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A Non-Explosive Methodology for Generating Wide Area Close-in Dynamic Blast Pressure Loads on Flexible Armor Panels

A dissertation submitted in partial satisfaction of the requirements for the degree Doctor of Philosophy in Structural Engineering

by

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2014
The Dissertation of Daniel A. Whisler is approved and it is acceptable in quality and form for publication on microfilm and electronically:

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Chair

University of California, San Diego

2014
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<tr>
<td>AD</td>
<td>Areal density, in units of kg/m²</td>
</tr>
<tr>
<td>ANFO</td>
<td>Ammonium nitrate and fuel oil (high explosive)</td>
</tr>
<tr>
<td>AS</td>
<td>Solid aluminum layer</td>
</tr>
<tr>
<td>BG</td>
<td>Blast generator</td>
</tr>
<tr>
<td>C4</td>
<td>Composition C-4 (high explosive)</td>
</tr>
<tr>
<td>CG</td>
<td>Center of gravity</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data acquisition system</td>
</tr>
<tr>
<td>FE</td>
<td>Finite element</td>
</tr>
<tr>
<td>Fm</td>
<td>Aluminum foam layer</td>
</tr>
<tr>
<td>HE</td>
<td>High explosive</td>
</tr>
<tr>
<td>HMMWV</td>
<td>High mobility multipurpose wheeled vehicle</td>
</tr>
<tr>
<td>HMX</td>
<td>High molecular weight explosive, also known as octogen, chemical name Cyclotetramethylenetetranitramine</td>
</tr>
<tr>
<td>Hn</td>
<td>Stainless steel honeycomb layer</td>
</tr>
<tr>
<td>HOPK</td>
<td>Hopkinson bar tests on coupon specimens, June – August 2013</td>
</tr>
<tr>
<td>IED</td>
<td>Improvised explosive device</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Description</td>
</tr>
<tr>
<td>--------------</td>
<td>-------------</td>
</tr>
<tr>
<td>LBST</td>
<td>Large Blast and Thermal Simulator</td>
</tr>
<tr>
<td>MRAP</td>
<td>Mine resistant ambush protected vehicle</td>
</tr>
<tr>
<td>MTSF</td>
<td>Panel specimen MTS uniaxial compression test, fast-rate 0.250 m/s</td>
</tr>
<tr>
<td>MTSQ</td>
<td>Panel specimen MTS uniaxial compression test, quasi-static 10 mm/min</td>
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<td>NEM</td>
<td>Non-explosive methodology</td>
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<tr>
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<tr>
<td>OBLT</td>
<td>Oregon Ballistics Laboratory blast tests, December 2012</td>
</tr>
<tr>
<td>PE4</td>
<td>Plastic explosive 4 (high explosive)</td>
</tr>
<tr>
<td>PI</td>
<td>Performance index</td>
</tr>
<tr>
<td>RDX</td>
<td>Research department explosive (high explosive), also known as cyclonite, hexogen, and T4, chemical name Cyclotrimethylenetrinitramine</td>
</tr>
<tr>
<td>RHA, RH</td>
<td>Rolled homogeneous armor (steel)</td>
</tr>
<tr>
<td>SG</td>
<td>Strain gauge</td>
</tr>
<tr>
<td>SHPB</td>
<td>Split Hopkinson pressure bar</td>
</tr>
<tr>
<td>SP</td>
<td>Polymer layer</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Description</td>
</tr>
<tr>
<td>--------------</td>
<td>-------------</td>
</tr>
<tr>
<td>TF</td>
<td>T-flex dry aramid layer</td>
</tr>
<tr>
<td>TNT</td>
<td>Trinitrotoluene (high explosive)</td>
</tr>
<tr>
<td>UCSD</td>
<td>University of California, San Diego</td>
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</tbody>
</table>
LIST OF SYMBOLS

\( \varepsilon_L \) Locking strain for foam, signals onset of densification

\( \varepsilon_{no} \) Nominal (engineering) strain

\( \varepsilon_{pl} \) Plastic strain

\( \varepsilon_{tr} \) True strain

\( J_{SP} \) Specific impulse, in units of Pa-s or N-s/m²

\( k \) Abaqus parameter compression yield stress ratio

\( k_t \) Abaqus parameter hydrostatic yield stress ratio

\( v_{pl} \) Abaqus parameter plastic Poisson’s ratio

\( P \) Pressure

\( Q_E \) Mass specific energy

\( \sigma_{no} \) Nominal (engineering) stress

\( \sigma_{tr} \) True stress

\( \sigma_C \) Collapsing stress for foam

\( T \) Time

\( W_E \) Equivalent weight
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ABSTRACT OF THE DISSERTATION

A Non-Explosive Methodology for Generating Wide Area Close-in Dynamic Blast Pressure Loads on Flexible Armor Panels

by

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Dynamic blast testing of armor components often requires explosives to generate the high pressure, wide area impulses. While explosives provide the most realistic loading conditions, they are difficult to replicate consistently and necessitate remote test facilities for safety. Non-explosive methodologies, such as gas guns and shock tubes, can produce high impulse dynamic events with higher repeatability and increased safety, but are often limited to smaller-sized targets. To impact wide area flexible armor panels with
the blast characteristics of a close-in detonation, a non-explosive methodology was investigated using the U.C. San Diego Blast Simulator. The objective was to create a consistent, economical, and scalable methodology for comparing conventional steel and prototype sandwich panel performance with validation of damage modes and extent of damage using actual blast tests and finite element modeling.

A tiled projectile array having spatially and temporally varying pressure pulses was developed to replicate the spherical loading profile of a close-in detonation. Using a high speed servo-hydraulic actuator, the projectile was launched at 23.0 – 24.6 m/s, equivalent to 7,520 – 8,460 Pa-s with less than 1.9% standard deviation over 43 tests. This was comparable to 1.37 kg of C4 at 305 mm standoff, which was also used to test five armor panels. Sandwich panel transmitted pressures, measured indirectly via transmission plate acceleration, showed up to 75% reduction in maximum values compared to the steel armor panels, with up to 49% weight savings. Deformation profiles of the non-explosive tested panels were similar in both shape and magnitude compared to the blast tested panels, but with more consistency and symmetry. Blast tested panels showed more extensive core crushing for the sandwich panels but no difference for steel.

Finite element modeling predicted similar deformation profiles and transmission plate velocities and accelerations. The models showed higher core crush for the blast tested panels and stress concentrations that matched both sets of test results. The applied impulses for the non-explosive tests were also predicted to be higher than the blast tests.

The UCSD Blast Simulator was able to achieve similar levels of damage compared to an actual blast test, with greater repeatability between tests.
I INTRODUCTION

Buildings, bridges, dams, and other civil structures have long offered the means of mitigating the effects of nature, such buildings providing shelter from sun and rain. With the advent of black powder as early as the ninth century China and modern war vehicles in the twentieth century, structures in hostile environments faced a new threat (explosives) requiring a different design element: armor. Just as importantly, the vehicles themselves, used both offensively to cause destruction and defensively to protect and transport, required lightweight armor systems capable of withstanding the latest threats. The battlefield of the twenty first century saw the rise of a new “weapon of choice for global insurgents and terrorists”—the improvised explosive device (IED) [1].

Since 2001, IEDs have been attributed to over 50% of casualties to coalition forces in Iraq and Afghanistan; killing or wounding 64,000 civilians and military service members in 2008-2010 [1]. In 2011 alone, there were over 6,800 IED events worldwide (not including Iraq and Afghanistan) resulting in 12,286 casualties in 111 countries [2]. In response to this threat, the United States of America began a four year, $35 billion dollar project to develop a series of Mine Resistant Ambush Protected (MRAP) vehicles for use in hostile environments. The vehicles proved successful in Afghanistan, having reduced injuries and deaths from IEDs by 30% and lowering the fatality rate by 65% compared to troop transportation using the High Mobility Multipurpose Wheeled Vehicle (HMMWV) [3].
However, the nature of an *improvised* explosive device is that the threat level is constantly evolving, potentially becoming more dangerous with every iteration. Adding more armor protection is a necessity in order to protect against the latest threats, but for conventional steel armor, increasing blast protection often equates to increasing weight. On vehicles that are already in excess of 12,700 kg (28,000 lbs), the added weight reduces efficiency, capacity, maneuverability, and speed—all of which are critical to safe transport in a hostile environment. Lighter armor that meets or exceeds the blast resistance performance of conventional steel armor is desired. A test methodology that facilities both rapid development of the armor for the end-user while providing high quality data for researchers and engineers is therefore necessary.

The current practice for testing armor panels involves both explosive and non-explosive techniques. Each method has their own set of benefits and limitations, and while explosive detonations will continue to be used to develop armor components, the non-explosive methodologies offer many benefits that the explosive techniques simply cannot provide. Thus, the explicit objectives of this research project are to develop a non-explosive methodology for rapid, repeatable, economical, and scalable wide area pressure pulse loading on rolled homogenous armor (RHA) steel and flexible sandwich armor panels that replicates the pressure pulse characteristics and damage produced by the detonation of a close-in high explosive. Specifically, this requires:

i. Establishing non-explosive methods capable of replicating blast events to create the close-in wide area pressure pulses. This includes developing the tools and components to achieve this simulated blast load.
ii. Examining the performance of the sandwich armor panels in terms of pressure pulse attenuation and impulse absorption compared to conventional RHA steel.

iii. Comparing the damage modes and extent of damage caused by the non-explosive methodology with that generated by actual explosive blast loads.

iv. Validating finite element (FE) models simulating the response of panels subject to both theoretical blast pressure pulse and non-explosive loading.

To meet these challenges, the University of California, San Diego (UCSD) Structural Engineering Department and Armorworks Enterprises LLC of Chandler, AZ, USA began a four year collaboration under an Office of Naval Research STTR Phase II program (contract no. N00014-11-C-0288) to pursue a non-explosive methodology (NEM) for testing prototype armor panels. Armorworks designed and fabricated the armor panels (some designs were supplied by other manufacturers) while UCSD developed the test methodology and performed the experiments. While the non-explosive methodology refers to the actual large panel tests incorporating a novel projectile engineered to permit consistent wide area dynamic loading, the actual investigation included two other crucial components as well: dynamic material testing and finite element analysis. Details of the methodology, testing, and finite element modeling are presented in full within this body of work.
2 BACKGROUND

This chapter introduces explosive blast loading and typical characteristics of the pressure pulse time history. It discusses methods of replicating all or some of the blast profile characteristics in a controlled environment via explosives, shock tubes, gas guns, and the Blast Simulator.

2.1 A PRIMER ON EXPLOSIVES

Explosives, as defined within this manuscript, refer specifically to a class of chemical substances known as high explosives, e.g., trinitrotoluene (TNT) or Composition C-4 (C4). These substances are characterized by their ability to detonate, a process whereby rapid change in potential energy results in the sudden release of gases and heat at supersonic speeds [4 – 6]. The terms detonation, explosion, and blast are used interchangeably within this work to indicate the sudden and violent process that is a defining characteristic of chemical high explosives (HEs). Low explosives, such as black powder, are also included in the category of chemical explosives, but since they do not derive their power through detonation (relying instead on a relatively slow burn rate or deflagration) they are not considered when referring to explosives within this dissertation document. Natural phenomena such as lightning spark discharge or man-made physical event like a bursting pressure vessel, while technically defined as an explosion in the
general sense, are also not considered explosive events within the context of this document [7].

High explosives are often categorized by their sensitivity to detonation, with the easiest to ignite compounds such as mercury fulminate and lead azide given the name primary explosives [6]. These chemicals are typically used in small quantities as blasting caps to ignite a larger quantity of secondary explosives that have a higher resistance to impact, flames, or sparks, such as TNT and C4. A block of C4, for example, will not detonate if shot with a bullet or lit on fire. Tertiary explosives, such as ammonium nitrate and fuel oil (ANFO), are even less reactive to external stimuli and in many cases cannot even be detonated with only primary explosives (they require detonation by a secondary explosive). It should be emphasized that the term secondary or tertiary explosive is a classification based on resistance to detonation and does not necessarily imply a less powerful explosive medium. Also, for the purpose of this investigation, the composition of the secondary or tertiary explosive (which usually constitutes a much larger percentage of the charge than the primary explosive) will refer to both the explosive and its detonating mechanism; e.g., 1.37 kg C4 implies 1.37 kg of C4 and the necessary blasting cap.

For high explosives, the detonation event and more specifically, the shockwave produced by the detonation is the primary means of doing work on the surroundings [5, 6]. Upon detonation, gases are released at high temperatures, around 7,000 °C, most of which propagate out into the surrounding atmosphere, although some enter the remaining explosive charge [5]. The high temperature gases increase the rate of the chemical
reactions within the explosives, which then release gases at a faster rate, and the exothermic process continues until the entire explosive is consumed [5]. As the process is near instantaneous (within the time scale of interest), the gases are assumed to release into the atmosphere at the same instant which compresses the surrounding air into a thin, extremely high pressure layer around 20 – 30 GPa [8] which forms the shockwave front. If the initial charge source is idealized as a point source in air (away from any structures or the ground, see Figure 2.1), then the shock wave is a spherically expanding front, which has a radius $R$ that is proportional to time $t$ as described by Equation 2.1 [8].

$$R(t) = S(\gamma)t^{2/5}E^{1/5}\rho_0^{-1/5}$$

(2.1)

The term $\rho_0$ is the density of the atmosphere, $E$ is the energy released and $S(\gamma)$ is a function of the ratio of the specific heats of air.

![Figure 2.1. Ideal point source blast detonation in air](image)
The radius grows rapidly from its source since the shockwave travels at supersonic speeds, 7,000 m/s (TNT) to 8,000 m/s (C4) in standard atmospheric conditions [5]. Should an object exist within the blast area or blast exclusion zone, the shockwave interacting with the object’s surface induces a pressure pulse load. This pressure pulse, sometimes termed the overpressure, can be represented as a function of time $t$ by the modified Friedlander equation presented in Equation 2.2. The modified Friedlander equation is often used as the original does not include atmospheric pressure [9].

$$P(t) = P_0 + P^{max}(1 - t/T)e^{-\alpha t/T}$$  \hspace{1cm} (2.2)

Here $P_0$ is the ambient pressure, $P^{max}$ is the maximum overpressure, $T$ is the positive overpressure duration (time between $T_0$ and $T_A$), and $\alpha$ is the waveform parameter [10] with values ranging between 0 – 10 [11]. Equation 2.2 is plotted with an intentional time offset ($T_0-T_D$) in the x-axis in Figure 2.2. This is to show that detonation is occurring at $T_D$ at a short distance away from the target. A pressure device set at the target measures the arrival of the shockwave at time $T_0$. Almost instantaneously, the blast wave increases the pressure above ambient to the peak overpressure value $P^{max}$ at time just after $T_0$. As the shockwave continues to travel radially outward, the pressure pulse no longer applies to the target, and the measured pressure drops. The pressure reaches ambient at time $T_A$, and as the high temperature gases behind the shockwave cools, the pressure falls below ambient. Finally, at $T_B$, the pressure at the measured point returns to ambient conditions.
Figure 2.2. Pressure time history for theoretical blast

The integration of the pressure time history curve between $T_0$ and $T_A$ yields the positive specific impulse, more commonly referred to as simply impulse within the blast community. It is expressed in Equation 2.3 and herein given the symbolic term $J_{sp}$.

$$J_{sp} = \int_{T_0}^{T_A} P(t) dt$$ (2.3)

By this definition, impulse has units of Pa-s or N-s/m² (and not N-s) and may also be derived as the total (force) impulse divided by the contact area of the target. In this manner, impulse can be used to conveniently express magnitude of the blast event.
irrespective of the target dimensions. To simplify matters further, the positive pressure pulse time history curve is often represented as a right triangle of height $P_{\text{max}}$ and base of $T_A$, in which case the impulse can be represented by Equation 2.4.

$$J_{SP} = \frac{1}{2} P_{\text{max}} (T_A - T_0) \quad (2.4)$$

The same integration could be applied to the negative pressure regime, but in most instances, $P_{\text{min}}$ is much smaller than $P_{\text{max}}$, so this portion of the pressure time history is much less important than the initial, positive pressure loading [12, 13]. For this investigation, emphasis is placed on creating the positive impulse profile only.

As different high explosives produce different pressure pulse characteristics, it is common to state a TNT equivalent when expressing explosive charge parameters. Typically, the composition, weight, and standoff distance of the charge are the minimum descriptors for indicating blast potential. These values may be expressed in terms of the original explosive (if TNT is not used) or more commonly, through the TNT-equivalent (TNT$_E$) value. For obtaining the TNT$_E$ weight of an explosive $W_E$, the mass specific energy of the substance $Q_E$ is scaled by the mass specific energy $Q_{\text{TNT}}$ and weight $W_{\text{TNT}}$ of TNT by Equation 2.5.

$$W_E = \frac{Q_{\text{TNT}}}{Q_E} W_{\text{TNT}} \quad (2.5)$$
Values of $Q_E$ for common explosives are provided in Table 2.1 which is reproduced from Mays [14]. C4, which is approximately 91% by weight RDX [15], has an approximate scaling factor of 1.34 [16], although a 1.27 scaling factor was used for this investigation based on input from Oregon Ballistics Laboratory. Using Equation 2.5, a 1.37 kg charge of C4 is equivalent in destructive power to 1.74 kg TNT.

Table 2.1. Mass specific energies for select high explosives, reproduced from Mays [14]

<table>
<thead>
<tr>
<th>Explosive</th>
<th>Mass specific energy $Q_x$ (kJ/kg)</th>
<th>TNT Equivalent $(Q_x/Q_{TNT})$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitroglycerin</td>
<td>6,700</td>
<td>1.481</td>
</tr>
<tr>
<td>C4 (91% RDX)</td>
<td>--</td>
<td>1.19-1.37</td>
</tr>
<tr>
<td>HMX</td>
<td>5,680</td>
<td>1.257</td>
</tr>
<tr>
<td>Semtex</td>
<td>5,660</td>
<td>1.250</td>
</tr>
<tr>
<td>RDX (cyclonite)</td>
<td>5,360</td>
<td>1.185</td>
</tr>
<tr>
<td>Compound B</td>
<td>5,190</td>
<td>1.148</td>
</tr>
<tr>
<td>TNT</td>
<td>4,520</td>
<td>1.000</td>
</tr>
<tr>
<td>Blasting gelatin</td>
<td>4,520</td>
<td>1.000</td>
</tr>
<tr>
<td>ANFO (94% ammonium nitrate, 6% fuel oil)</td>
<td>3,932</td>
<td>0.870</td>
</tr>
<tr>
<td>Dynamite (60% nitroglycerin)</td>
<td>2,710</td>
<td>0.600</td>
</tr>
</tbody>
</table>

Simple scaling laws are also used to relate peak overpressures when changing the standoff distance for a constant explosive charge weight (or changing the charge weight for a constant standoff distance) as well as more complex interactions with buried explosives or near-surface detonations. Buried charges are affected by factors such as the soil type and compaction level, which determine the scaling rule. For near surface charges, the location of structures, ground, and even the ambient conditions may affect the pressure time history curve. A discussion of these interactions is outside the scope of
this project. It is sufficient to state that for certain conditions such as an explosives
buried in soil, amplification factors are used to adjust the peak overpressure and blast
duration to more accurately represent the values observed through experimentation [17].

