analysis. The selection of center point deflections at 1.0 msec as a basis of comparison is driven by the low confidence in the precision of the data past that point in time. Large areas of cavitation and bubble activity develop following that point in time and significantly impact the quality of the calculated DIC results. Figure 13 (b) illustrates the center point deflections with the AWR penalty applied. When added mass is accounted for the thick coating results in an increase in normalized deflection of 54%, from 4.8 mm to 7.4 mm. The thin coating results in an increase in normalized displacement of 42%, from 4.8 mm to 6.8 mm. This degradation in performance was also observed in previous studies by LeBlanc, et al., [5,8,9] on both flat and curved plates subjected to far field loading as well as near field UNDEX loading of flat composite plates. In [9], LeBlanc, et al., studied an array of polylurea coating thicknesses on the response of E-glass/epoxy cross-ply panels and found that there is a coating thickness which does provide an improvement in transient response characteristics even when weight penalty is considered. A similar result for near field

UNDEX loading of composite cylinders with polyurea coatings cannot be ruled out by the findings of this study.

Figure 13. Center Point Displacements at 1msec – 5.08 cm Charge Standoff, (a) Absolute, (b) Weight Penalty Applied

5.2.2 Charge Standoff - 2.54 cm

With a charge standoff of 2.54 cm the deflection of all three cylinder configurations is characterized by global deformation in the positive radial direction during the time domain for which DIC analysis is possible. It is not clear whether or not the cylinders develop the negative radial dimpling observed for the cylinders tested at 5.1 cm display further out in time. Figure 14, below, provides an illustration of the radial displacement of the center line of each cylinder configuration over time.

Figure 14. Centerline Displacements for 2.54 cm Standoff

Again it can be observed that the uncoated cylinder lags in central displacement initially and then overtakes the coated cylinders over time. At 3.0 msec the center point displacement of the uncoated cylinder is 23.9 mm in the positive radial direction (toward the cameras). The thin coating results in a displacement of 21.9 mm and the thick coating 21.3 mm.

Full field radial displacement contours are shown in Figure 15, below. The bowed shape indicated by the line segment plots in Figure 14 can be discerned in the contour plots.

Figure 15. Radial Displacement Contours for 2.54 cm Charge Standoff

Center point deflection at 3.0 msec is used to compare the performance of the uncoated and coated cylinders in accordance with the method outlined in the previous section. Figure 16 (a) shows the absolute displacement while Figure 16(b) shows the normalized displacement. Again, it can be seen that the application of the polyurea coatings degrades performance of the cylinders when the additional weight is accounted for. The normalized peak displacement is increased from 23.9 mm to 35.3 mm for the case of the cylinder with the thin coating, 48%. For the thickly coated cylinder normalized peak displacement increases 77%, from 23.9 mm to 42.4 mm. This result shows that at the closer charge standoff (2.54 cm) the application of the polyurea coatings has a much more deleterious effect on the response of the cylinder (as adjusted for weight) than at the larger (5.08 cm) standoff, where the change in normalized displacement were 42% and 54% for the thin and thick coating, respectively.

Figure 16. Center Point Displacements at 3 msec – 2.54 cm Charge Standoff, (a) Absolute, (b) Weight Penalty Applied

5.3 Damage

While the application of the polyurea had minimal effect on the absolute response of the cylinders, and a detrimental effect when accounting for weight penalty, there was a significant effect of the polyurea coatings on the damage observed in the cylinders post-test. For both charge standoffs the damage was significantly reduced. For this study damage assessments are limited to post-mortem evaluation. Damage evolution could not be ascertained by inspection of the high speed video as most of the damage occurred on the charge side of the cylinders and was not visible to the cameras.

Figures 17 and 18 provide interior and exterior views, respectively, of the damage in the cylinders tested at 2.54 cm. Damage in the uncoated cylinders was dominated by large cracks and missing sections of material. At the center point of the cylinder, nearest the charge location, sections of delamination can be seen along the edges of the missing portions of the cylinder, Figure 19 (a). The damaged section extends 23 cm along the

40.64 cm length of the cylinder. Additionally, curving cracks, suggestive of an ellipsoid indenting of the cylinder, at approximately $\pm 90^\circ$ from the cylinder centroid can be seen, Figure 19 (b).