In summary, chemical high explosives are capable of generating high pressure
and short time duration loading profiles. Specific features of the pressure curve including
the peak overpressure and positive pressure pulse duration are dependent on the charge
composition, weight, and positioning relative to the target. Ambient conditions and the
presence of structures also affect the pressure pulse. To accommodate various test
parameters, scaling factors are used to equate the blast characteristics and pressure curve
from one system or test scenario to another.

2.2 METHODS FOR CONTROLLED BLAST TESTING

Creating the high pressure, short time duration blast load in a controlled test
environment is possible using actual explosives, shock tubes, gas guns, and the Blast
Simulator. A review of these techniques with emphasis on panel-type specimens is
provided.

Using explosives to replicate the pressure pulse of an explosive is one of the most
common methods for creating blast loads in a controlled environment. Combined with
scaling rules for weight, standoff, and ground/structure interactions, a small quantity of
explosives can be used to reproduce a wide range of blast conditions. It should be
emphasized that the size of the explosive package is very small, on the order of kilograms
(or less) in many investigations. This enables an explosive charge to be used in many different test setups and configuration on or within a structure which is a feature that other methods cannot achieve. Thus, researchers often use HEs when creating blast events, especially for wide area pressure loads on panel-type specimens. Dharmasena et al. [18] tested 610 x 610 mm stainless steel honeycomb core sandwich panels in a 410 x 410 mm window fixture subject to 1.0, 2.0, and 3.0 kg TNT charges at 100 mm offset (366, 458, 506 MPa and 21,500, 28,400, 33,700 Pa-s). They also investigated pyramidal lattice core sandwich panels of the same size against 150 g C4 plastic explosives at 75, 150, and 200 mm offsets (producing impulse on the order of 1,500 Pa-s) [15]. They found that metallic core sandwich panels were able to reduce deflection over monolithic materials but for the weaker core pyramidal lattice panel, the reductions were much less pronounced [15]. The sandwich panels were also observed to transmit less force to the supports compared to a monolithic material [15]. Rimoli et al. [19] used the same test configuration from [15] to examine a “wet-sand” blast where 375 g of C4 inside an 80 mm sphere which in turn, was placed at the center of a 152 mm sphere. The outer sphere was filled with water and 2.47 kg glass microspheres to replicate the wet-sand. This was tested against 610 x 610 mm corrugated core aluminum panels at 150 to 300 mm standoff and was found to reduce maximum permanent deflections by up to 20% compared to monolithic panels of similar areal density [19].

Measuring the applied impulse for explosive testing was accomplished by Zhu et al. [20] where a four wire ballistic pendulum was used to measure the impulse from a 20 g TNT charge at 200 mm offset. The charge was used to test 5052 aluminum honeycomb
core and 2024 aluminum facesheet sandwich panels with dimensions 310 x 310 x 12.5 mm. They observed that localized deformation on the front face occurred in the presence of thicker facesheets, higher core density, and larger charge sizes whereas global deformation occurred for thinner facesheets, lower density cores, and smaller charge sizes. They also observed a linear relationship between backside deformation and impulse level. Larger charge sizes were investigated by Hanssen et al. [21] using a 1,400 - 2,700 kg pendulum suspended on a 2.0 m arm. The pendulum supported 684 x 700 mm aluminum foam core panels which were subject to 1.0 and 2.5 kg plastic explosive 4 (PE4, similar in blast effects as C4) at 500 mm standoff. They found that an aluminum foam sacrificial layer could control the contact pressure levels but could not reduce the global impulse transferred.

Although these investigations demonstrated how explosives could provide the necessary loading conditions for testing wide area sandwich panels, there are some very significant limitations. Explosives are inherently dangerous and subsequently require remote facilities for safe operation and extreme caution in handling. The resulting intense shock wave damages equipment and sensors, and the creation of a flash, dust, and debris cloud hinders direct observation via high speed video cameras (see Figure 2.3). Also, there is difficulty in measuring the exact pressure pulse applied to the system since sensors are typically not installed on the loading surface. In order to overcome these limitations, especially in a research setting, non-explosive test methods such as shock tubes and gas guns are often used instead.
A shock tube is a cylindrical tube divided into a driver section and a driven section. The driver section is where gases are released either by a compressed gas source
bursting a membrane or an explosive detonation. The driven section is where the rapidly expanding gases (from the driver section) create a high pressure wave front that travels the length of the chamber to impulsively load the target located at the exit [7, 12]. Naturally, the loading region is limited to the diameter of the driven section. Testing by LeBlanc et al. [7] and Tekalur et al. [12] for example, showed composite panels up to 305 mm in span loaded by a 75 mm diameter shock tube generating up to 8.15 MPa peak overpressure. LeBlanc et al. [7] observed surface fiber damage occurring first at the clamped boundary conditions (away from the pressure-loaded zone) and then at the center with increased damage levels. Tekalur et al. [12] found stitched foam core sandwich panels had higher load transfer to the core and unstitched sections showed the front facesheet deforming independent of the back facesheet, thereby resulting in early specimen failure. In both instances, the size of impact zone was significantly smaller than what would be needed for panels with > 300 mm spans [12]. Large diameter shock tubes do exist, such as the 12 m wide semi-circular unit in Gramat, France and the 20 m wide, 11 m tall semi-circular Large Blast and Thermal Simulator (LBST) unit in White Sands, New Mexico. These units use multiple compressed gas driver tubes that may not generate the same pressure wave characteristics compared to a real blast event and often produce reflected waves which create secondary shocks within the decaying portion of the blast wave [7, 22]. Large diameter shock tubes that utilizing explosive driver sections exhibit some of the same issues associated with actual blast detonations.

In a manner of operation similar to the shock tube, gas guns have been used to dynamically load test specimens. The gas guns have a “driver” section whereby a
compressed gas source releases the gas into a cylinder. In the “driven” section, a projectile (not air) is accelerated to a high velocity and physically impacts the target located at the exit of the tube [23]. Through careful projectile tuning, various pressure loading profiles are possible [24, 25]. To generate a large magnitude overpressure with short time duration, Radford et al. [26] used aluminum foam projectiles of length 50 and 100 mm. Their 28.5 mm diameter projectiles achieved over 100 MPa peak pressures with a short, 250 µs pulse duration. Raj et al. [27] used a 19.5 mm diameter split Hopkinson bar setup at 1,000 1/s and up to 50 MPa to show TiH₂ decomposition aluminum closed cell foam with relative densities ranging from 0.06 – 0.4 exhibited increased plateau stresses with increased density and strain rates. Relative density is a measure of the foam material density divided by the solid material density. The energy absorption capacity was also found to increase with strain rate, making this material well suited in lightweight blast mitigation systems. An investigation by D’Mello et al. [28] using a smaller, 12.7 mm split Hopkinson bar and gas gun setup demonstrated strain rate sensitivity in the crush load for three and seven cell circular honeycomb materials. The crush initiation took place at the end(s) of the specimen and propagated throughout the specimen until full crushing was achieved. For all gas gun experiments, sample size is limited by the barrel diameter which is typically on the order of tens of millimeters. Creating a mechanical impact without the use of explosives on targets up to several meters in size requires significantly larger systems, such as the University of California, San Diego (UCSD) Blast Simulator.
The UCSD Blast Simulator, shown in Figure 2.4, is a servo-hydraulic system capable of applying an impulse up to 84,000 Pa-s on specimens up to several meters in size without the use of explosives. This is achieved through the use of high speed, computer-controlled actuators, termed “blast generators” (BG) which are able to impact the specimen at 25 m/s (BG25) or 50 m/s (BG50). These are nominal velocities and the actuators are reaching up to 34 m/s and 66 m/s, respectively, when accelerating smaller masses [29, 30].
The Blast Simulator can be run with a single actuator or multiple actuators. If multiple actuators are used, they can either be synchronized to within 1 ms to impact the specimen at the same time or staged to impact different positions on the specimen at different time offsets. Regardless of the number of actuators, the system can incorporate one of three impact methods:

i. **Push**- actuator impacts the target and keeps driving in until it comes to rest (similar in operation to a sled impact).

ii. **Punch**- actuator impacts the target and retracts such that a single pulse is created.

iii. **Flyer**- actuator operates similar to the punch in order to launch a free-flying projectile that impacts the target. The actuator is retracted before any contact is made with the target.

It is the third configuration that is of interest to this investigation as it offers the most realistic pressure loading profile to replicate a close-in detonation. Thus, development of the non-explosive test methods for wide area impacts on flexible sandwich panels relies on the actuator accelerating a projectile to create the desired impact characteristics.

It should be mentioned that despite the use of the high speed actuators (fast for servo-hydraulics), the Blast Simulator still operates at velocities two orders of magnitude lower than the shockwave velocity of a high explosive. To overcome this limitation, researchers often rely on impulse matching. In this technique, the area underneath a high pressure, short time duration pressure pulse curve (impulse) is replicated using a lower pressure and longer time duration pressure pulse (see Figure 2.5). Impulse matching is
not unique to the Blast Simulator and can be used with any of the explosive and non-explosive methodologies previously discussed. As the overall pressure pulse duration is on the order of a few milliseconds, impulse matching with the Blast Simulator is still within the realm of a high-rate dynamic impact event which allows the system to “simulate” the strain rate effects excited by an actual explosive detonation.

![Idealized impulse matching for two pressure loading profiles](image)

**Figure 2.5. Idealized impulse matching for two pressure loading profiles**

Impulse matching with the Blast Simulator has been used to study various masonry walls, high performance blast-resistant walls, concrete columns, steel columns, and composite structures since its introduction in 2005 [31 – 39]. The damage modes and the extent of damage were verified through finite element analysis and actual blast tests. In these investigations, the Blast Simulator was used successfully to apply high impulse,
short duration dynamic loads similar to a high explosive detonation but without the shockwave or debris cloud obscuring the high speed cameras. It offered high repeatability for comparison studies and relatively safe operation. These latter benefits cannot be emphasized enough. Blast testing with explosives is notoriously inconsistent and poses significant challenges to researchers needing repeatable and controlled data sets. Quantifying the performance of small variations in armor design, for example, cannot easily be obtained if consistent application of the pressure load due to a HE detonation cannot be achieved. The non-explosive Blast Simulator develops repeatable pressure pulse events, has excellent visibility, and allows multi-channel real-time sensor measurements that would otherwise be impossible in more traditional explosive testing.

The benefits of the UCSD Blast Simulator was first demonstrated by Rodriguez-Nikl [31, 32] who captured the first ever recordings of shear crack formation and spalling of concrete covers in his 2006 investigation of reinforced concrete columns (both as-built and carbon composite wrapped) subject to an impulse of 6,800 – 15,700 Pa-s. This impulse level was equivalent to a vehicle blast load from 560 kg high explosives, 0.9 m off the ground, and standoff distances 3.5 – 6.1 m. Stewart [33, 34] examined steel columns, primarily W10x49 and W14x132, subject to a similar vehicle blast loading (450 kg HE at standoff distances 1.0 – 7.6 m) in both strong and weak axis and found good agreement in both damage mode and extent of damage to finite element simulations and actual blast tests. Oesterle [35] investigated impulses in the range of 1,000 – 2,000 Pa-s on concrete masonry walls with various reinforcements while Freidenberg [36] investigated high strength steel stud-sheathing prototype wall systems developed
specifically for hostile environments. All of these investigations simulated standoff distances of several meters and equivalent charge sizes of several hundred kilograms to essentially produce planar shockwaves representative of vehicle bombs rather than the spherical close-in detonations typical of a buried IED.

Close-in detonations were briefly discussed by Wolfson [37], who examined cellular steel structures 1.07 m wide with TNT charges placed at standoff distances less than 330 mm. This close-in blast sheared circular sections of the steel plate below the explosive and propelled it toward the opposite wall in the cellular steel structure. The Blast Simulator tests were used to replicate the plate-structure interaction by launching a representative steel plate to impact a steel panel. Attempts to recreate the close-in pressure pulse detonation itself were not pursued. Huson [38] demonstrated the use of shaped water-filled bags to load balsa wood core composite sandwich structures and joints. The focus was on creating a short duration, uniform pressure pulses for detonation standoffs less than a few meters [39] rather than replicating the close-in spherical pressure loading profile that is of interest in this investigation.

In summary, explosive charges, shock tubes, gas guns, and the Blast Simulator are all capable of creating the high overpressure, short time duration impulse typical of a HE detonation. Each method has its benefits and limitations, which are qualitatively presented as a radar plot in Figure 2.6. The perimeter of the plot represents the best performance and center represents low performance within the context of this investigation. For example, explosives are capable of wide area pressure loads with short time duration (1/t favors short time periods) on large targets, but have limited visibility of
the impact event due to the dust and debris field, low repeatability and control of the pressure pulse application, and are relatively dangerous. On the other side of the spectrum, gas guns offer good control of the applied pressures and good visibility during impact with safe operation, but cannot generate wide area loads on the large sized targets typical of a HE event. Shock tubes, depending on size, offer performance in between the gas gun and actual explosive detonation as they use dynamic pressure pulse waves to load their targets. The Blast Simulator offers unique capabilities to recreate large panel deformations non-explosively that combines the benefits of high repeatability, visibility, and safety of the gas gun impacts with the wide area pressure pulses and large size targets more common in actual explosive tests. Previous investigations with the Blast Simulator typically focused on generating large planar pressure pulses to replicate vehicle car explosives at several meters standoff distance. For this investigation, the capabilities of the system to generate short duration, spherical loading profiles typical of a close-in detonation will be examined. This will help validate and define the extent of using the impulse matching techniques to replicate blast-level damage on flexible armor sandwich panels and will be the first project on the Blast Simulator to attempt temporally and spatially varying pressure pulses using a single projectile package.
Figure 2.6. Radar plot comparison of dynamic pressure pulse generation methods

Parts of Chapters 2, 3, and 4 have been accepted for publication by the International Journal of Impact Engineering, 2014, D. Whisler and H. Kim. The dissertation author was the primary investigator and author of this material.
This chapter details the evolution of the large panel non-explosive methodology though a series of related research projects. This includes the large panel gas gun experiments, a foam material study for pressure pulse shaping, the projectile development for the large panel non-explosive test method, and the transmission plate design for both non-explosive and actual blast tests.

3.1 PRELIMINARY GAS GUN TRIALS (GGUN TEST SERIES)

The first set of Armorworks panel tests commenced with the gas gun trials in winter 2009. The 79 mm diameter gas gun was capable of launching a 70 mm diameter, 104 mm long, 2.76 kg steel projectile at speeds up to 74 m/s (see Figure 3.1). As with all gas guns, the target area was limited by the barrel diameter (and projectile) which represented a concentrated point load compared to the 610 x 610 mm panels. In order to achieve a wide area pressure pulse as opposed to the ballistics-style projectile impact, a spreader plate system consisting of aluminum and foam layers was devised to spread the dynamic force load from the 70 mm diameter projectile over a much wider 305 x 305 mm area. The 160 kg/m³ rigid polyurethane foam and 6061-T6 aluminum plates were layered as follows: 76.2 x 76.2 x 25.4 mm foam, 152.4 x 152.4 x 12.7 mm aluminum, 304.8 x 304.8 x 12.7 mm aluminum, 304.8 x 304.8 x 25.4 mm foam (see Figure 3.2).
Figure 3.1. Overview of the gas gun tests

Figure 3.2. GGUN test DHC01 showing the layered spreader plate system and double honeycomb panel
The original $J_{sp}$ of the gas gun investigation was 4,715 Pa-s, equivalent to 1.0 kg TNT at 305 mm standoff. After a few preliminary tests, the safe limit operation of the gas gun was found to be less than 50% of this value (2,093 Pa-s) with additional, uncalculated losses due to the momentum transfer between the projectile and spreader plate system. Although the impulse levels and losses were not ideal for creating blast-like damages, they would still provide a consistent loading for assessing the relative performance of the various panels, which met one of the required program outcomes for Armorworks. Thus, the gas gun method was acceptable for these early panel tests, especially for identifying the best performing sandwich construction designs.

One of the test metrics for comparing panel performance was the transmitted impulse of the various panel designs measured using a “transmission plate”. The 4.05 kg 305 x 305 x 12.7 mm thick aluminum plate had an accelerometer installed at the center of gravity (CG). The idea, based on similar plates used by blast researchers, was that the impulse not absorbed or attenuated by the armor panel would captured by the transmission plate and used to accelerate it a measurable velocity. Panels with high impulse absorption would accelerate the plate to a lower velocity compared to panels with a low impulse absorbing design (for additional details of operation, see Chapter 3.4).

The transmission plate rested on low friction blocks and was bonded to the back of the panel with a small quantity of cyanoacrylate (see Figure 3.3). On impact, the deforming armor panel accelerated the transmission plate, which released from the back of the panel at peak deformation and continued at a constant velocity until it was arrested a short time later through the use of nylon straps and a friction catch mechanism. In the
event that data was lost due to cable failure or connector issues with the accelerometer, high-speed video was used to calculate the plate velocity.

Armorworks supplied 18 different sandwich panels including three RHA monolithic plates which are summarized in Table 3.1. Panel identification was provided by Armorworks via five character names, where the first letter indicated single layer core (S) or double layer core (D) followed by two characters for the core material, either aluminum foam (AC) or stainless honeycomb (HC), and two digits for the variant. By this naming convention, SAC01 is a single layer, aluminum foam core sandwich panel.
and the first variation. RHA panels were identical 6.35 mm thick monolithic plates. Four additional ceramic panels were tested using the 306x designation, but details of their construction were not available as they were supplied by a different manufacturer.

Table 3.1. Armor panel designs for gas gun tests

<table>
<thead>
<tr>
<th>Panel ID</th>
<th>Front Face Mat (mm)</th>
<th>Layer 2 Mat (mm)</th>
<th>Layer 3 Mat (mm)</th>
<th>Layer 4 Mat (mm)</th>
<th>Layer 5 Mat (mm)</th>
<th>Layer 6 Mat (mm)</th>
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<tr>
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<td>AS</td>
<td>6.35</td>
<td>SP</td>
<td>3.18</td>
<td>Fm</td>
<td>10.0</td>
</tr>
<tr>
<td>DAC02</td>
<td>AS</td>
<td>6.35</td>
<td>Fm</td>
<td>10.0</td>
<td>TF</td>
<td>3.18</td>
</tr>
<tr>
<td>DAC03</td>
<td>TF</td>
<td>6.35</td>
<td>SP</td>
<td>3.18</td>
<td>Fm</td>
<td>10.0</td>
</tr>
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<td>DAC04</td>
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<td>6.35</td>
<td>Fm</td>
<td>10.0</td>
<td>TF</td>
<td>3.18</td>
</tr>
<tr>
<td>DHC01</td>
<td>AS</td>
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<td>SP</td>
<td>3.18</td>
<td>Hn</td>
<td>7.62</td>
</tr>
<tr>
<td>DHC02</td>
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<td>SP</td>
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<td>Hn</td>
<td>7.62</td>
</tr>
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<td>SP</td>
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<td>10.0</td>
</tr>
<tr>
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<td>12.7</td>
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<tr>
<td>SHC02</td>
<td>TF</td>
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<td>SP</td>
<td>3.18</td>
<td>Hn</td>
<td>12.7</td>
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<td>RH</td>
<td>6.35</td>
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<td></td>
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<td>306x CERAMIC</td>
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<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
</tbody>
</table>

*Front face and sandwich panel layers are abbreviated as: solid aluminum (AS), T-flex dry aramid fiber (TF), polymer shock layer (SP), RHA (RH), aluminum foam (Fm), and stainless steel, square honeycomb (Hn). The original foam densities and honeycomb cell size were not obtained.

Each gas gun test began by first securing a panel in the window fixture with twenty 12.7 mm diameter bolts. The fixture was made from 12.7 mm thick A36 steel and stiffened with tubular steel having a 6.35 mm wall thickness. A spreader plate package
was installed on the panel front face and secured with clear adhesive tape. The transmission plate was installed on the back face of the panel and all data acquisition systems set to capture. Instrumentation consisted of a dual laser photogate for capturing the projectile velocity, a Dytran 3200B 100,000 g shock sensor to record transmitted acceleration (and through integration, velocity), and dual Phantom V7.3 high speed cameras running at 3,000 frames per second (fps). The projectile was then breech loaded and two layers of 0.05 mm thick Mylar membranes were installed behind the projectile and in front of the expansion chamber. The purpose of the Mylar was to collect the gas in the expansion chamber before suddenly bursting which provided a more consistent and slightly higher projectile velocity (compared to a non-Mylar test, see Appendix E). Then the main nitrogen fill tank was pressurized to 4,212 kPa (610 psi) and the helium actuated release valve pressurized to 1,000 kPa (150 psi). On triggering, the helium ball valve opened, releasing the high pressure nitrogen into the expansion chamber. The Mylar membranes burst and the sudden release of gas accelerated the projectile down the 2.26 m long barrel and through a dual laser photogate velocity measurement box. The projectile continued along its 0.33 m free flight before impacting the specimen (see Figures 3.4 and 3.5).
Figure 3.4. Gas gun system showing reservoir, ball valve, and barrel

Figure 3.5. GGUN projectile striking RHA panel
On impact, the projectile crushed the spreader system and momentarily came to rest while the panel deformed. Since the projectile stopped, it was assumed that the momentum (2.76 kg at an average velocity of 70.5 m/s) over the 305 x 305 mm impact area would provide an impulse of 2,093 Pa-s. The transmitted impulse was recorded by the transmission plate system (see Figure 3.6).

Figure 3.6. GGUN transmission plate flight for RHA panel

Impulses for the GGUN tests are summarized in Table 3.2. The transmitted impulse column was calculated primarily from the accelerometer data, although some
instances used the high speed camera (e.g., GGUN test SAC03). The baseline RHA steel panels were subject to an average impulse of 184.7 N-s (42% of the desired low pressure impulse) and accelerated the transmission plate to average velocity of 14.5 m/s (impulse is 58.11 N-s). Based on input from Armorworks, a performance index (Equation 3.1) was created that favored high impulse absorption and low areal density.

\[
PI = 100 \cdot \frac{J_{\text{Projectile}} - J_{\text{Transmission}}}{J_{\text{Projectile}} - AD}
\]  

The baseline RHA performance index, when accounting for the relatively high areal density of steel, averaged 1.40 m²/kg. All panels except 306C had a higher performance index compared to the RHA steel plates, which they achieved primarily due to their low areal densities. For example, Armorworks panels SAC01 and SAC02 both had a transmitted impulse lower than steel but their performance indexes of 2.10 m²/kg for SAC01 and 1.93 m²/kg for SAC02 were higher than that of RHA due to their lower areal density. The second highest performing panel, SHC02, had the second highest transmitted impulse at 90.69 N-s, but coupled with an areal density less than half of the baseline RHA steel, gave it the performance index of 2.55 m²/kg. Finally, all three panels with permanent deformations at the panel center (dent depth) greater than 10 mm, SAC03, DAC03, and DAC04, had high performance indexes. No permanent deformation was recorded after impact at the panel center for RHA steel (dent depth of zero).
Table 3.2. GGUN test results

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Panel ID</th>
<th>AD (kg/m²)</th>
<th>Velocity</th>
<th>Impulse</th>
<th>Perfor. Index</th>
<th>Dent depth</th>
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<tbody>
<tr>
<td></td>
<td></td>
<td>Flyer (m/s)</td>
<td>Trans. (m/s)</td>
<td>Flyer, F (N-s)</td>
<td>Trans., T (N-s)</td>
<td>(F-T)/F (%)</td>
</tr>
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<td>DAC01</td>
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<td>15.04</td>
<td>183.2</td>
<td>15.04</td>
<td>1.82</td>
</tr>
<tr>
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<td>34.96</td>
<td>66.99</td>
<td>17.92</td>
<td>184.6</td>
<td>17.92</td>
<td>1.74</td>
</tr>
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<td>28.12</td>
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<td>19.76</td>
<td>202.3</td>
<td>19.76</td>
<td>2.15</td>
</tr>
<tr>
<td>DAC04</td>
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<td>73.37</td>
<td>20.19</td>
<td>202.2</td>
<td>20.19</td>
<td>2.23</td>
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<tr>
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<td>16.0*</td>
<td>2.34</td>
</tr>
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<td>199.9</td>
<td>17.4*</td>
<td>1.98</td>
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<td>202.3</td>
<td>14.26</td>
<td>1.70</td>
</tr>
</tbody>
</table>

* High speed camera velocity

Although several aluminum foam and stainless steel honeycomb core sandwich panels were identified as high performing, the gas gun system could not produce the high impulse, wide area pressures necessary to replicate the original blast detonation. It was apparent that a new projectile design was necessary to increase both impulse and contact area while imparting a more spherical blast-like pressure load on the armor panels.
3.2 Development of the Polyurethane Foam Pulse Shapers

Pressure pulse tuning for a dynamic impulse event is necessary to ensure that the desired loading curve is applied by the projectile to the target. For a flexible armor panel target, the projectile impact should engage the panel in a manner that maintains contact for the entire duration of the loading pulse as the panel deforms (no repeated hits). The projectile should accommodate some impact misalignment, as its free flight will not always be perfect or planar. In instances of metal-on-metal impacts, the projectile should reduce the high frequency noise or “ringing” in the accelerometer data while also providing similar loading characteristics in the event of non-metallic impacts.

While a significant portion of the projectile loading characteristics depend on the projectile design itself (which is discussed in depth in the next section), the pulse shaping media layer that first makes contact with the target is an integral part of the loading process. Previous investigations with the Blast Simulator addressed this issue by relying on a hard elastomer, Adiprene L-100/MOCA, to serve as the intermediate layer between the solid metal projectiles and the target structure (see Figure 3.7). The elastomer layer was specifically molded into a matrix of tetrahedrals that deformed on impact to influence both the shape and the time duration of the pressure pulse curve [40]. Although the viscoelastic material eliminated the high frequency content of the loading pulse and was re-useable for multiple impacts, it was relatively heavy and overly stiff for impacting the flexible armor panels since it was designed for much larger structures (e.g., concrete and steel columns). Additionally, finite element modeling of the viscoelastic material was computational expensive and had instabilities due to mesh distortion at large strains.
(> 40%) [40]. These shortcomings combined to prior experience using tuned foam projectiles led to the development of a polyurethane foam-based pulse shaping media.

The General Plastics Last-A-Foam part number FR-7110 polyurethane foam was the same foam used for the GGUN spreader plate system. The rigid, machinable
polyurethane foam was purchased in sheets 1,220 x 610 x 25.4 mm in size. The manufacturer reported [41] density is 160 kg/m$^3$ and the compressive strength measured at 24° C is 2.16 MPa parallel and 1.70 MPa perpendicular to the cell rise direction.

The typical compressive loading profile for polymer-based foams is divided into three distinct stages: the initial elastic response, the constant stress regime, and the densification stage (see Figure 3.8). In the elastic loading portion, the foam behavior is governed by the compression and bending of the cell walls and (for closed-cell foams) the compression of trapped gases within the cell walls [42, 43]. The volume of the foam material may be recovered during the elastic loading regime by applying a tensile force to the foam [44]. However, should compression continue, the cell walls will buckle, rupture, and plastically deform as the foam exceeds its yield stress (which is lower than the yield stress of the base solid polymer) and reaches the “collapsing” or plateau stress ($\sigma_C$). The collapsing stress is of particular interest, as it is relatively constant over a large range of increasing strain. This regime is where the foam is able to absorb considerable energy. In some cases, especially when loading in the cell rise-direction, strain softening occurs whereby the stress decreases slightly immediately following the elastic loading phase [42]. Once the cells have fully collapsed and reach the “locking” strain ($\varepsilon_L$) the foam densifies and behaves similar to the solid material from which the foam was originally produced [45]. For shock mitigation, the polymer foams, including polyurethane, provide desirable initial loading characteristics as they do not allow high stress transmission until very high strains, well after densification has been reached [46].
The stress-strain profile shown in Figure 3.8 for polyurethane foam is desirable in order to tune dynamic projectile impacts since stresses greater than the collapse stress cannot be transmitted to the supporting structure until compaction occurs at high strain levels [46]. This means that for a sufficiently low collapse stress, the pulse shaping projectile can impact flexible targets and correct for any projectile-to-target misalignments (e.g., due to projectile flight or specimen irregularity) without causing high contact stresses that may prematurely load the structure. Once engaged and with a properly aligned loading path, the densification of the foam enables the projectile to transfer its momentum to the target panel at a higher contact pressure while maintaining contact throughout the loading phase. This reduces the likelihood of double hits, high frequency ringing in the data collection systems caused by metal-to-metal contact, and provides the means for projectile loading correction for non-perfect flight. Careful tuning
of the crushable foam layer thickness and geometry, especially the transition to the densification stage, is necessary for achieving an ideal pressure pulse shape.

The onset of densification for the polyurethane foam used in this particular dynamic loading application is influenced by two main factors: the material properties and physical geometry of the foam itself. Concerning the material properties, studies suggest that the onset of densification (locking strain), collapse stress, and elastic modulus are all strain rate sensitive [45] for both open and closed cell foams. This is due to the viscoelastic behavior of air inside the foam that under loading, must be compressed and forced outside the collapsing material [45, 47]. Supporting stress-strain data at dynamic (> 1,000 1/s) strain rates is sparse [48, 49] and in some cases, contradictory [45]. For example, while strain rate sensitivity is expected [48, 50] certain polyurethane foams do not exhibit such behavior [45, 47]. Also, although it is typically necessary to apply hydrostatic stress states to obtain material behavior in order to create accurate finite element material models, certain polyurethane foams do not exhibit a strong enough sensitivity to the confinement pressures to warrant the hydrostatic testing over basic uniaxial compression testing [51, 52].

Concerning the physical geometry of the foam itself, introducing shaped foam projectiles has been shown to change the pressure pulse characteristics [46, 51], although systematic studies comparing different shapes and their pressure pulse characteristics are often not available. This uncertainty and incomplete data at various strain rates can be attributed in part to the difficulty in testing low impedance foam materials at high strain rates [45] (see Appendix H). Nonetheless, the current body of research highlights the
need for experimental testing to verify and validate the specific foam material properties used in testing, and especially for different physical geometries under consideration. Thus, a study was conducted to examine the foam materials and the geometries, first to support the pressure pulse shaping mechanism for the Blast Simulator projectile package and later, to support material models for finite element analysis (see Chapter 6). Since the anticipated contact stresses (based on GGUN test observation) would easily exceed the collapse stress of the foam and the Blast Simulator would only be launching the projectile within a narrow range of velocities, the initial focus was on the shaped geometries. Furthermore, because nearly 1,500 foam blocks would be required for the Blast Simulator tests, it was necessary to select geometries that would facilitate rapid production.

Three foam geometries were examined: rectangle, V channel, and tetrahedral. The 64 x 64 x 25.4 mm base foam unit could be machined into each shape through an increasing number of planar cuts—zero for the rectangle, two for the V channel, and four cuts for the tetrahedral. The V channel and tetrahedral specimens faces were cut to a height of 12.7 mm with the remaining 12.7 mm being prismatic rectangles (see Figure 3.9). Each foam shape was tested dynamically with the pendulum impactor (up to 7.30 m/s) and the gas gun from the previous GGUN test series (up to 28.8 m/s). The pendulum impactor used a Dytran 1060v5 force sensor suspended at the end of a 1.40 m arm along with 4.55 kg steel mass. Three different velocities were tested: 40 J at 4.2 m/s, 58 J at 5.0 m/s, and 121 J at 7.30 m/s. The impact event was captured with a Vision Research Phantom V7.3 high speed camera at 10,000 fps (see Figure 3.10).
Figure 3.9. Shaped foam specimens having base dimensions 64 x 64 mm

Figure 3.10. Pendulum impact test on a V channel specimen
Testing at higher velocities and strains required using the gas gun in conjunction with a 75 mm diameter, 3.19 m long aluminum Hopkinson bar that permitted up to 1.1 ms of force data capture (see Figure 3.11). Projectiles were solid aluminum bodies with hard plastic front tips onto which the shaped foam materials were mounted. The resulting 875 g projectile was launched at 28.8 m/s, which was selected to provide a direct comparison with the anticipated Blast Simulator projectile velocity. The gas gun tested foam pulse shapers were smaller than the final design (64 x 64 mm vs. 76.2 x 76.2 mm) to fit inside the gas gun barrel.

Force data were captured using dual 1,000 Ω strain gages (gage factor 2.08) installed 180° apart on the Hopkinson bar in a full Wheatstone bridge configuration. The signal was passed through a Vishay 2310B signal conditioning amplifier set to a gain of 100 and an excitation of 10 V. The output was connected to a Picoscope 3424 data acquisition system (DAQ) set to capture at 1 MHz. As before, the Phantom V7.3 high speed cameras captured the impact event at 10,000 fps. Direct observation of the impact event showed the original 25.4 mm foam height was not achieving complete crush due to the amount of total deformation (and energy absorption capacity) available to the 25.4 mm thick specimens prior to reaching the locking strain. An identical set of tests were conducted with the prismatic base for all three specimens reduced by 11 mm. This resulted in 64 x 64 x 14 mm tall specimens, which for the V channel and tetrahedral specimens, was almost entirely the shaped region at the front of the foam block (see Figure 3.11). Decreasing the height of the specimens in this manner increased the amount of core crush for all three specimens.
Figure 3.11. Hopkinson bar test with 14 mm thick tetrahedral shaped foam at 28.8 m/s
Examining the pendulum impact force data, it was readily apparent that the rectangular foam was not ideal due to its high frequency ringing. It also had a very sharp initial loading pulse, which would normally be preferred for creating a sudden impulse event, but would result in multiple impacts on the flexible targets (see Figure 3.12). Both V channel and tetrahedral showed a finite rise time where the foam crushed down before achieving maximum force, but the tetrahedral had a concave force time history uploading curve which was preferred as it signified that loading was constantly increasing. The V channel had a convex force time history which was undesired as a decreasing applied loading rate would allow the target panel to separate (outrun) the impinging projectile. Thus, the tetrahedral shape appeared to be an ideal candidate for the projectile, but the pendulum impactor did not have sufficient energy to fully crush the specimens in order to assess their performance at higher strains. Gas gun tests were therefore conducted.

For the gas gun tests, differences between the V channel and tetrahedral foam initial force response specimens were less pronounced at 28.8 m/s and both offered a more progressive crush response than the rectangular shaped foam block, as shown in Figure 3.13. The tetrahedral foam showed a higher peak force compared to the other shaped geometries, a trend that was consistent with the 7.30 m/s pendulum impact tests (compare Figures 3.12 with 3.13). The 14 mm thick specimens, which experienced a more complete crush in the shaped regions, emphasized the trends observed for the 25.4 mm thick specimens, namely, that the tetrahedral provided a lower force during initial contact engagement, a sharper peak densification regime, and a higher peak contact force at full densification when compared to the other two shaped foam geometries (see Figure
These qualities made the tetrahedral the preferred geometry for the Blast Simulator projectile, which coincidentally, was already employed on the elastomer layer design.

Figure 3.12. Force time history for 7.30 m/s pendulum impacts on all three shaped foam geometries
Figure 3.13. Force time history for 25 mm thick shaped foam geometries at 28.8 m/s

Figure 3.14. Force time history for 14 mm thick shaped foam geometries at 28.8 m/s
At this stage, the foam shapes could be machined in mass. Several sheets of 610 x 1,220 x 25.4 mm foam boards were obtained, and the foam cut via band saw into 76.2 x 1,220 x 25.4 mm strips, then 76.2 x 76.2 x 25.4 mm blocks (see Figure 3.15). An aluminum jig was made to hold two foam blocks at a precise, 19.3° angle with respect to the vertical plane for cutting the faces of the tetrahedral. The jig was attached to a guide rail and slid past the vertical band saw blade to cut each angled face (see Figure 3.16). Four passes in total were required with the foam block rotated 90 degrees each time to create the tetrahedral foam shapes.

Figure 3.15. Tetrahedral foam pulse shaper (dimensions in mm)
3.3 Large Panel Non-Explosive Projectile Design

The Blast Simulator projectile design commenced in mid 2010 with a careful assessment of the design constraints. The test fixture offered a 483 x 483 mm planar loading area for the projectile within the window frame (see Figure 3.17). The BG25 actuators were capable of 25 m/s velocity when launching a 300 kg mass and up to 35 m/s when launching a smaller, 50 kg mass [31]. To facilitate faster setup between tests and keep operation costs at a minimum, the projectile was required to be maneuverable by a 1 – 2 person team without using heavy machinery. These constraints resulted in a 60 kg, 406 x 406 mm projectile package to be launched at 27.8 m/s, providing a $J_{sp}$ up to
10,134 Pa-s. The design permitted the projectile to be handled without the need for a crane or hoist. The projectile dimensions allowed for a 38 mm gap between the edge of the projectile and the window fixture, a necessity as the projectile would be in free flight with possible misalignment prior to impact.

Figure 3.17. Test fitting panel inside window fixture (scale shown in inches)

With the general projectile shape determined, work commenced on generating a pressure pulse loading profile. Previous investigations with the Blast Simulator used steel and aluminum solid projectiles known as “programmers” with shaped elastomer front layer. The mass of these programmers provided the necessary momentum to
generate the high impulses while the shaped elastomer layer help tune the pressure pulse. This design served as the foundation for the projectile package, with the exception of using polyurethane foam to replace the elastomer layer. As verifying the pressure loading profile was not possible with a physical test prior to actually setting up for the real tests, a model of the impact event was created using the Abaqus/Explicit v6.10-EF1 finite element analysis software [53]. The target panel was a 6.35 mm thick RHA steel plate, which was anticipated to provide the least deformable surface (based on experience with the GGUN trials) and subsequently, the highest contact forces that the fixtures were required to withstand. The model included a large majority of the actual test components, which was necessary to verify the peak stresses in the different components to ensure their survival during the actual tests. To reduce computation time, the model was reduced to a quarter symmetric representation of the actual system (see Figure 3.18). Full details of the finite element simulation are provided in Chapter 6 as the focus at this stage of projectile design was the general pressure pulse distribution profiles.

The results of the finite element study showed that the initial contact pressures due to the projectile impacting the steel plate were uniform. Within a short time period thereafter, the panel responded with a deformation profile that was highest at the panel center (due to its position furthest from the window fixtures) and lowest at the boundary with the window fixtures. Thus, as the rigid monolithic projectile continued to drive into the panel, the boundary-adjacent regions had less deformation and subsequently experienced a higher loading compared to the center of the panel. This resulted in the high stress concentrations at the corners and edges as shown Figure 3.18. The observed
stress concentrations was not characteristic of an actual blast load which had a more uniform distribution throughout the short pulse duration, and for a close-in detonation, the highest contact pressures would be at the panel center. The effects of this non-blast pressure profile due to a solid projectile would be worse for the sandwich panels which were expected to have higher center deformations (and therefore less contact with the projectile) than the RHA steel plate. Thus, the monolithic projectile, even with the addition of the compliant layer of pressure pulse shapers, was not sufficient to replicate the close-in loading profile of a high explosive detonation on the flexible armor panels.