Figure 17. Interior View of Cylinder Damage – 2.54 cm Charge Standoff, (a) Uncoated, (b) Thin Coating, (c) Thick Coating

Figure 18. Exterior View of Cylinder Damage – 2.54 cm Charge Standoff, (a) Uncoated, (b) Thin Coating, (c) Thick Coating

(a)

(b)

Figure 19. Delamination Along Crack Edge (a), Curved Crack at $\pm 70^\circ$ (b)

For the thinly coated cylinders tested at 2.54 cm the curving cracks are also observed. They occur at a similar angle although extend only 3.8 cm, Figure 20 (a). The damage to these cylinders is dominated by large circumferential and longitudinal cracks emanating from the point closest to the charge location. At the nexus of the longitudinal and circumferential cracks the damage extends through the thickness of both the composite and coating, Figure 20 (b). The circumferential crack continues to extend through the coating to its termination at $\pm 90^\circ$. The longitudinal crack extends through the coating for only 4.1 cm on either side of the center point and then continues an additional 5.6 cm through the thickness of the base composite only. As with the uncoated cylinder delamination can be observed near the area closest to the charge on the interior and exterior surfaces, Figure 20 (b) and Figure 17 (b).

Figure 20. Damage in Thin Coated Cylinder – 2.54 cm Charge Standoff, (a) Curving Crack, (b) Nexus

In the cylinder with a thick coating of polyurea the damage was similar in character to that observed in the thinly coated cylinder but lesser in extent. Again, longitudinal and circumferential cracks extend from the center point, nearest the charge location. Delaminations can be observed on the interior of the cylinder, Figure 17 (c). The circumferential crack, which ranges $\pm 90^\circ$ from the centroid extends through the thickness of the base composite as well as the coating. Fiber pull-out along the interior edge of the crack can be seen in Figure 17 (c). The curving cracks at the termination of the circumferential cracks in the uncoated and thinly coated cylinders are not present in the thickly coated cylinders. The longitudinal crack, visible in Figure 18 (c), runs 7.6 cm along either side of the center point but extends only through the thickness of the base composite.

As would be expected, the damage to the cylinders tested with a charge standoff of 5.08 cm was less severe for all configurations. Figures 21 and 22 provide interior and exterior views, respectively, of the damage in these cylinders. For the uncoated cylinders

the damage is primarily described by a “punched-in” ellipsoid area circumscribed by a fairly clean crack through the thickness of the composite. An additional crack, running 19.7 cm along the length of the cylinder, is visible below the main ellipsoid crack. It can be seen clearly on the interior of the cylinder, Figure 21 (a). Emanating from this secondary longitudinal crack is a circumferential crack along the interior of the cylinder. This crack does not extend through the thickness of the cylinder.

Figure 21. Interior View of Cylinder Damage – 5.08 cm Charge Standoff (a)
Uncoated, (b) Thin Coating, (c) Thick Coating

Figure 22. Exterior View of Cylinder Damage – 5.08 cm Charge Standoff (a) Uncoated, (b) Thin Coating, (c) Thick Coating

In the thinly coated cylinder exposed to a charge standoff of 5.08 cm longitudinal and circumferential cracks can be seen on the interior of the cylinder, Figure 21 (b). The circumferential cracks extend $\pm 70^\circ$ about the centroid, however, they do not extend through the coating, only the base composite. The longitudinal crack extends 8.9 cm on either side of the point closest to the charge location and penetrates through only the base composite, not the polyurea coating.

The cylinders with the thick polyurea coating (5.08 cm standoff) showed significant reduction in damage even as compared to the thinly coated cylinders. In these cases the damage was confined to two small sections of damage at $\pm 60^\circ$ from the centroid. These damage areas consisted of circumferential cracks of 2.5 cm length and

longitudinal cracks of about 1.3 cm centered against the circumferential cracks. These cracks, which extend only through the base composite, can be seen in Figure 21 (c).

6. Summary and Conclusions

The effects of polyurea coatings on the response and damage of submerged, air-backed, composite cylinders subjected to near field UNDEX loading has been investigated through a series of experiments. A large diameter water filled test tank was used to impart the shock loading and pressures in the surrounding fluid were recorded. High speed video and DIC technique were used to determine the response of the cylinders. Post mortem damage analysis provided a basis of comparison between the coated and uncoated cylinders.

Three cylinder configurations were investigated; base composite (1.14 mm thick), base composite with thin polyurea coating (2.34 mm thick), and base composite with thick polyurea coating (3.04 mm thick). Each configuration was tested at charge standoff distances of 2.54 cm and 5.08 cm. Center point deflection histories show that for the case of the 2.54 cm standoff the peak center point displacement at 3.0 msec is decreased by 8.4% in the thinly coated cylinder as compared with the uncoated cylinder. The thick coated cylinder shows a 10.9% reduction in displacement. When a weight penalty is applied the response of the cylinder is degraded. A similar trend was found for the cylinders tested at a 5.08 cm charge standoff. Application of a thin coating resulted in a 12.5% reduction in center point displacement while a thick coating produced a 22.9% reduction. At the greater charge standoff the polyurea had a more mitigating effect on the center point displacement, however, when accounting for weight penalty the response was again degraded. Damage to the coated composites was dramatically reduced as

compared with the baseline cylinders, with thicker coating application providing increased protection.