Figure 3.18. Solid projectile showing high contact stresses at the panel corner and boundary (left) compared to a tiled projectile showing contact stresses more evenly distributed toward the panel center (right)
Since the non-uniform pressure pulse distribution was a result of the deforming target panel losing contact with the rigid solid block projectile, then creating a projectile as a tiled array of discrete blocks rather than a single monolithic mass would enable the projectile to dynamically conform to the deforming panel. On initial contact, the projectile would be planar to uniformly load the panel surface in a similar manner as the solid projectile design. As the panel deformed in a curved shape with the highest displacement at the center, the individual tile blocks would follow the curvature by rotating and differentially translating as necessary to maintain contact. This would allow the panel to be uniformly loaded with pressure and would reduce the concentration of loading to the stiffest path at the edges and corners (see Figure 3.18).

The number of tiles covering a desired area could be adjusted such that the pressure pulse spatial variations could be controlled, e.g., to achieve close-in blast pressure pulse distribution. However, as assembly time for many individual blocks constituting a projectile package would increase and delay testing, a compromise was made by using 25 steel blocks, each 2.4 kg with a 76.2 x 76.2 mm areal footprint and 50.8 mm in height. The block were arranged in a 5 x 5 tile pattern, with a 6.35 mm gap between adjacent blocks (impact area approximately 406 x 406 mm) and a 38.1 mm gap between the outermost projectile tiles and the panel window fixture (inner dimension 483 x 483 mm). This spacing allowed each block to have a degree of independent rotation and translation without striking the window fixture or another block.

Unlike the idealized finite element analysis, the real tile blocks were not all identical (see Figure 3.19). The center block required more design work to serve as both
the attachment point for the projectile assembly and to physically support the other 24 steel blocks when loaded on the pusher plate. The purpose of the single attachment point was to minimize contact with any supporting component which may have interfered with its release and free flight characteristics. Often multiple attachments bind or have different levels of friction, resulting in an uncontrollable alignment during release at high speed and making consistent test conditions a challenge. The solution was to use a single 25.4 mm diameter precision shaft to support the projectile at the projectile package’s center of gravity. The center block of the projectile was bored to accept a bronze sleeve bushing which would fit over the support shaft to an insert depth of less than 25 mm. On release, the entire projectile would smoothly slide off the support shaft and enter free flight.

One minor detail that may not be obvious was that the shaft, bushing and center tile were precision machined and matched such that they would actually form an airtight seal. Loading a 60.3 kg projectile would be a challenge as the seal would prevent the locating shaft from inserting into the bushing. The release at high speed would also be affected if breaking the seal was not a consistent and repeatable process. To address this problem, four 9.53 mm diameter outgas ports were drilled through the center tile block, one on each side face (see Figure 3.19). The outgas ports connected to the bored hole for the bushing and were located ahead of the fully inserted shaft. Inserting or unloading the center tile block from the shaft would force air through the outgas ports and permit unrestricted sliding.
To hold the blocks in place and allow the projectile package to be handled as a single unit, all 25 individual tiles were fastened to a 3.18 mm thick aluminum sheet (see Figure 3.20). The aluminum sheets were waterjet cut to 508 x 406 mm with the longer side serving as an attachment for the assembly process. Holes were also waterjet cut on the face of the sheet, one per tile block and five for the center block. The 24 tile blocks each had one 1/4-20 tap (threaded hole) on their mounting surface which would accept a bolt passed through the aluminum sheet to hold the block at its correct position (see Figure 3.19). The center block, which was bored in the center to clear the locating shaft, used four tapped holes near its corners to attach to the aluminum sheet and support the
weight of the projectile package. The thin aluminum sheet would be a consumable item as it was designed to plastically deform on impact.

Figure 3.20. NEM2 aluminum projectile sheet prior to assembly showing bolt locations and notches

After each test, the blocks were cleaned and reinstalled for the subsequent tests. However, during the first set of NEM1 tests, a high number of the tiles were shearing their attachment bolts despite the use of a relatively weak aluminum sheet. The sheared bolts required a time consuming extraction process which slowed the overall rate of testing. For the second series of Blast Simulator tests, the aluminum backing sheet was
notched at each bolt location (as shown in Figure 3.20). This would allow the bolt heads to pull away from the aluminum sheet which not only prevented shearing but also increased the amount of independent motion of the tile block and thereby allowed less constrained interaction with the target panel.

Although the tiled projectile design served its purpose to provide a wide area pressure pulse loading on the flexible armor panels, it did not fulfill the other characteristics of a close-in spherical blast detonation, specifically, a spatially and temporally varying pressure time history profile. The desired spatially varying pressure pulse distribution was provided by Armorworks and via calculations from the ATBlast software package in 25.4 mm intervals along the entire 610 x 610 panel surface [54]. Recalculating this distribution as a percentage of the highest impulse at the panel center is shown in Figure 3.21 along with a superimposed quarter-section outline of the unique tile blocks constituting the projectile system. These regions, labeled A – F use the weighted average of the distribution map to compute the required theoretical impulse levels for the actual tile blocks impacting each region.

Since the projectile package of blocks were launched via a flat pusher plate as a single unit, and impulse is the product of mass and velocity, then achieving a spatially varying impulse distribution for a constant velocity could be accomplished by adjusting the mass of the individual tile blocks. The center block would be the heaviest and each ring of blocks around the center would be made lighter. This was achieved by using different amounts of steel and aluminum to form the tile blocks as per Table 3.3, where each block label matched the region mapped in Figure 3.21. To facilitate assembly and
reduce the number of unique tiles, two sets of blocks were combined: B/C and D/E. Thus, rather than create two unique blocks B and C with a 78.1% and 82.0% of maximum impulse, respectively, a single block B/C was designed to have a mass of 1.858 kg (84.0% of maximum impulse). Similarly, tile D at 89.3% and tile E at 94.1% of maximum impulse both used D/E block design with an anticipated mass of 2.071 kg, equivalent to 92.0% of maximum impulse. The center block F was designed to apply the maximum impulse, 100%, using an all-steel construction.

Figure 3.21. Impulse distribution for replicating close-in blast detonation on panel
Table 3.3. Ideal impulse distribution

<table>
<thead>
<tr>
<th>Location</th>
<th>Required Impulse (Pa-s)</th>
<th>Ideal Design</th>
<th>Predicted</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(%) Steel (mm) Alum (mm)</td>
<td>Mass (kg)</td>
<td>Impulse (%)</td>
</tr>
<tr>
<td>A</td>
<td>5,811 70.0 25.40 25.40</td>
<td>1.504</td>
<td>68.0</td>
</tr>
<tr>
<td>B</td>
<td>6,490 78.1 31.75 19.05</td>
<td>1.681</td>
<td>76.0</td>
</tr>
<tr>
<td>C</td>
<td>6,809 82.0 38.10 12.70</td>
<td>1.858</td>
<td>84.0</td>
</tr>
<tr>
<td>D</td>
<td>7,384 88.9 44.45 6.35</td>
<td>2.035</td>
<td>92.0</td>
</tr>
<tr>
<td>E</td>
<td>7,816 94.1 45.72 5.08</td>
<td>2.071</td>
<td>93.6</td>
</tr>
<tr>
<td>F</td>
<td>8,306 100.0 50.80 0.00</td>
<td>2.212</td>
<td>100.0</td>
</tr>
</tbody>
</table>

The actual tile blocks did not have the exact theoretical masses due in part to the additional steel bolts required to attach the aluminum and steel sections together. Also, for the center block F, the bored hole required for the bushing and outgas ports resulted in less mass than block D/E (2.09 vs. 2.15 kg). Block B/C was 1.92 kg and block A was 1.54 kg. The actual block masses are shown in Table 3.4 along with the percent difference from the anticipated masses.

Table 3.4. Actual tile block mass distribution

<table>
<thead>
<tr>
<th>Location</th>
<th>Steel (mm)</th>
<th>Aluminum (mm)</th>
<th>Mass (kg)</th>
<th>Difference from Predicted (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>25.4</td>
<td>25.4</td>
<td>1.54</td>
<td>2.32</td>
</tr>
<tr>
<td>B</td>
<td>38.1</td>
<td>12.7</td>
<td>1.92</td>
<td>12.4</td>
</tr>
<tr>
<td>C</td>
<td>38.1</td>
<td>12.7</td>
<td>1.92</td>
<td>3.21</td>
</tr>
<tr>
<td>D</td>
<td>44.45</td>
<td>6.35</td>
<td>2.15</td>
<td>5.34</td>
</tr>
<tr>
<td>E</td>
<td>44.45</td>
<td>6.35</td>
<td>2.15</td>
<td>3.69</td>
</tr>
<tr>
<td>F</td>
<td>50.8</td>
<td>0</td>
<td>2.09</td>
<td>-5.85</td>
</tr>
</tbody>
</table>
In addition to spatially varying pressure pulses, temporal variations in the pulse arrival times were desired since an idealized spherical blast load detonated at the panel center would arrive at the center first, followed a short time later with loading towards the panel edges and corners. The ideal pressure pulse arrival time map with 25.4 mm spatial discretization is shown in Figure 3.22 based on the ATBlast software.

Achieving this time delay mechanically given a single projectile velocity was accomplished by offsetting the height of the blocks in a concentric manner similar to the technique used to vary the impulse. The center block F was spaced +3.18 mm, the next ring (D/E) were set at +1.59 mm and the rest (A and B/C) were set at 0 mm. Assuming a
projectile velocity of 25 m/s, this would theoretically create a 127 µs delay between the first contact of the center block F and the arrival of the last blocks A-C (see Table 3.5). The order of magnitude difference in time delay for the non-explosive projectile impact compared to the ideal blast was necessary due the order of magnitude difference in impact duration between the two test methods.

<table>
<thead>
<tr>
<th>Location</th>
<th>Ideal time delay (µs)</th>
<th>Actual offset (mm)</th>
<th>Expected delay, 25 m/s (µs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>-23.33</td>
<td>0</td>
<td>-127</td>
</tr>
<tr>
<td>B</td>
<td>-11.39</td>
<td>0</td>
<td>-127</td>
</tr>
<tr>
<td>C</td>
<td>-6.67</td>
<td>0</td>
<td>-127</td>
</tr>
<tr>
<td>D</td>
<td>-3.96</td>
<td>1.5875</td>
<td>-63.5</td>
</tr>
<tr>
<td>E</td>
<td>0</td>
<td>1.5875</td>
<td>-63.5</td>
</tr>
<tr>
<td>F</td>
<td>0</td>
<td>3.175</td>
<td>0</td>
</tr>
</tbody>
</table>

The different tile blocks are shown in Figure 3.23 and the assembled NEM2 projectile system is shown in Figure 3.24. Total mass for the projectile was 50.4 kg with the hybrid tile blocks. For the NEM1 all-steel block design, the projectile mass was 60.3 kg.
Figure 3.23. Tile block designs

Figure 3.24. NEM2 assembled tiled projectile
3.4 TRANSMISSION PLATE DESIGN

The transmitted pressures for an armor vehicle transporting personnel is a key metric for assessing blast resistant armor panel designs, especially considering the brain trauma and lethality of high peak overpressures, as explored through animal testing [55]. Low transmitted pressures will result in less shock to the personnel and thereby improve survivability. Compared to the conventional monolithic RHA plates, all of the armor panels tested used a sandwich core construction with various core materials designed to crush and absorb the blast pressure load while attenuating the transmission of pressure waves through the panels. Measuring this transmitted pressure on the large panel tests was achieved indirectly via the transmitted plate accelerations, and since force = mass x acceleration = pressure x area, then for a given area and mass of the transmission plate, the accelerations measured will have direct correlation to the transmitted pressure (see Figure 3.25). Thus, any trends observed to increase or decrease the accelerations can be directly interpreted as trends applicable to the pressure pulse transmission characteristics of each armor panel design. This is due to the fact that acceleration of the transmission plate is directly proportional to the transmitted pressure which acts as a normal traction acting between the panel back-face and the transmission plate, as illustrated in Figure 3.25.
Identical in functionality to the transmission plate used in the GGUN test series, the transmission plate design for large panel testing required a few notable changes. To match the area of the incoming projectile, the transmission plate would need to be physically larger (see Figure 3.26). The increased impulse levels were expected to impart even higher forces to the plate, so the plate was made thicker to increase stiffness and reduce the global bending from the impact event. Three plates were machined from aluminum 6061 with nominal dimensions 406 x 406 x 25.4 mm to minimize downtime in case any plate was damaged in the field beyond repair. The remaining changes to the plate were due primarily to the restraining mechanism.

Figure 3.25. Measuring transmitted acceleration/pressure
In the GGUN test series (and the OBLT tests as well), the backside of the panel fixture was open which permitted ample room for restraining mechanisms. The GGUN trials, for example, used a friction arresting nylon strap to both confine and stop the transmission plate while the OBLT tests did not use any restraints. Unfortunately, in the NEM tests, the test fixture would be mounted to a concrete block such that spacing between the back of the panel and the surface of the block was limited to 267 mm. Approximately 50 mm of the available space would be necessary to permit unrestricted panel motion post impact and another 25 mm was required for the thickness of the plate itself. This left < 200 mm to arrest the 13.7 kg mass with an expected velocity of 25 m/s. The initial idea for the transmission plate restraining mechanism was to use a pair of hard rubber bumpers nearly 102 mm thick mounted to the face of the concrete block to stop the plate (see Figure 3.27). The solid rubber bumpers were replaced with 50.8 mm of polyurethane foam and 50.8 mm rubber in later tests to minimize the transmission plate rebound.
Linear bearings on each side of the transmission plate would allow it to slide along two shafts and served to contain the plate on rebound. Space constraints required machining two indents on the transmission plate in order to bolt the bearings to either side (see Figure 3.26). The bearings would serve double function as the optical “flag” for high speed camera tracking of the transmission plate velocity in the event of accelerometer damage. Unfortunately, the linear bearings did not survive due several factors including: the very high impulses involved, binding on the shafts, and damage by striking the front shaft supports on transmission plate rebound from the rubber bumpers (see Figure 3.28).
A decision was made to continue the NEM1 tests with the actual bearings removed and relying only on the bearing housings to slip around the shafts (but not actually make contact). In this manner, the housings could still serve as both restraining mechanism as well as the optical flag for the high speed camera. Tracking the flag also faced issues during NEM1 tests since the inertial effects resulted in a delay between the transmission plate motion and bearing housing. Also, as the bearing housing attached to the side of the transmission plate, optical tracking was affected by the dynamic plate bending.

![Linear bearing](image1.png)

![Bearings](image2.png)

Figure 3.28. NEM1 impact showing bearings being ejected from housing

The restraint system was redesigned for the NEM2 tests. To minimize the cost, the revised restraining mechanism was required to interface with the existing test fixture.
Noting that the field-replaced housing/shaft design worked well to prevent the transmission plate from causing damage to the accelerometer, cabling, and itself, the revised system likewise used linear rails that did not physically contact the transmission plate. The rails were split in the center to allow for direct high speed camera viewing (NEM1 linear shaft prevented direct viewing), and the bearing housing was eliminated as the rails were thick enough to fit inside the transmission plate side pockets (see Figure 3.29). Three sets of rails were machined from aluminum, which could be replaced if heavily damaged during testing.

As a final modification, the optical tracking flag was moved from the side of the transmission plate to the center (at the plate’s CG). Due to the split-rail design, the flag could still be tracked directly via high speed camera. However, as the center of the transmission plate was reserved for the accelerometer, the optical flag had to be hollow in order to fit around the sensor. A solution was found by using white nylon plastic pipe fittings attached to a short stainless pipe adapter (visible between rails in Figure 3.29). This in turn was attached to a metal flange that was bolted over the accelerometer to protect it from impact damage. The flags were made from plastic so in the event that they were damaged, the fittings could easily be replaced in the field. To minimize delays in testing, thirty such plastic pipe fittings were hand painted with black stripes for high contrast visibility.
In summary, non-explosive testing via gas gun was unable to apply the necessary wide area, high impulse loading to replicate the blast tests. Nonetheless, these early tests served as a foundation for exploring key components of the large panel non-explosive methodology, such as tuned foam pressure pulse shapers, a tiled projectile design, and the use of a transmission plate to record transmitted impulses. These components would later be implemented and often improved during the Blast Simulator tests.

Parts of Chapters 2, 3, and 4 have been accepted for publication by the International Journal of Impact Engineering, 2014, D. Whisler and H. Kim. The dissertation author was the primary investigator and author of this material.
This chapter details the setup and instrumentation used during the large panel Blast Simulator (NEM) and actual blast tests (OBLT). Results are provided in order to first compare sandwich panel performance relative to RHA steel, and then to assess the damage modes and extent of damage for the two large panel test methodologies.

4.1 EQUIVALENT EXPLOSIVE BLAST IMPULSE

The TNT$_E$ of the Blast Simulator was a 1.0 kg charge at 305 mm standoff (same as the GGUN trials) buried under 50.8 mm soil to replicate a typical vehicle close-in blast detonation event. Due to soil confinement increasing the destructive power of a surface-laid charge, a scaling factor of 2.2 was applied to the pressure time history curve to define a total impulse of 10,134 Pa-s. The pressure loading profile was assumed to be a simplified triangular pulse with a peak overpressure of 145 MPa and 140 µs duration.

Momentum calculations during the actual NEM1 Blast Simulator tests showed an average impulse of 8,460 Pa-s was achieved using the all-steel tile block projectile. This assumed that the projectile was momentarily at rest when the panel was at peak displacement during impact. Although this impulse was 17.1% less than the target impulse of 10,134 Pa-s, the damage shape and extent of damage was observed by
Armorworks to be similar in magnitude to an actual explosive detonation test. Thus, the impulse design level for all large panel tests was set to 8,460 Pa-s.

The second set of non-explosive Blast Simulator tests (NEM2) attempted to achieve this level of impulse loading with temporally and spatially varying pressures. The projectile, by nature of using lower density aluminum hybrid tile blocks, was 16.4% lighter and was expected to match the previous impulse level through a proportional increase in velocity. However, the calculated impulse during testing averaged just 7,520 Pa-s. Although this result appeared to provide a different loading scenario, the damage modes and extent of damage was very similar to NEM1 testing at 8,460 Pa-s.

Actual blast tests were performed by Oregon Ballistics Laboratory (OBLT test series). At the time, using an actual buried explosive was not desirable as variations in orienting the charge and level of soil compaction would be a source of inconsistency between tests. To achieve the most repeatable and consistent spherical shockwave, it was desired to use a ground surface mounted C4 charge, which was shaped into a cylinder. Using the ATBlast software, Armorworks recalculated the required charge weight of TNT at 305 mm standoff with no soil covering that would be equivalent to the impulse of a buried explosive using an amplification factor. The TNT charge weight was determined to be 1.74 kg, and the \( T:\text{E}_C4 \) weight was 1.37 kg at the same 305 mm standoff distance. This would provide the target 8,460 Pa-s (actual value was 8,306 Pa-s) at the panel center. It should be noted that the target area for the actual blast test was not the same as the non-explosive tests. The NEM tests required some spacing between projectile and panel outer fixture boundaries, so the target area was 406 x 406 mm. The
actual explosive blast was not limited to certain areas and could freely apply pressure to
the entire 483 x 483 mm panel surface defined by the window fixture. It also
indiscriminately loaded all nearby surfaces to varying impulse levels. As sensors were
not available on the panel surface to confirm the application of pressure, it would have to
be assumed that the 8,460 Pa-s was applied at the panel center with a spatially and
temporally varying impulse distribution based on position from the center.

4.2 SPECIMEN DESIGN AND NOMENCLATURE

Sandwich panel specimen designs were derivatives of the best performing GGUN
test panels and constructed with the same geometry. The panels were nominally 610 x
610 mm wide, with thicknesses ranging from 33.1 – 40.4 mm for the sandwich panels
and 6.35 mm for the solid RHA steel panel. Twenty holes were waterjet cut around the
panel periphery prior to delivery (see Figure 4.1). The 12.7 mm diameter holes were
used to clamp the panel specimens to the test fixture.

Each sandwich panel set was assigned a four digit numeric identification, e.g.,
1435 was an aluminum face, aluminum foam core specimen. Individual panels within a
group had two more digits appended, the first being the panel revision number and the
second identifying the specific panel. For example, 1435.0.5 referred to a 1435 series
panel, original design (0), and the 5th replicate. RHA panels were numbered sequentially,
e.g., RHA04 referred to the 4th RHA panel. A single 9.53 mm thick aluminum alloy 6061
monolithic panel (ALUM) was also tested for comparison with finite element analysis.
Details of the large panel constructions are provided in Table 4.1 and Figure 4.2. The notation used in Table 4.1 to designate each layer (up to 6 total) is as follows: front face is solid aluminum (AS), T-flex dry aramid fiber (TF), or RHA steel (RH), and for each sandwich panel layer, (SP) is a polymer shock layer, (F) is aluminum foam, and (H) is stainless steel square honeycomb. The number following the F and H represent the core density, where F3 = 300 kg/m³, F4 = 400 kg/m³, F5 = 500 kg/m³ aluminum foam core and H6 = 6.35 mm, H9 = 9.53 mm, and H12 = 12.7 mm cell size for the honeycomb core specimens. The remaining abbreviations are (SF) for syntactic foam, (NN) for Nanovate core, and (NC) for nano-crystalline core. The latter three panel configurations were included for testing but were not fabricated by Armorworks. Areal densities for the
sandwich panels averaged 23.4 to 46.6 kg/m² while the 6.35 mm thick RHA plate was 48.8 kg/m² (see Table 4.2).

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<tr>
<th>Panel ID</th>
<th>Front Face Mat</th>
<th>Layer 2 Mat</th>
<th>Layer 3 Mat</th>
<th>Layer 4 Mat</th>
<th>Layer 5 Mat</th>
<th>Layer 6 Mat</th>
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<td>TF 4.00</td>
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<td>H6 22.4</td>
<td>TF 5.40</td>
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<td>NN 25.4</td>
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<td>SF 25.4</td>
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<td>H9 9.30</td>
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<td>TF 7.20</td>
<td>NC 8.00</td>
<td>TF 4.00</td>
<td></td>
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</tbody>
</table>

ALUM AS 9.53
RHA RH 6.35

* Front face and sandwich panel layers are abbreviated as: solid aluminum (AS), T-flex dry aramid fiber (TF), polymer shock layer (SP), RHA (RH), aluminum foam (F), and stainless steel, square honeycomb (H). The number following the F and H represent the core density, where F3 = 300 kg/m³, F4 = 400 kg/m³, F5 = 500 kg/m³ aluminum foam core and H6 = 6.35 mm, H9 = 9.53 mm, and H12 = 12.7 mm cell size. The remaining abbreviations are (SF) for syntactic foam, (NN) for Nanovate core, and (NC) for nano-crystalline core.
Not every panel series was represented at each stage of testing. A full test matrix for the dynamic large panel tests and coupon specimens is shown in Table 4.2. The first non-explosive Blast Simulator tests were conducted on the 1400 series panels and RHA. Only 1436, 1437, and RHA from NEM1 along with the 1500 series panels were tested using the revised projectile during the second NEM2 Blast Simulator trials. The 1435 – 1438 panels and RHA were tested via actual blast tests OBLT. In total, five panels (1435 – 1438 and RHA) were subject to both NEM and OBLT tests and of those, only three (1436, 1437, and RHA) were tested with NEM1, NEM2, and OBLT systems.
Table 4.2. Specimen test matrix

<table>
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<th>Panel ID</th>
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<th>Coupon</th>
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<td>3</td>
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<td>27.09</td>
<td>3</td>
<td>3</td>
</tr>
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<td>1438</td>
<td>23.36</td>
<td>3</td>
<td>-</td>
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<td>1469</td>
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<td>RHA</td>
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<td>3</td>
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</tbody>
</table>

TOTAL TESTS COMPLETED | 18 | 27 | 5 | 57 | 146 |

4.3 Blast Simulator Test Setup (NEM 1 & 2 Test Series)

The Blast Simulator tests relied on high speed, servo-hydraulic actuators known as blast generators (BG) to provide the impulsive loads. Operation of the BG25 (and the larger BG50) actuator depends on three main components: the accumulator, the piston rod, and the deceleration chamber (see Figure 4.3). The accumulator is an external tank where pressurized nitrogen up to 34.5 MPa is stored along with hydraulic oil. The accumulator is connected to the acceleration chamber located just behind the piston rod. The deceleration chamber is the forward volume of the piston rod, located inside the
BG25. It is charged with nitrogen at up to 20.7 MPa. With both accelerator and deceleration chamber charged to a preset pressure, the system is ready to fire.

Once the command is given, a poppet valve on the accumulator is opened via software control and the high pressure hydraulic oil rushes into the acceleration chamber of the piston inside the BG25. This accelerates the piston rod and attached projectile package to the desired velocity. At the same time, the nitrogen inside the forward deceleration chamber is compressed and pushes against the accelerating piston, although initially, it is unable to overcome the pressure supplied by the hydraulic fluid. The BG25 piston rod is able to accelerate up to 35 m/s, which is dependent on stroke distance, mass, and pressure. After a preset elapsed time, the accumulator supply poppet closes and stops the flow of hydraulic fluid into the piston. At the same time, another valve leading to an empty return tank opens which diverts the hydraulic fluid away from the acceleration
chamber of the piston and reduces the pressure applied to the back of the rod. This allows the gas inside the deceleration chamber, which has been compressing during the entire loading event, to push back on the front side of the piston rod. Without the hydraulic fluid pressurizing the back of the piston rod, the nitrogen inside the deceleration chamber slows the piston rod to a complete stop and forces it to reverse direction. This entire operation requires only a few milliseconds to complete.

The accumulator/decelerator pressures, the valve timings, the mass of the projectile system, and the BG unit(s) employed (either BG25 or BG50, single or multiple units) determine the maximum velocity and minimum stroke distance to reliably and repeatedly impact the target. Appendix A provides the actual parameters necessary to calibrate the BG25 actuator for the NEM1 and NEM2 tests.

For NEM testing, the BG25 was installed in the Blast Simulator and a 9,000 kg reaction block was installed opposite to the actuator to support the panel (see Figures 4.4 and 4.5). To keep operational costs low, a pre-existing reaction block was repurposed that was significantly larger in both size and weight for the anticipated impulse levels. This placed many of the design constraints on the packaging of the reaction components and also affected the maximum velocity of the system. The 60.3 kg projectile would need to be accelerated to 27.8 m/s in order to provide the desired impulse (at the time) of 10,134 Pa-s. The velocity was theoretically possible with the BG25, albeit at its upper limit given the projectile mass and stroke distance (although this was only known after NEM1 tests were well underway). While switching to a larger BG50 could have provided the necessary impulse levels, the BG25 was preferred for its faster response
(due to a lighter piston rod with less inertia) and fine positioning attachments (it slid on a vertical track). It was also much more compact than the larger BG50, which was ideal given the size of the reaction block. Thus, while certain compromises were initially made when designing the system around the BG25 and 9,000 kg reaction block, these components were used for the full set of NEM1 tests. The components were then carried over for the NEM2 tests as they had been calibrated to provide the anticipated 8,460 Pa-s.

Setting up the Blast Simulator first required preparing the test area. Existing fixtures had to be removed and organized and the concrete floor cleaned and covered with a plastic sheet. The 9,000 kg block was trucked in and crane hoisted in place with fine positioning (within 2 mm of the centerline) accomplished via 180 kN hydraulic jacks. The block was bolted to the strong floor and tensioned to 450 kN using four 32 mm, 3.6 m long (minimum) length steel tensioning bars that were dropped in by hand from the top of the 1.60 m tall block. Hydro-Stone gypsum cement was poured into the base of the reaction block (between the block and the floor) and the back face (between the block and the reaction wall) to ensure uninterrupted contact between the two surfaces.
On the impact face of the reaction block, a 12.7 mm thick aluminum plate was installed to provide a smooth surface for mounting the test fixture components (see Figure 4.6). The test fixture top and bottom supports were attached using two 38.1 mm
diameter threaded shafts with matching nuts that co-bolted the aluminum plate to the concrete block. Two side supports resisted the compressive load from the impacts, but did not physically bolt to the concrete block. A welded tubular frame was bolted to the plane defined by the four supports. The rear window fixture and transmission plate guides were then bolted to the tubular frame. The panels were fully clamped between two window fixtures sized 610 x 610 x 12.7 mm with 483 x 483 mm square openings. The window fixtures were originally made from A36 mild steel and were later switched to high strength, 4130 chromoly units prior to NEM2 and OBLT testing.

Figure 4.6. Test fixture attached to the reaction block showing the aluminum plate, top and side support, tubular frame, rear window fixture, and transmission mass
On the loading side, the cantilevered BG25 was maneuvered in place by releasing the tension in the base bolts holding the actuator to the slotted tracks. Using two 180 kN hydraulic jacks, the height of the actuator was positioned before re-tightening the bolts. This setup process was only required once per test session (2x total for both NEM trials). With the actuator in place, the loading components could be installed. A pusher plate assembly was bolted to the end of the actuator (see Figures 4.4 and 4.5). The pusher system was designed as a sandwich structure incorporating 12.7 mm thick aluminum front and back faces with 101.6 mm wide, 3.18 mm wall thickness aluminum rectangle tube core to provide a low weight and high stiffness launch surface for the projectile (see Figure 4.7). The reason for the pusher “plate” to be so rigid was that the 114 mm diameter actuator collar could not provide the rigidity to all points on the relatively flexible 406 x 406 mm projectile during acceleration. The low weight was desired to allow the actuator to reach the maximum possible acceleration given the constraints on the stroke distance. Steel guide rails were maneuvered into position on either side of the pusher plate and bolted in place (see “rails” in Figures 4.4 and 4.5). The entire pusher assembly interfaced with the two steel rails using phenolic sliding bearings. During acceleration, the phenolic bearings would skim along the steel rails to minimize side-to-side and twisting motion. The guide rails required periodic adjustments throughout the test day as the pusher plate was often hit with the rebounding projectile, which resulted in the misalignment. A third rail containing a magnetic sensor was installed below one of the guide rails in order to monitor the position of the pusher plate (see Figure 4.7).
A notable design feature of the pusher plate stems from the projectile requiring the use of bolts to attach the metal tile blocks in the 5 x 5 array. On the projectile, the button head cap screws did not allow the front face of the pusher system to fully engage the back of the projectile. This was undesirable as the thin aluminum backing sheet would be unable to support the tile blocks during acceleration, thereby negatively affecting the launch and subsequent free flight of the tiled projectile. Complicating the situation was the pusher system, which was devised to be modular and required bolts to hold the 12.7 mm thick aluminum faces to the core. Drilling the front face plate with a sufficient quantity of holes to assemble the pusher plate system while also clearing the projectile bolt heads would significantly weaken the structure. Welding the components would have potentially introduced warping to the parallel and planar surfaces and would make changing parts during testing difficult. A solution was found by strategically placing the pusher plate internal supports at the same location as the projectile bolt heads which had the added benefit of positioning the most reinforcements directly behind the tile blocks (all tile block tapped holes with the exception of the center tile were at the center). Then each bolt head location on the pusher plate front face was counter-bored to a depth of 7.62 mm, which was 4.27 mm deeper than required for the button head cap screws. The front face plate of the pusher system would be bolted through those holes to the core, and the same counter-bores were deep enough to serve as pockets for the bolt heads on the back of the projectile. This permitted the pusher system to fully engage the aluminum sheet and tile blocks of the projectile array (see Figure 4.7).
Testing typically commenced early in the morning to maximize the number of runs. First, the pressures were set on the nitrogen fill tank to 17.2 MPa and deceleration chamber to 10.3 MPa (see Figure 4.8). The shock accelerometer and flag were installed on the transmission plate, new foam padding was attached to the arresting system, and the transmission plate installed inside the test fixture. The test panel and window fixtures were bolted in place and a small amount of vacuum grease was applied to the transmission plate center to keep it in contact with the back of the panel surface while the weight was supported by two low friction guide blocks. On the loading side, a projectile package was lifted onto the locating shaft of the pusher plate and the two components were clamped together. The BG25 system checks could now commence as the first early
morning shot required approximately 30 min for the hydraulic oil to reach operating temperature. During this time, the high speed cameras were setup approximately three meters away from the target panel. One Vision Research V7.1 high speed camera was aimed to capture the impact on the front of the panel and set at a resolution of 800 x 600, wide view, 5,000 fps, exposure 100 µs, and f/5.6 using a single 250 W halogen lamp. A second high speed camera was aimed perpendicular to the transmission plate flag, which was much more difficult to see inside the test fixture (especially for the NEM2 revised transmission plate split rails). This camera required three focused 250 W halogen lamps (see Figure 4.9) and used a resolution of 800 x 104, zoomed in to view the flag only and capturing at 20,000 fps, exposure 16 µs or lower, and f/5.6.

Figure 4.8. Blast Simulator general setup procedure
After a several system checks including a common trigger to the data acquisition system (High-Techniques 14 bits, 1 MHz, up to 32 simultaneous channels), the system was ready to fire. The clamps holding the projectile to the pusher plate were removed. It should be noted that there was no locking mechanism keeping the projectile in place during the warm up stage (hence the need for the clamps). The entire 50.4 or 60.3 kg package rested solely on 25.4 mm of the shaft protruding from the pusher plate. Since the aluminum backing sheet was fairly thin, the entire projectile could easily bend and fall off the pusher plate, especially with the low frequency cyclic motion used during the BG warm up stage. Thus, after the clamps were removed, the projectile was tied to the two guide rail towers with a monofilament line. The system was ready for testing.

With the area cleared, the BG25 was activated and the panel impacted (see Figure 4.10). All data were saved and the hydraulic system shut down for safety. Once the site
was safe to approach, the old projectile was hauled away for processing and the tested panel removed. A new panel was installed, the transmission plate reset, and a new projectile lifted into place. With one technician dedicated to the Blast Generator controls, a single test cycle could be completed in as little as 20 minutes—5 minutes to change out panels, install projectile, and reset the DAQ, and 15 minutes for BG warm up (once the first test of the day was completed). Thus, the entire process facilitated rapid, safe testing with several impacts per day.

![Figure 4.10. High speed video capture of NEM2 test on panel 1436.1.4](image)

\[ t = 37,775 \text{ µs} \quad t = 44,575 \text{ µs} \]

\[ t = 48,175 \text{ µs} \quad t = 51,175 \text{ µs} \]
To assist with the demands of rapid testing, three complete tile arrays were machined for both NEM tests. This would allow test setup with one projectile while another was assembled and ready to load, and a third was being processed from an earlier test. Processing of the blocks included clearing off each block, mounting new foam pulse shapers to each, and then mounting blocks to the aluminum sheet. For the first projectile design, total of 76 tile blocks were machined: 72 identical steel tile blocks and 4 specially bored center blocks (3 necessary plus 1 spare). For the hybrid projectile design, a total of 144 separate steel and aluminum pieces were machined: 12 corner block sets (A), 36 outer ring blocks (B/C), and 24 inner ring block sets (D/E). The pieces were then joined together using flat head bolts to sit sub-flush on the impact side each block. Spacers were sandwiched between the two metal parts for the D/E tiles and applied to the back for the center block. The four center blocks were reused from the NEM1 tests and all blocks were color coded for easy identification on test day (see Figures 4.11 and 4.12).

The 25 foam pulse shapers were attached with spray adhesive prior to testing. The first Blast Simulator tests applied the adhesive directly onto the projectile faces, which proved difficult to remove after each test (required scraping). The second Blast Simulator test used a layer of 76.2 mm wide low-tack masking tape applied first to the projectile faces and then the foam shapers were adhered to the tape. After impact, the foam remnants were simply removed along with the tape without requiring scraping. This greatly reduced the post impact processing time and facilitated faster change outs.
Figure 4.11. NEM2 color-coded tile blocks prior to attaching foam pulse shapers

Figure 4.12. NEM2 assembled tile blocks awaiting testing
4.4 Actual Blast Testing (OBLT Test Series)

All explosive blast tests were conducted by Oregon Ballistics Laboratory. As such, detailed methodology and protocol are not available, although one test was witnessed during a trip to Bend, Oregon. The test site was prepared by locating a suitable position for the test fixture. Concrete blocks were positioned over legs welded to the original fixture setup to provide the necessary mass for holding the panel to the ground surface. A shallow hole was dug beneath the test fixture to set the correct standoff distance and a 305 x 305 x 25.4 mm steel plate was placed level into the bottom of the hole. The steel plate served to deflect the blast upward and provide a more controlled and repeatable pressure pulse rather than relying on the variable compaction level of the bare ground surface. Atop the steel plate was placed a 50 mm thick foam pad followed by the cylindrical charge of 1.37 kg C4, which was located 305 mm (12 in) from the armor panel face and centered (see Figure 4.13). The test panel and front window fixture was then installed with 20 bolts as shown in Figure 4.14.

A 50,000 g accelerometer was attached to the transmission plate and laid on top of the panel (see Figure 4.14). A Phantom V7.1 camera was setup near the bunker and pressure transducers placed at 3.05 m intervals up to 9.15 m away from the center of blast. The Phantom and DAQ system was triggered by a trip wire installed near the explosive that would break upon detonation of the C4 charge. Three separate real time video cameras were setup around the area to capture both audio and video, which would later prove essential for calculating the transmission plate’s hang time (duration of time that the plate is in the air).
Prior to detonation, all non-essential personnel relocated to the bunker while the blasting cap was installed in the C4. The charge was detonated and once the area was safe to approach, the damage was assessed—to both panel and the test fixture. For this initial test on panel 1438.1.2, the fixture was cracked in multiple places and the concrete blocks moved, possibly due to the amount of confinement the blocks provided to the blast. The test fixture was repaired and reconfigured as shown in Figure 4.15 for the remaining four panel tests.