For both of the charge standoffs investigated the UNDEX bubble split upon interaction with the cylinders into two bubbles. The bubble on the non-charge side of the cylinder collapsed in close proximity to the surface of the cylinder and produced local pressures 18% of the initial shock loading and caused a visible reaction in the cylinder. The main bubble on the charge side of the cylinder also collapsed, although at a greater standoff. The magnitude of that collapse pressure was not able to be ascertained.

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CHAPTER 4

CONCLUSIONS AND FUTURE WORK

1. CONCLUSIONS

This work has used computational tools and experimental methods to investigate the response and damage evolution in composite materials subjected to extreme underwater loading events. The goal of this study has been to address gaps in knowledge of the response of composites to extreme loading as well as computational methods required to develop predictive capabilities to inform future composite structural design for extreme environments. Highlights of the contributions made towards these goals are outlined below.

1. While there are numerous studies on the effects of preload on composite panels subjected to ballistic and low-velocity impact in air, there is a gap in the literature on the effects of preload on the UNDEX loading of composites. Many marine structures composed of composites are subject to operational loads which would be superimposed upon any shock that it may endure. This study addressed the gap in understanding of the response of preloaded composites to UNDEX through computational simulations performed with a validated finite element model. Overall, it has been shown that for thin composite plates subjected to underwater shock loading conditions there is a minimal effect of preload on the response of the plates. The primary effect is seen in the rate of recovery of the displaced shape. There is minimal effect on the amount of damage and delamination sustained; however there does appear to be a slight protective effect from moderate tensile preloading.

2. Prior simulation and modeling efforts detailed in the literature have focused on the effect of composite composition, imperfections and layup on the buckling mode and collapse pressure of composite cylinders subjected to hydrostatic pressures. All of these efforts have been focused on the structure itself and do not address the effects on the surrounding environment of a composite implosion. This study has focused on developing computational methods to simulate the collapse of the structure itself as well as the pressures resulting as a first step toward developing predictive tools and defining the mechanisms that must be captured in order to do so. This study demonstrates the feasibility of simulating composite implosion and highlights mechanisms that must be captured for reliable results as well as a path forward toward a predictive capability.
3. Very little work exists on the near-field UNDEX loading of composites, particularly the loading of composite cylinders coated with a strain-rate dependent material such as polyurea. This study has shown that the coating has a significant protective effect when applied to the exterior of the cylinder, with increasing coating thickness providing increasing protection from damage. Further, at the standoff distances investigated in this study it was observed that the UNDEX bubble split around the structure and caused successive loadings of the cylinder following collapse of each bubble.

2. FUTURE WORK

Much work remains in the quest to understand the responses of composite materials to extreme environments. In order to make the most use of their high strength-to-weight ratio in structural designs we need to move forward with experimental work to

uncover damage and response phenomena as well as work to develop computational tools that will lead to a high confidence predictive capability.

1. A worthwhile extension of the investigation of preload on the shock response of composites could include the effects of laminate variation. The use of an unbalanced and/or asymmetric layup can lead to bending/stretching coupling whereby in-plane loading results in out of plane displacement. The resulting displacement would change the presentation of the plate with respect to the impinging load as well as induce bending moments in the plate which may affect the plate response. Such effects could prove important in design consideration.

2. Alternative modeling and analysis strategies should be considered in the case of the implosion of composite cylinders. More mature material models and analysis codes, such as LS-DYNA, have a greater potential for capturing the structural damage evolution during the collapse. High fidelity pressure predictions in the surrounding area clearly depend upon being able to capture the dominant failure mechanisms of the structure. This work has shown that finite element analysis of the implosion of composite cylinders is feasible and has provided a view of potential pitfalls and paths forward. Further effort is warranted to explore alternative strategies with an end goal toward a truly predictive capability in the future.

3. Many studies in the literature have indicated that the location (loaded or non-loaded side) of reinforcing polyurea layers on a structure can have a significant impact on the response of, and damage sustained by, a structure. In some cases the placement can lead to degradation in performance as compared with a non-coated structure. Future work

should address the placement of the reinforcing polyurea layer on the interior of composite cylinders subjected to near field underwater explosive loading.

4. In addition to extreme loading, extreme environments (and the interplay of the two) should be investigated. One such environment gaining interest is the extreme cold of the Arctic. The effects of extremely cold temperatures on the material properties of composites as well as their response to dynamic events such as air blast loading at low temperature should be explored. Furthermore, the effects of extreme cold on polyurea coatings and their ability to mitigate blast should be considered.

5. Another area for future exploration is the bubble dynamics observed during the investigation of near-field underwater explosive loading of composite cylinders. No analog was found in the literature to describe the splitting of an UNDEX bubble around a structure. Necessary conditions for this phenomenon, such as standoff distance and structure geometry, should be investigated as well as a more close accounting of the resulting loads imparted to the structure.