Figure 4.13. OBLT C4 explosive 305 mm below target surface
Figure 4.14. OBLT test configuration for panel 1438.1.2
4.5 **TEST RESULTS: TRANSMITTED IMPULSE PERFORMANCE**

One of the first test metrics for comparing panel performance was the transmitted impulse through the panel, which required calculating the momentum of both the projectile and the transmission plate. The applied impulse was calculated from the high speed video camera using reference points such as the tile blocks of the projectile. The average projectile impact velocity for NEM1 tests which used a 60.3 kg steel tiled array mass was 23.0 m/s with a standard deviation of 0.44 m/s. The computed impulse was 1,385 N-s (8,460 Pa-s over a 406 x 406 mm area) with 1.9% deviation for 18 tests (see Figure 4.15. OBLT test configuration for all panel impact tests except panel 1438.1.2)
The average projectile impact velocity for NEM2 tests, which used a 50.4 kg hybrid metal tiled projectile array, was 24.6 m/s with a standard deviation of 0.24 m/s. The calculated impulse was 1,240 N-s (7,520 Pa-s) with 0.98% standard deviation over 27 tests (see Table 4.4). The transmission plate mass was 13.7 kg for all large panel tests.

Table 4.3. NEM1 applied and transmitted impulses

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<th>Specimen ID</th>
<th>Panel AD (kg/m^2)</th>
<th>Velocity Flyer (m/s)</th>
<th>Impulse Flyer (N-s)</th>
<th>Trans. (m/s)</th>
<th>Impulse Trans. (N-s)</th>
<th>(F-T)/F (%)</th>
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<th>Dent depth (mm)</th>
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For the actual blast tests, it was assumed that the C4 detonation applied 1,956 N-s (8,460 Pa-s over 483 x 483 mm), which without confirmation, was the best-known value for all five tests (see Table 4.5). The transmission plate velocity relied on the hang time
of the plate provided by the real-time cameras, which was calculated based on Equation 4.1.

\[ V = \frac{g t}{2} \]  

\( (4.1) \)

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The average impulse absorption for NEM1 tests was 79.1% with a 1.89% standard deviation which was similar to the NEM2 values at 76.3% with 1.36% standard deviation. Sandwich panels averaged 79.6% impulse absorption, which was slightly higher than RHA at 76.8% for NEM1, a trend that was also observed during the NEM2 tests with a 76.4% absorption rate being higher than the 75.5% observed for RHA steel. Actual blast test data showed all panels averaging 78.6% absorption, with the sandwich designs slightly lower than RHA (77.8% vs. 81.5%). However, this may be due to the single anomalous data point in the set, panel 1438.1.2, which showed significantly lower impulse absorption (69.7%) compared to the rest of the panels. As mentioned earlier, the first test of the OBLT series (1438.1.2) was conducted using a different test setup with significantly more obstructions to the lateral gas expansions of the C4 detonation. This
setup was damaged, repaired, and reconfigured for all subsequent tests. It was very likely that the obstructions forced more pressure loads into the front face of the panel, resulting in an unusually high transmission plate velocity and the subsequent low absorption calculation. As no replicate blast data were available, this transmission impulse was assumed valid. Despite this particular panel data, the average impulse absorption across all tests was 77.5% with a 2.49% standard deviation, highlighting the ability of the non-explosive Blast Simulator to generate similar impulses and absorption rates as the actual blast tests. It also demonstrated the ability of the sandwich panels to perform near or better than the baseline RHA from a transmitted impulse point of view. By contrast, the GGUN tests averaged much lower absorption rates of 64.9% with a standard deviation of 6.37%. Combining the near RHA absorption levels with lighter than steel areal densities, the sandwich panels show higher performance per unit weight compared to RHA using the impulse absorption based performance index equation originally defined for the GGUN tests (Equation 3.1).

Areal density notwithstanding, using the absorbed impulse is not a discriminating metric for assessing panel relative performance because the values are relatively unchanged across all panel construction types and test methodologies. This result confirms the belief that impulse tends to be conserved irrespective of the non-linear processes the test panel may undergo during its response to the original threat. The fact that this was not observed at the GGUN level could be attributed to additional losses that, due to the low transmitted velocities, amplified the differences between panels. An
improved metric was defined in place of the absorbed impulse that made direct use of the transmission plate accelerometer data.

4.6 Test Results: Transmitted Pressure/Acceleration Performance

A more discriminating and critical test metric for comparing the performance of the sandwich panels with respect to RHA steel is the transmitted pressures, measured indirectly through the transmission plate accelerometer. The accelerometer time history data for all panels, especially the RHA plate, consisted of high frequency oscillations that generally followed a sharp rise immediately after projectile impact. During this brief time period post impact and prior to the transmission plate reaching a steady-state free flight velocity, the full transmission plate surface was in direct contact with the back of the armor panel. Thus, accelerations should be directly correlated with the transmitted accelerations through the panel. For RHA steel, this time period was approximately 100 µs and approximately 500 µs for the sandwich panels. This time period was confirmed through integration of the accelerometer data and examining the resulting velocity time history for an approximately linear increase (see Figures 4.16 and 4.17).

Two acceleration performance metrics, the initial-average and the maximum accelerations, were defined during the initial impact event. The initial-average acceleration of the transmission plate represented the average transmitted acceleration during the period of direct contact with the deforming panel. It was calculated by averaging the acceleration during the initial panel deformation stage, which was
equivalent to computing the slope of the velocity time history curve when the transmission plate was being accelerated during projectile contact. The maximum acceleration (and maximum transmitted pressure) was simply the highest recorded value during this same period. Both performance metrics were of interest as the initial-average represented the average pressure transmitted during dynamic loading while the maximum pressure represented the highest instantaneous pressure that may be experienced while the armor structure was being loaded. Using Figure 4.16 as an example, the initial-average acceleration for the NEMI test on an aluminum front face, aluminum foam core specimen 1435.0.3 was 4,050 g and the maximum acceleration was 6,900 g. The initial-average acceleration (but not the maximum acceleration) could also be found by using a linear regression on the velocity time history profile as shown in Figure 4.17.
Several NEM1 tests had accelerometer data issues due to poor sensor cable routing. The suspect data sets typically did not reach a steady-state velocity upon integrating the acceleration time histories (see Figures 4.18 and 4.19). Properly measured acceleration data produced velocities that oscillated around a constant value (approximately 20 – 30 m/s) which indicated that the transmission plate experienced non-accelerating free flight after losing contact with the back face of the armor panel. This steady-state velocity was expected and could be confirmed through high speed camera observation. Data sets with the suspect accelerometer cabling, by contrast, often showed a linearly increasing velocity that did not reach a steady-state value. This can be observed by comparing the velocity time history data for 1435.0.3 (Figure 4.17) with the 1435.0.1 panel (Figure 4.19).
Figure 4.18. NEM1 1435.0.1 acceleration time history

Figure 4.19. NEM1 1435.0.1 velocity time history
This experience led to using much improved sensor cable routing in the NEM2 test series that minimized errors in the recorded accelerometer data for most tests. In some instances, the acceleration time histories for the suspect data were not actually lost. As the time period of interest for the initial-average and maximum values was $< 500 \text{ µs}$ post impact, the shock sensor and cabling were often working and not experiencing large displacements affecting the cable. This meant that the sensor and cable were likely to produce valid data and therefore were assumed valid unless noted. Even when the data was anticipated to be in error, typically only the velocity data were affected due to the longer time duration required for steady-state calculations.

Explosive blast testing also resulted in lost acceleration data, but not due to cable routing issues. As the transmission plate was unrestrained, the sensor cables and in some cases, the accelerometers, were damaged. While clearly not desirable, a broken sensor or cable was not an issue in itself since the data region of interest occurred early in the impact event (prior to cable tearing). However, the pressure magnitude of the close-in detonation also saturated the accelerometer time history in almost every tests, and for panel 1437.1.5, the data was lost entirely (see Figure 4.20). Without the full acceleration time history, neither maximum nor initial-average values could be computed for direct comparison with the non-explosive test series. One important observation though was the extreme difference in the rate of acceleration for the RHA panel in comparison to the sandwich panels, which indicated that the RHA panel exhibited a more dynamic impact response. The sole surviving accelerometer time history for sandwich panel 1438.1.2 indicated that the OBLT blast tests resulted in accelerations an order of magnitude higher
than the NEM projectile impact tests. However, as this test used the earlier test fixture setup, its data were not necessarily comparable with the remaining tests which used the modified test setup (see Figures 4.14 and 4.15).

The initial-average and maximum accelerations for the NEM1 tests are shown in Table 4.6. Sandwich panels for both acceleration criteria exhibited much lower values than the RHA steel, by a mean of 74.7% for initial-average and 55.4% for maximum accelerations. Sandwich panels using an aramid front face and single core layer (series 1436 and 1438) showed the highest reductions for both initial-average and maximum accelerations compared to RHA steel. Differences between sandwich panels were not as noticeable for the initial-average accelerations.
Table 4.6. NEM1 initial-average and maximum transmitted accelerations

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* Excluded from average

The initial-average and maximum accelerations for the NEM2 tests are shown in Table 4.7. Examining NEM2 sandwich panels, it was evident that the core material continued to play a significant role in reducing transmitted pressures compared to RHA, with mean initial-average accelerations showing a 31.2% reduction and mean peak accelerations showing a 67.2% reduction, excluding panel 1574. Panel 1574, which used a nano-crystalline core, actually showed a 38.9% higher initial average acceleration.
compared to RHA with a 52.7% reduction in peak accelerations compared to RHA. This could be due in a large part to the strain rate sensitivity of the material itself, but as this particular panel was not designed by Armorworks, further investigation was not undertaken.

The NEM1 initial-average acceleration data showed higher reductions for the sandwich panels with respect to RHA steel compared to the NEM2 reductions for similar panel designs by a factor of approximately two. This was most likely due to the projectile differences between the two tests as the same RHA showed an initial-average acceleration for NEM1 that was significantly larger than the NEM2 trials. General trends for both NEM tests showed lower transmitted accelerations (higher reductions compared to RHA) for the honeycomb core sandwich panels compared to the aluminum foam core panels. This was observed for the NEM1 tests in which the dual and single honeycomb cores experienced a mean of 77.4% initial-average and 49.8% maximum acceleration reductions while the aluminum foam cores achieved a mean of 69.8% and 47.0% reductions compared to RHA.

The NEM2 tests likewise showed all honeycomb core panels with a 33.2% and 72.5% mean initial-average and maximum acceleration reduction, respectively, compared to RHA while the aluminum foam core panels had a mean 29.6% and 63.2% reduction with respect to RHA. As the honeycomb panels also had a lower areal density (25.9 kg/m² average for both NEM1 and NEM2 panels) compared to the foam core specimens (32.8 kg/m² averaged for both NEM tests), it can be inferred that the honeycomb panels offered greater attenuation of transmitted accelerations/pressures compared to the
aluminum foam core designs with less mass and also improved performance in both acceleration and weight compared to RHA steel. Unfortunately, OBLT blast data were not available for comparison to the NEM results, but the onset of a maximum acceleration and the appearance of an approaching maximum acceleration for the sandwich panels indicated the general trend that the sandwich panel structures were greatly reducing the transmitted acceleration compared to RHA steel. As shown in Figure 4.20, the acceleration time history for RHA appears with a steeper slope and no maximum value that suggests a higher transmitted acceleration compared to the sandwich panels.

In summary, the transmitted pressures, measured indirectly through the transmitted accelerations, showed more attenuation for the sandwich panels compared to the RHA steel baseline. Given that impulse absorption as a function of the loading impulse were virtually constant for a given test methodology, the sandwich panels appear to offer significant performance advantages over RHA steel at a lower areal density.
<table>
<thead>
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<th>Panel ID</th>
<th>Acceleration Init.-Avg. (g)</th>
<th>Max (g)</th>
<th>Mean Accel. and Reduction w.r.t. RHA Init.-Avg. (g)</th>
<th>Reduction (%)</th>
<th>Max (g)</th>
<th>Reduction (%)</th>
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</table>

* Excluded from average
4.7 Test Results: Deformation Profiles

In addition to the impulse and acceleration-based performance metrics comparing the sandwich panels with RHA steel, two damage-based assessments were developed for comparing the non-explosive and actual blast test methodologies. The two criteria were the deformation profile and the internal damage state of the armor panels.

The deformation profile provided a quantifiable metric for examining the extent of external damage to the various panels using a custom built 762 mm long pin gauge. Each panel was measured along the horizontal centerline by pressing the pin gauge against the deformed panel shape. Profiles were taken along both the impact (front) face and the back face. The pin gauge profile was traced onto sheets of paper that were scanned at 300 dpi (11.8 dots per mm). The digital images were stitched together manually to minimize optical errors, the major artifacts were removed, and the image processed through a Matlab script [56]. The script was able to extract the deformed shape as a function of position along panel centerline, even filling in small gaps in the traced deformation profile by curve fitting to data on either side of the discontinuities.

Deformation profiles are presented here for panels that were subject to both NEM and OBLT with additional deformation plots provided in Appendix B. Each plot contains front face and back face traces measured along the centerline of the actual panel and each plot is qualitatively assessed based on up to three criteria: repeatability, symmetry, and for cases when both NEM and OBLT data are available, consistency. Repeatability between tests for a given test methodology is observed by examining how well each profile overlays with the other replicate tests. Symmetry of the profiles about an
imaginary centerline drawn at the 0 mm mark measures the ability of the test method to load the panel uniformly. Finally, consistency between tests compares the mechanically induced dynamic impacts of the NEM methods with the actual explosive pressure pulse loads of the OBLT tests.

For example, the NEM1 tested aluminum front face, aluminum foam core panel 1435 is shown in Figure 4.21 with near-perfect overlay front and back profile traces and good symmetry about the centerline. Averaging all three NEM1 deformation profiles and plotting with respect to the 1435.0.4 OBLT test results emphasizes the ability of the NEM projectile to induce similar levels of plastic deformation in the sandwich panel (see Figure 4.22). The OBLT test caused higher front face deformation but in a more localized area, which could be an error with this particular test only, or a consequence of the using the shaped C4 charge in general. The bump at the center of the back face deformation was due to the thin aluminum sheet plastically forming into the hole in the transmission plate used for mounting the shock sensor.

Deformation profiles for the aramid front face, aluminum foam core panel 1436 are shown for both NEM tests in Figure 4.23 and actual blast in Figure 4.24. The non-explosive tests had high repeatability with the exception of two NEM2 deformation profiles. These were later found to be caused by front face separation that was not known from observing the outer damage only. The back face deformation, however, was similar for all tests and may be considered more accurate due to the possibility of front facesheet separation. The OBLT panel had a non-symmetric pressure loading that cause more deformation on one side of the panel front face.
Figure 4.21. NEM1 deformation profiles for panel 1435

Figure 4.22. NEM1 average deformation profile, OBLT deformation profile for panel 1435
Figure 4.23. NEM1 and NEM2 deformation profiles for panel 1436

Figure 4.24. NEM1 and NEM2 average deformation profiles, OBLT deformation profile for panel 1436
The deformations for the honeycomb core panels, 1437 and 1438, were significantly lower than the aluminum foam core panels 1435 and 1436 (see Figures 4.25 to 4.29). This was true for both NEM and blast tested panels. Comparison with the OBLT results confirmed the ability of the NEM tests to achieve blast-level deformation of the armor panels. Due to the irregular back face surface, only the front face deformation profile was captured in Figure 4.26 for the OBLT panel 1437.1.5. The single honeycomb core panel 1438 showed higher deformations compared to the double honeycomb core specimen, with good correlation to the OBLT results (see Figures 4.27 and 4.28). Finally, the RHA impacts showed consistency between NEM test methods in Figure 4.29, with higher than average deformations compared to the OBLT tested RHA panel in Figure 4.30.
Figure 4.25. NEM1 and NEM2 deformation profiles for panel 1437

Figure 4.26. NEM1 and NEM2 average deformation profiles, OBLT deformation profile for panel 1437, 1437.1.5 only back profile due to front panel damage
Figure 4.27. NEM2 deformation profiles for panel 1438

Figure 4.28. NEM2 average deformation profile, OBLT deformation profile for panel 1438
Figure 4.29. NEM1 and NEM2 deformation profiles for RHA panel

Figure 4.30. NEM1 and NEM2 average deformation profiles, OBLT deformation profile for RHA panel
In summary, the non-explosive method was able to induce a similar plastic deformation state on sandwich and conventional RHA steel armor panels relative to the deformation states produced by actual blast tests. The non-explosive methodology was able to consistently and repeatedly load the panels such that their deformation profiles were similar for replicate tests to the degree that they appeared to overlay when plotted as a function of panel position. This was also a testament to the consistency of the panel construction process. The only two instances where the front face deformations for the NEM panels did not match was later discovered to be caused by front face separation from the core and would not have been known at this stage of non-destructive analysis (see Figure 4.23). In general, the NEM-induced deformation profiles were very symmetric about the panel centerline thus indicating that the non-explosive methodology, and more specifically, the tiled array projectile package, was able to load the panel surface with a symmetric pressure distribution profile over a wide area.

4.8 Qualitative Assessment of Internal Damage

Qualitative comparisons between the test methods were also performed by assessing the internal damage state of the armor panels. Several sandwich panels, both non-explosive and blast tested, were sectioned via waterjet along the centerline to qualitatively assess the internal damage of the core which was otherwise not visible from the external surfaces. RHA panels were not sectioned as the internal damage was not anticipated to exist. Cut cross-sections presented here show the damage for the four NEM and OBLT sandwich panels with additional section cuts provided in Appendix B.
The actual blast tested panels exhibited a similar damage mode compared to the non-explosive method, just more extensive. For panel 1435, the C4 detonation also produced such a localized high temperature and pressure environment that the aluminum front face actually melted and flowed outward from the center of the blast (see Figure 4.31). By comparison, the non-explosive 1435 panel test did not have a melted front face, nor the extensive core crush of the OBLT test (see Figures 4.32 and 4.33). Remaining blast tested panels, including RHA steel, showed charring on the front face but did not melt.

Figure 4.31. OBLT panel 1435.0.4 showing melted aluminum front face
Figure 4.32. OBLT panel 1435.04 showing extensive core crushing at center and cracks near the boundary
The extensive internal damage for the blast tested panels compared to the non-explosive tested panels may be explained, in part, by the much faster dynamic application of the close-in pressure pulse for the actual blast tests. When detonated, the high pressure shockwave from the C4 charge moved at a velocity of up to 8,000 m/s and impacted the front surface of the panel first. This accelerated the front surface and caused it to displace, especially at the panel center where the impulse was highest, before the back surface could respond. The difference in acceleration between the two surfaces resulted in a high degree of core crush which was intensified by the transmission plate resting on the back surface. This served to provide an additional inertial reaction force further restricted the back face deformation.

For the NEM test series, the less intense acceleration resulted in less inertial reaction from the transmission plate, thereby resulting in less core crushing (see Figure 4.33.)
4.34). However, as the total impulse level was designed to be equivalent to the actual blast test, the final external deformation profile was similar between the test methodologies. For the monolithic RHA panels, which did not use a sandwich core construction and hence both front and back faces moved in sync at all impact velocities, no visible external damage indicated any differences between the test methods.

Figure 4.34. NEM1 panel 1436.0.2 showing intact core with shear cracks near clamped boundary

As was noted in the previous section, the front face deformation was not consistent for all tests, particularly the NEM2 panel 1436 (see Figure 4.23). This was due
to the composite front face debonding from the core by breaking free at the polymer layer (see red layers in Figure 4.35). Debonding occurred for two of the three NEM2 tests and did not occur at all for the NEM1 tests. The blast tested panel also had front face debonding as shown in Figure 4.36.
The honeycomb core sandwich panels experienced less deformation compared to the aluminum foam core panels. For the non-explosive methods, the damage to the core was minimal at the panel center, with only some shear effects visible near the clamped boundary condition (see Figure 4.37). In most cases, the cell walls did not even show any signs of buckling, which was surprising especially for the 25.4 mm thick single layer 1438 panel (see Figure 4.38). Results were consistent for both NEM tests on panel 1438 (see Figures 4.37 and 4.39).
Figure 4.37. NEM1 panel 1437.0.3 with intact core at panel center and shear effects near clamped boundary

Figure 4.38. NEM1 panel 1438.0.2 with intact core at panel center and shear effects near clamped boundary
By contrast, the actual blast tested panels, 1437.1.5 and 1438.1.2, showed significant core damage, especially at the panel center (see 4.40 and 4.41).
Figure 4.40. OBLT panel 1437.1.5 with extensive core crush throughout
Parts of Chapters 2, 3, and 4 have been accepted for publication by the International Journal of Impact Engineering, 2014, D. Whisler and H. Kim. The dissertation author was the primary investigator and author of this material.
This chapter describes the test setup and results for the coupon specimens subject to quasi-static and high-rate dynamic loading conditions. Force time history plots and force as a function of impact velocity are used to examine the material capabilities in the absence of large panel interactions and boundary conditions.

5.1 MTS Uniaxial Compression Tests (MTSQ & MTSF Test Series)

Coupon specimens were waterjet cut from pristine sandwich panels and tested in uniaxial compression at crosshead speeds of 10 mm/min (quasi-static test series MTSQ) and fast rate 250 mm/s (fast rate test series MTSF). The nominally 51 x 51 mm specimens having total thicknesses of 33.1 – 40.4 mm (depending on facesheet and core) were used to examine basic through thickness stiffness and transmitted force through the material itself. The fast rate tests required the Dytran 1060v5 dynamic load cell to capture dynamic stresses and a 1.59 mm layer of polyurethane foam to pulse shape the initial contact. The MTSQ tests relied on the standard MTS load cell to measure force. Both tests used a clamped fixture on top and bottom to provide a planar and parallel surface for testing (see Figure 5.1). In no instance was eccentric loading experienced, and all specimens were loaded to a predefined displacement or up to 75 kN force, whichever limit was reached first. RHA specimens were not tested via MTS as they did not have a core to crush.
The force vs. displacement plot for the different coupon specimens resembled the three stage stress-strain response of a polymer foam (see Figure 3.8). The aluminum foam specimens, for example, showed an elastic regime, collapsing stress, and densification (see Figures 5.2 and 5.3). This behavior was observed for both quasi-static and dynamic crushing with similar loading profiles despite three orders of magnitude velocity difference. Some inconsistencies in the aluminum foam construction including localized high density regions and the presence of large voids resulted in different force loading curves (see Figure 5.2). This behavior for was not present for all aluminum foam core specimens (e.g., specimen type 1436 was consistent as shown in Figure 5.3) and was not observed during the large panel tests due to the relative size of the defects with
respect to the total panel volume and the low strain levels during the highly controlled NEM testing.

While the MTSQ force data for all specimens had a continuous increase during the elastic loading regime, the dynamic crush tests experienced a brief pause at approximately 5.8 kN (see Figure 5.2). This corresponded to the 2.16 MPa quasi-static collapse stress of the thin foam pulse shaping layer (44.5 x 44.5 mm yields 2.9 MPa) and did not affect the performance of the material as the coupon specimens were still being loaded with a pressure equal to the collapse stress of the foam until densification. This artifact was corrected by offsetting the force vs. displacement curve by the height (1.59 mm) of the polyurethane foam layer (see Figure 5.2, “MTSF 1435 corrected”).

![Figure 5.2. MTSQ and MTSF force vs. displacements for specimen 1435](image-url)
Figure 5.3. MTSQ and MTSF force vs. displacements for specimen 1436

The dual honeycomb core specimens showed an initial maximum contact force corresponding to a honeycomb layer buckling then a second maximum force as the second honeycomb layer buckled. Only after both cores crushed did the specimen undergo densification (see Figure 5.4). The two layers crushed independently of each other, often with the rear layer first. The quasi-static loading profile was also repeatable as these specimens utilized a more consistent construction. The force vs. displacement of the single honeycomb layer specimen was similar to the double honeycomb core, with the exception of having a single peak force initially then densification after the core fully crushed (see Figure 5.5).
Figure 5.4. MTSQ and MTSF force vs. displacements for specimen 1437

Figure 5.5. MTSQ and MTSF force vs. displacements for specimen 1438
The nano-crystalline core specimen showed almost no force response until full core crush (see Figure 5.6). This was attributed to a very low core collapsing stress.

Figure 5.6. MTSQ and MTSF force vs. displacements for specimen 1574

The quasi-static force vs. displacement plots could be used to establish the stress-strain behaviors for finite element modeling. The term strain in this sense referred to the core crush amount and may not necessarily be the actual strain in any of the constituent materials. However, due to the complex interaction of dry aramid fibers in compression and the buckling behavior of honeycomb cores, the only sandwich specimen included in FE simulations was the aluminum face, aluminum foam core type 1435 specimen (see Chapter 6). Additional force vs. displacement plots are included in Appendix C.
5.2 Dynamic Testing Using the Hopkinson Bar (HOPK Test Series)

Dynamic testing of the various panel designs using representative coupon specimens provided a more direct quantitative measurement of the transmitted and attenuated pressure pulses compared to the large panel tests. The captured force time histories demonstrated the potential of the panel material themselves, free from the effects of the large panel’s boundary conditions, to reduce the transmitted pressures compared to the RHA steel baseline. In this manner, the performance of each material could also be assessed and provide insight into the large panel tests, which experienced a wide range of crush behavior, from minimal (NEM tests) to extensive (OBLT tests).

Three nominal impact velocities (38, 70, and 98 m/s) were selected to create a different intensity impulse (1,930, 3,560, and 4,990 Pa-s, respectively) in order produce three levels of core crush and deformations approaching those observed at the large panel level. At each velocity range, it was desired to have at least three repeat data sets for each of the 12 specimens (RHA included) which required a minimum of 96 tests. The actual HOPK velocities and number of tests are summarized in Table 5.1.

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<th>Actual Velocity (m/s)</th>
<th>Std. Dev. (%)</th>
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<td>97.97</td>
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Each specimen was mounted to the Hopkinson bar with a 3.05 mm thick foam pulse shaper, then impacted using an aluminum tipped foam projectile (see Figure 5.7).

Figure 5.7. HOPK aluminum tipped, foam body projectile
The 78.9 mm diameter, 126.8 mm long projectile (excluding 54.0 mm tall stabilizing tail) were individually cast using 80 kg/m³ polyurethane foam. The actual density of the foam was estimated to be almost 160 kg/m³ due the casting process (see Appendix D). A 3.175 mm thick, 76.2 mm diameter aluminum 6061 disk formed the impact surface disk and provided a short duration, high pressure pulse on contact with the specimen. An impact sequence is shown in Figure 5.8 for the single honeycomb core specimen 1438 at 96.72 m/s and details of the core response is shown in Figure 5.9.

The amount of core crush for the coupons generally increased with increasing velocity. In some instances, especially for the aluminum foam core specimens, the crush response was not consistent at a given impulse, i.e., the same velocity impact on the same specimen resulted in two completely different crush levels (compare crush levels for specimen type 1570 at 70 m/s in Figure 5.10). This result was also observed for the MTSQ tests and can be attributed, in part, to the presence of defects or voids in the foam specimens that was not typically observed for the more uniform honeycomb core specimens. The honeycomb core specimens often showed buckling progressing from the rear of the specimen (away from projectile, see Figure 5.9). For the double honeycomb specimens, this meant that the layer closest to the Hopkinson bar crushed first. This can be observed at both 38 m/s with specimen 1571.0.3 and at 70 m/s with specimen 1437.1.1 in Figure 5.10.
Figure 5.8. HOPK dynamic loading of specimen 1438 at 96.72 m/s and captured at 10,000 fps; time is measured from first contact with pulse shaper.
Figure 5.9. HOPK core crush sequence of specimen 1438 at 96.72 m/s and captured at 68,000 fps; time is measured from first contact with pulse shaper.
Figure 5.10. HOPK typical specimen crush levels increasing with increasing velocity; impact applied to the top surface with bottom surface contacting Hopkinson bar

5.3 Test Results at 38 m/s

As Figure 5.10 shows, impacts at 38 m/s did not necessarily damage the sandwich specimens, especially for the higher relative density cores. The aluminum foam core
specimens, specifically, 1436, 1568, and 1569 were similar in construction used a progressively decreasing core density, i.e., 1436 used a 500 kg/m³ aluminum foam core, 1589 used a 400 kg/m³ core, and 1569 used a 300 kg/m³ core (see Figure 5.11). Plotting the transmitted force time histories in Figure 5.12 shows that core density was generally proportional to the maximum contact force at impact. This was not always observed due to the varied core construction, so it was possible that a higher density core specimen like 1436 could also behave similar to the lower density core specimens 1568 and 1569. The lack of a second densification stage (i.e., a second force peak) indicated that the sandwich coupons were not tested to their full capacity.

Comparing specimens 1435 and 1436, with the only difference being the front layer material (aluminum for 1435 and aramid for 1436) showed a similar force response (see Figures 5.13 and 5.14). This suggested that the transmitted forces at the coupon level might not necessarily be influenced by the front face material, but rather, the core itself. The two aramid front face specimens showing both 10 and 22 kN collapsing force during the initial impact behavior may be due to varied foam core porosity resulting in different crush levels.

Finally, plotting 1568 and 1570 together showed that the presence of the polymer layer increased the duration and decreased the magnitude of the transmitted forces through the sandwich specimens (the two specimens are identical otherwise, see Figures 5.15 and 5.16). All aluminum foam core specimens had a lower initial force response compared to RHA steel at 38 m/s (see Figure 5.16).
Figure 5.11. Aramid front face coupon specimens with varying core densities

Figure 5.12. HOPK force time histories for aluminum foam core specimens 1436, 1568, and 1569 at 38 m/s
Figure 5.13. Aluminum and aramid front face coupon specimens with identical core densities

Figure 5.14. HOPK force time histories for aluminum foam core specimens 1435 and 1436 at 38 m/s
Examining the honeycomb core specimens, 1437, 1571 and 1572, showed that the transmitted forces increased with density (and decreasing cell size, see Figures 5.17 and 5.18). This trend was also observed for the aluminum foam core specimens, but the
consistency in the force time history plot for the honeycomb impact tests at 38 m/s indicated less variability in the core construction.

Figure 5.17. Double honeycomb core specimens of varying cell size

Figure 5.18. HOPK force time histories for honeycomb core specimens 1437, 1571, and 1572 at 38 m/s
The lack of the polymer layer in 1573 compared to 1571, all other materials identical, did not result in a different transmitted force pulse previously observed for the aluminum foam core specimens (see Figures 5.19 and 5.20). This may be due to the construction method. For the aluminum foam core specimens, the polymer layer was bonded to a flat layer of aluminum that contained the core itself. By comparison, the honeycomb specimens were bonded to the polymer layer directly at the edges of the cells. This minimized the contact area for the honeycomb specimens and therefore, did not appreciably influence the force transmission.

Differences between single core and double core honeycomb layers also did not seem to significantly alter the transmitted force response through sandwich specimens 1437 and 1438, both of which used a stainless honeycomb core with a 6.35 mm cell spacing (see Figures 5.21 and 5.22). This was most likely due to same cell size and the minimal core crushing to cause buckling. The previous uniaxial compression studies (see Figures 5.4 and 5.5) indicated different force curves occurred for core crush levels > 5 mm (approximately 22.3 – 26.9 % core crush).

The transmitted force response for the nano-crystalline core specimen 1574 was originally believed to have matched the RHA response (see Figure 5.23). However, further investigation revealed that the core offered almost no resistance to the projectile, so the initial force response was much lower than measured. The recorded peak force was actually the onset of densification (see Figure 5.24). As the design of specimen 1574 was similar to the double layer honeycomb specimen 1573 (with the exception of the core itself), it was apparent that the core material played a significant role in attenuating the
transmitted pressure pulse. This result was also noticed when comparing the aluminum and aramid facesheet specimens 1435 and 1436 with identical aluminum foam core densities.

Figure 5.19. Double honeycomb core specimens with and without polymer layer

Figure 5.20. HOPK force time histories for honeycomb core specimens 1571 and 1573 at 38 m/s
Figure 5.21. Single and double honeycomb core specimens with identical cell size

Figure 5.22. HOPK force time histories for honeycomb core specimens 1437 and 1438 at 38 m/s
5.4 Test Results at 70 m/s

At the 70 m/s projectile impact velocities, the aluminum foam cores experienced more crushing but typically still had some energy absorption capacity before
densification. As with the 38 m/s data, the initial response of the aluminum foam core specimens depended on the foam densities, and the denser 1436 specimen transmitted higher initial forces than the less dense 1569 specimen (see Figure 5.25). However, both 1568 and 1569 specimens started showing increasing force during the latter stage of loading, indicating the onset of densification. The aluminum front face specimen 1435 on the contrary, appeared to be well within the material limits for energy absorption since it did not appear to be at the locking strain (see Figure 5.26).

![HOPK force time histories for aluminum foam core specimens 1436, 1568, and 1569 at 70 m/s](Figure 5.25. HOPK force time histories for aluminum foam core specimens 1436, 1568, and 1569 at 70 m/s)
Figure 5.26. HOPK force time histories for aluminum foam core specimens 1435 and 1436 at 70 m/s

At this stage, the differences between the sandwich panels and the monolithic RHA were apparent. The initial force response for RHA was nearly three times larger than the sandwich panels. This indicated limited pressure pulse attenuation through the RHA panel. In addition, the lack of a polymer layer for the aluminum foam core specimen type 1570 was becoming evident when compared to the otherwise identically constructed specimen 1568 (see Figure 5.27). This difference was evident by the densification force in specimen 1570 which indicated a completely crushed core. In contrast, all specimen type 1568 remained below the densification stage.
The behavior of the honeycomb specimens subject to a 70 m/s projectile impact was similar to the behavior at 38 m/s. The lower core densities resulted in lower initial contact force but resulted in more core collapsing and subsequently, the onset of densification and a higher secondary peak (see Figure 5.28). The lack of a solid surface for the polymer layer to spread the dynamic loads caused both polymer and non-polymer constructed sandwich panels to have the same response (see Figure 5.29). As core crushing increased, the responses of single and double honeycomb layers were diverging (see Figure 5.30) such that the double honeycomb core specimen 1437 was experiencing a second peak force whereas the single layer core in specimen 1437 was still collapsing.

Forces developed upon full densification for the nano-crystalline core specimen was even surpassing the initial RHA response (see Figure 5.31).
Figure 5.28. HOPK force time histories for honeycomb core specimens 1437, 1571, and 1572 at 70 m/s

Figure 5.29. HOPK force time histories for honeycomb core specimens 1571 and 1573 at 70 m/s
Figure 5.30. HOPK force time histories for honeycomb core specimens 1437 and 1438 at 70 m/s

Figure 5.31. HOPK force time histories for nano-crystalline core specimens 1574 and RHA at 70 m/s
5.5 Test Results at 98 m/s

At 98 m/s, nearly all sandwich specimens reached densification, i.e., had a second transmitted force that exceeded the initial contact force. This was observed for the aluminum foam core specimens, but only for the less dense 300 – 400 kg/m$^3$ cores (see Figures 5.32 and 5.33). The aluminum faced, aluminum foam core specimen continued to show almost no change in transmitted force despite the increase in velocity (see Figure 5.33). This was most likely due to the stiff, heavy front layer that required considerable momentum to put into motion which was then dissipated through a uniform crushing of the core. The inclusion of the polymer layer reduced both initial contact force and maximum densification force while also extending the pulse duration for the aluminum foam core specimen (see Figure 5.34).

The maximum densification value was inversely related to the initial contact force (and core density) for the honeycomb specimens, while the polymer layer did not significantly reduce transmitted forces (see Figures 5.35 and 5.36). Subtle differences appeared to separate the initial response of the single and double layer honeycomb core specimen, but the second peak for the double layer 1437 indicated a fully crushed second core, but not necessarily densification of the complete specimen (see Figure 5.37). Densification was not reached for the single layer honeycomb specimen.

The nano-crystalline core specimen continued to achieve a high densification force that made differentiation with the initial contact force almost imperceptible (see Figure 5.38). Since the collapsing stress was too low to be reliably measured, the densified force was used for discussing its dynamic load response at 98 m/s.
Figure 5.32. HOPK force time histories for aluminum foam core specimens 1436, 1568, and 1569 at 98 m/s

Figure 5.33. HOPK force time histories for aluminum foam core specimens 1435 and 1436 at 98 m/s
Figure 5.34. HOPK force time histories for aluminum foam core specimens 1568, 1570, and RHA (several plots excluded for clarity) at 98 m/s

Figure 5.35. HOPK force time histories for honeycomb core specimens 1437, 1571, and 1572 at 98 m/s
Figure 5.36. HOPK force time histories for honeycomb core specimens 1571 and 1573 at 98 m/s

Figure 5.37. HOPK force time histories for honeycomb core specimens 1437 and 1438 at 98 m/s
The maximum initial contact force (MTSQ, MTSF and HOPK) and the dynamic densification force (HOPK) for all quasi-static and Hopkinson bar impact tests are plotted as a function of velocity in Figure 5.39. For a given specimen, the large markers are used to show the initial force and the smaller markers are used to show the densification force. For example, specimen type 1570 uses a square marker, which shows similar initial and densification force at 38 m/s, but the two forces diverge as the increase in velocity results in a large increase in the densification force relative to the initial peak force. Specimens 1574 and RHA use only the large markers to indicate the maximum force (densification for type 1574 and initial for RHA) achieved during loading.
Figure 5.39. HOPK and MTSQ/MTSF average maximum force and HOPK average densification force for all coupon specimens.
In general, the specimens with the lowest initial contact force (e.g., 1568, 1569, and 1572) also possessed the lowest density cores among the sandwich specimens. As such, when the impact velocity increased, their cores did not have the energy absorption capacity of the higher density specimens. This resulted in larger densification forces. Specimens such as 1435 and 1438 experienced low initial contact forces with little or no densification force increase due to the capacity of their cores to absorb the energy of the impact event. Since transmitted force is directly proportional to transmitted pressures, then these specimens, along with 1436 and 1437 exhibited some of the best performance for attenuating pressures at initial impact and densification. However, this does not necessarily imply that panels utilizing these materials on a large scale will have the best performance since the boundary conditions and other interactions may contribute to the global response and performance of the armor panel.

As velocity increased from 38 to 98 m/s, the difference in the initial transmitted forces between the sandwich panels and the RHA baseline also increased. As can be seen in Figure 5.39, with the exception of the nano-crystalline core specimens 1574, the RHA steel formed the upper limit of transmitted forces (i.e., would transmit the highest pressures) across the range of velocities investigated. This was a direct result of pressure pulse attenuation through the core of the sandwich specimens. RHA, which did not have any core to crush or any voids/interfaces to attenuate the propagating stress wave, transmitted a high magnitude and short duration (100 – 200 µs) force pulse.

Computing the impulse by integrating the force time histories indicated that all specimens transmitted approximately the same impulse, but the sandwich specimens with
their lower force time histories extended the pulse over a longer duration and thus, reduced the maximum transmitted force (see Figure 5.40). The benefits of this force response to armor panels is that a sudden impact event can be mitigated by transmitting a lower and less impulsive force when using a sandwich panel design compared to the traditional RHA steel. In many respects, this was similar to the damage observed when comparing the NEM panel to the OBLT panels, i.e., the initial impulse event may have been the same, but the damage was much more extensive when the transmitted forces were more dynamic in nature.

Figure 5.40. HOPK impulse for all coupon specimens at 38 – 98 m/s impact velocity
6 Finite Element Analysis

This chapter details the finite element modeling progression from material validation studies of the elastic-plastic polyurethane foam to the large panel dynamic impact tests using RHA, aluminum, and aluminum foam core sandwich specimen 1435. The quarter symmetric large panel FE models are then used to assess the ability of the non-explosive projectile impact to replicate the impulsive loading of an idealized close-in blast detonation event.

6.1 Material Definitions

The majority of components used in actual testing were constructed from steel and aluminum alloys that did not undergo plastic deformation (e.g., NEM and OBLT test steel test fixtures). Modeling these linear elastic materials in Abaqus/CAE v10-EF1 required only three inputs for density, modulus, and Poisson’s ratio [53]. Materials that were expected to experience plastic deformations, required at minimum, three additional inputs to indicate the stress at the start of yielding and the stress-strain values at a later point in the plastic regime (see Table 6.1). RHA, for example, used elastic material properties and yield stress based on [57 – 59], while the 1.3 GPa stress at 0.65 strain was selected in order to provide a slight rising plastic portion in an idealized elastic-perfectly plastic material response (see Figure 6.1).
Table 6.1. FE elastic-plastic material properties

<table>
<thead>
<tr>
<th>Material</th>
<th>Elastic</th>
<th>Plastic</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Density (kg/m³)</td>
<td>Modulus (GPa)</td>
</tr>
<tr>
<td>Al 5083</td>
<td>2,660</td>
<td>71</td>
</tr>
<tr>
<td>Al 6061</td>
<td>2,827</td>
<td>70</td>
</tr>
<tr>
<td>RHA</td>
<td>7,850</td>
<td>200</td>
</tr>
<tr>
<td>Steel</td>
<td>7,850</td>
<td>200</td>
</tr>
<tr>
<td>Cast Plastic</td>
<td>1,048</td>
<td>0.31</td>
</tr>
<tr>
<td>Polymer*</td>
<td>300</td>
<td>0.027</td>
</tr>
</tbody>
</table>

*Polymer layer plastic properties were tuned to provide the most realistic loading response and are not necessarily representative of an actual material.

Figure 6.1. FE stress-strain curves for elastic and elastic-plastic metals

With the exception of the polymer, all materials in Table 6.1 were based on actual engineering values for the three elastic materials properties and yield stress. The polymer
layer that was used as a shock mitigator in many of the sandwich armor panels was simulated with an elastic-plastic material. The properties were tuned to provide the most realistic loading response at impact velocities up to 98 m/s and were not necessarily representative of an actual material (see Figures 6.2 and 6.3).

![Figure 6.2. Specimen 1435 showing polymer layer](image)

![Figure 6.3. FE stress-strain curve for elastic and elastic-plastic non-metals](image)
Replicating the high strain, nearly constant stress regime for the polyurethane foam pulse shapers was accomplished using the Abaqus crushable foam with volumetric hardening material model. This model requires uniaxial compression, tension, and hydrostatic stress data in order to compute the two parameters \( k, k_t \) that govern the evolution of the foam yield surface [53]. The parameter \( k \) is the ratio of uniaxial compression yield stress to hydrostatic compression yield stress and \( k_t \) is the ratio of hydrostatic tension yield stress to hydrostatic compression yield stress. Material properties obtained were obtained from General Plastics [41], the Abaqus/CAE user manual [53], and studies by Maji et al. [52] in order to compute \( k = 1.16 \) and \( k_t = 0.83 \) (see Table 6.2, parameters assume material properties in loading direction parallel to cell rise direction).

Table 6.2. Polyurethane foam material properties

<table>
<thead>
<tr>
<th>Property</th>
<th>(MPa)</th>
<th>Load to cell rise direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydrostatic yield stress (compression) [52]</td>
<td>1.863</td>
<td>parallel</td>
</tr>
<tr>
<td></td>
<td>1.829</td>
<td>perpendicular</td>
</tr>
<tr>
<td>Hydrostatic yield stress (tension) [53]</td>
<td>1.54</td>
<td>-</td>
</tr>
<tr>
<td>Uniaxial yield stress (compression) [41]</td>
<td>2.158</td>
<td>parallel</td>
</tr>
<tr>
<td></td>
<td>1.703</td>
<td>perpendicular</td>
</tr>
<tr>
<td>Uniaxial yield stress (tension) [41]</td>
<td>2.502</td>
<td>parallel</td>
</tr>
<tr>
<td></td>
<td>2.023</td>
<td>perpendicular</td>
</tr>
</tbody>
</table>

As the crushable foam material model still required plastic stress-strain data in the range of expected strains (< 90%), uniaxial compression testing was performed on 25.4 mm polyurethane foam cubes under quasi-static (5 – 10 mm/min) and fast rate (250
mm/s) crosshead speeds (see Figure 6.4). The elastic modulus and collapse stress from these tests are presented in Table 6.3 and the stress-strain plots are shown in Figure 6.5. The stress-strain curves were computed by assuming a uniform crushing within the 25.4 mm foam specimens. In general, the stress-strain responses at all MTS crosshead speeds were similar such that an average value could be used for input into the Abaqus FE simulation.

Figure 6.4. Quasi-static foam crush test on specimen FMQ0K
Table 6.3. Polyurethane foam quasi-static and fast rate test material data

<table>
<thead>
<tr>
<th>Test</th>
<th>Modulus Average (MPa)</th>
<th>St. dev. (%)</th>
<th>Collapse Stress Average (MPa)</th>
<th>St. dev. (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 mm/min</td>
<td>76.9</td>
<td>4.81</td>
<td>2.10</td>
<td>6.48</td>
</tr>
<tr>
<td>6 mm/min</td>
<td>60.8</td>
<td>19.31</td>
<td>1.85</td>
<td>10.98</td>
</tr>
<tr>
<td>10 mm/min</td>
<td>72.5</td>
<td>15.69</td>
<td>1.99</td>
<td>9.57</td>
</tr>
<tr>
<td>250 mm/s</td>
<td>76.3</td>
<td>12.37</td>
<td>2.54</td>
<td>3.57</td>
</tr>
</tbody>
</table>

Figure 6.5. Quasi-static and fast rate polyurethane foam stress-strain data

It should be noted that the stress-strain curves in Figure 6.5 are the nominal (engineering) stress-strain values ($\sigma_{no}$ and $\varepsilon_{no}$, respectively) in absolute terms whereas Abaqus requires the true stress $\sigma_{tr}$ and plastic strain $\varepsilon_{pl}$, also in absolute values. These quantities can computed from Equations 6.1 - 6.2 with the caveat that true stress ($\sigma_{tr}$ in Equation 6.1) should not be defined as per Equation 6.3 which assumes an incompressible material (no change in volume) and does not accurately reflect the
behavior of the polyurethane foam under crushing. Instead, it was assumed that the cross sectional area of the foam did not change (common for crushable foam materials having an effectively low Poisson’s ratio) such that the nominal stress $\sigma_{no}$ was used in place of the true stress $\sigma_{tr}$. No modifications were required for the true strain in Equation 6.2 since it is based on the change in length and not volume. After some tuning to account for both quasi-static and dynamic impacts up to 28.8 m/s impacts, the polyurethane crushable foam was defined (see Table 6.4 and Figure 6.6, “FE input”).

\[
\varepsilon_{pl} = \varepsilon_{tr} - \frac{\sigma_{tr}}{E}
\]  

(6.1)

\[
\varepsilon_{tr} = \ln (1 + \varepsilon_{no})
\]  

(6.2)

\[
\sigma_{tr} = \sigma_{no}(1 + \varepsilon_{no})
\]  

(6.3)

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m$^3$)</th>
<th>Elastic Modulus (MPa)</th>
<th>Poisson's ratio</th>
<th>Plastic Yield stress (MPa)</th>
<th>Stress (MPa)</th>
<th>Strain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poly foam</td>
<td>160</td>
<td>80.9</td>
<td>0</td>
<td>2.50</td>
<td>2.80</td>
<td>0.123</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>4.18</td>
<td>0.700</td>
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<tr>
<td>Abaqus Parameters</td>
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<td></td>
<td></td>
<td>8.24</td>
<td>1.200</td>
<td></td>
</tr>
<tr>
<td>$k$</td>
<td>1.16</td>
<td></td>
<td></td>
<td>500</td>
<td>2.234</td>
<td></td>
</tr>
<tr>
<td>$kt$</td>
<td>0.83</td>
<td></td>
<td></td>
<td>1000</td>
<td>4.343</td>
<td></td>
</tr>
</tbody>
</table>
Stress values > 8.73 MPa and the corresponding strains were selected to provide model stability in instances of high strain crushing and were not representative of actual values recorded during uniaxial testing (see Table 6.4 and Figure 6.6).

Figure 6.6. Various stress-strain profiles for polyurethane foam

A different 80 kg/m³ polyurethane foam was cast in a three step process to create the Hopkinson bar projectile foam bodies (see Chapter 5.2 and Appendix D). The material properties were variable depending on the quality of casting, but since the projectile body only served as the vehicle for the aluminum tip (which was responsible for the loading pulse), extensive validation tests were not performed. With the exception of density, the elastic properties and plastic stress-strain values shown in Table 6.5 were used without tuning based on [25] while Abaqus parameters $k$ and $kt$ were estimated from
Density was increased to 160 kg/m³ in order for the simulated mass to be equal to the actual projectile.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m³)</th>
<th>Modulus (MPa)</th>
<th>Poisson's ratio</th>
<th>Elastic Stress (MPa)</th>
<th>Plastic Stress (MPa)</th>
<th>Strain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foam proj</td>
<td>80</td>
<td>32</td>
<td>0</td>
<td>1.00</td>
<td>1.40</td>
<td>0.817</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2.10</td>
<td>1.415</td>
<td></td>
</tr>
<tr>
<td>Abaqus Parameters</td>
<td></td>
<td></td>
<td></td>
<td>5.00</td>
<td>2.361</td>
<td></td>
</tr>
<tr>
<td>k</td>
<td>1.32</td>
<td></td>
<td></td>
<td>10.0</td>
<td>3.300</td>
<td></td>
</tr>
<tr>
<td>kt</td>
<td>1.00</td>
<td></td>
<td></td>
<td>100</td>
<td>4.000</td>
<td></td>
</tr>
</tbody>
</table>

The 500 kg/m³ aluminum foam core of specimen 1435 also used a crushable foam material model like the polyurethane foam, but with isotropic hardening (not volumetric) based on [53, 60]. Abaqus parameter $k$ was estimated from [60] to be 1.43 and a plastic Poisson’s ratio $\nu_{pl}$ was defined in [53] to be a function of $k$ and computed to be 0.16. As before, the plastic loading regime required a stress-strain input, which could be obtained from the MTSQ 1435 quasi-static test data that included both force and displacement (see Chapter 5.1). Dividing the force by the cross sectional area provided the stress values. For the strain values, it was assumed that only the 21.7 mm thick core crushed. This assumption was possible since the other two remaining materials in the 1435 specimen were solid aluminum (non-yielding for the uniaxial compression stresses) and the thin polymer layer (4.5% of the total thickness and intact post-testing). The core was also assumed to have uniform crushing. After conversion to the true stress-strain using the
nominal stress substitution, the Abaqus inputs were defined as presented in Table 6.6 and
Figure 6.7. As with the polyurethane foam model, stress values > 15.3 MPa were
selected to provide model stability in high crush simulations.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m³)</th>
<th>Elastic Modulus (MPa)</th>
<th>Poisson's ratio</th>
<th>Plastic Yield stress (MPa)</th>
<th>Stress (MPa)</th>
<th>Strain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alum foam</td>
<td>500</td>
<td>202</td>
<td>0</td>
<td>2.04</td>
<td>2.41</td>
<td>0.086</td>
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<tr>
<td>Abaqus Parameters</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>15.0</td>
<td>0.600</td>
</tr>
<tr>
<td>k</td>
<td>1.43</td>
<td></td>
<td></td>
<td></td>
<td>45.0</td>
<td>1.200</td>
</tr>
<tr>
<td>v_{pl}</td>
<td>0.16</td>
<td></td>
<td></td>
<td></td>
<td>150</td>
<td>1.750</td>
</tr>
</tbody>
</table>

Figure 6.7. Various stress-strain profiles for specimen 1435
6.2 Model Validation at Quasi-Static to 7.30 m/s Loading Rates

With the material properties defined, a three component finite element model was created in order to examine the quasi-static responses of both polyurethane and aluminum foams. These materials were of particular interest since they used a more complicated material model definition. The FE model consisted of a fixed metal base and a loading “puck” designed to crush the test specimen using either prescribed displacement or velocity. All components used reduced integration solid elements (C8DR).

For quasi-static testing of the prismatic aluminum and polyurethane foam specimens, the loading puck was assigned a linearly increasing displacement in Abaqus/Standard (see Figures 6.8 and 6.9). The displacement was configured to achieve approximately 90% strain in the regions experiencing uniform crushing.

Figure 6.8. FE model of MTS quasi-static testing on 25 mm foam cube
Low speed dynamic testing using Abaqus/Explicit was conducted on the shaped polyurethane foams to replicate the 7.30 m/s pendulum impact tests previously discussed in Chapter 3.2. A similar three component FE model was used, but in this simulation, the loading puck was assigned an initial velocity of 7.30 m/s and the density was increased to reflect the actual 4.55 kg mass used in the pendulum impactor (see Figure 6.10). The loading puck was allowed to impact and rebound in one direction only to simulate an ideal pendulum impact event. Just as the real model, the force time history was measured rather than the stress-strain for validation with the actual data set. This model had the added benefit of directly comparing the effects of foam geometry on transmitted forces.
The stress-strain results of quasi-static testing using the crushable foam model are shown in Figures 6.11 and 6.12. The polyurethane foam used two meshes approximately 2.5 mm (1,000 elements) and 1.2 mm (8,400 elements) in size, but no appreciable differences in the stress-strain were observed (see Figures 6.8 and 6.11). Quasi-static testing of the aluminum foam matched the Abaqus/Standard FE predicted stress-strain data well (see Figure 6.12). The 2 mm mesh size for specimen 1435 (12,844 elements) was actually too refined to use in large panel FE testing, so a coarser 3 mm mesh (5,184 elements) was examined in Abaqus/Explicit and found to behave identically while using fewer computational resources. The low collapsing stress relative to the actual data was necessary for simulating the dynamic response during the HOPK 38 – 98 m/s impacts.
Figure 6.11. FE validation of 25.4 mm polyurethane foam cubes with crushable foam material model

Figure 6.12. FE validation of coupon specimen 1435 with crushable foam material model
Examining the 7.30 m/s pendulum impacts on the tetrahedral shaped foam demonstrated a good correlation with the actual test data using either 2.5 mm (11,930 elements) or 1.8 mm (32,592 elements) mesh size (see Figures 6.10 and 6.13). A similar 2.5 – 1.8 mm mesh refinement was applied to the rectangular and V channel shaped foam specimens (see Figure 6.14). The transmitted forces showed almost no difference between the two mesh sizes for either rectangle or V channel pendulum impacts (see Figures 6.15 and 6.16) although it observed to be higher than the actual test results for all three foam geometries. The higher contact forces was indicative of a stiff foam material system, which was necessary for matching the force response of the large panel tests at higher impact velocities (23.0 – 24.6 m/s).

Figure 6.13. FE validation of 7.30 m/s pendulum impact force time history on tetrahedral polyurethane foam with crushable foam material model
Figure 6.14. FE model of rectangle and V channel shaped foams

Rectangle
6,760 Elements

V Channel
6,552 Elements

Figure 6.15. FE validation of 7.30 m/s pendulum impact force time history on rectangle polyurethane foam with crushable foam material model
6.3 **MODEL VALIDATION AT 28.8 – 98 M/S DYNAMIC LOADING RATES**

For dynamic material validation, Abaqus models were constructed to replicate Hopkinson bar impact tests. The force time history response of the FE polyurethane was compared with the gas gun tests performed at 28.8 m/s while the aluminum foams were compared to the HOPK tests at 38 – 98 m/s.

Two different projectile models were created to match the actual gas gun test setup used for each material. Both models included the full length Hopkinson bar (3.19 m) consisting of an aluminum tube (3.16 m) merged with a 25.4 mm thick solid front cap. This was necessary in order to allow the stress waves to propagate within the FE models similar to the actual Hopkinson bar setup. The shaped polyurethane foam impact at 28.8
m/s used a solid aluminum body with plastic loading cap onto which the foam was
mounted (see Figure 6.17). The projectile was actually simplified as a rectangular prism
to match the geometry of the foam shaper and therefore used a modified density to yield
the same 875 g overall mass as the actual projectile.

![Figure 6.17. FE model of projectile and Hopkinson bar for polyurethane foam pressure pulse
shaping geometries (25 mm tetrahedral shown)](image)

The force time history for the tetrahedral shaped foam projectile impact showed
no difference between the 2.4 mm (33,272 elements) and 1.2 mm (103,936 elements)
mesh refinement (see Figures 6.18 and 6.19). In fact, the finer mesh did not complete the
full simulation and stopped at approximately 0.8 s, possibly due to hardware constraints.
The predicted response did match the actual Hopkinson bar tests performed on 25 mm
tall tetrahedral shaped foam. Compared to the low speed pendulum impact test, the
Hopkinson bar measured forces were lower than the actual test data but matched the
overall force time history profile very well. Since mesh refinement did not appear to
affect the force results, the material card was well tuned for the 28.8 m/s impact velocity.
Figure 6.18. FE details of mesh refinement for 25 mm tetrahedral foam

Figure 6.19. FE validation of 25 mm tetrahedral polyurethane foam shaper Hopkinson bar impact at 28.8 m/s
Another set of Hopkinson bar tests examined the reduced height, 14 mm tall foam pressure pulse shaper that were originally designed increase the strain in the shaped tetrahedral region (see Chapter 3.2). Using the same FE projectile and velocity as the 25 mm tall foam projectiles, two mesh refinements were modeled using approximate element sizes of 2.4 mm (1,050 elements) and 1.2 mm (63,616 elements) as shown in Figure 6.20. The contact force time histories were identical for the two meshes and also showed good correlation with the actual test data (see Figure 6.21). This correlation with the actual force data, which was also observed for the 25 mm tall specimen, validated both the tetrahedral geometry and the crushable foam material model for the FE tiled projectile which used a similar foam shape and velocity.

![Figure 6.20. FE details of mesh refinement for 14 mm tetrahedral foam](image)
Figure 6.21. FE validation of 14 mm tetrahedral polyurethane foam shaper Hopkinson bar impact at 28.8 m/s

For replicating the HOPK tests on specimen 1435 at 38 – 98 m/s, an aluminum tipped foam body projectile was created in Abaqus/Explicit (see Figure 6.22). The aluminum foam core specimen was positioned at the center of the Hopkinson bar strike face and the projectile assigned a velocity of 38, 70, and 98 m/s. A 44.5 x 44.5 x 3.05 mm polyurethane foam was applied to the impact side of the aluminum face, aluminum foam core specimen for pulse shaping similar to the actual HOPK tests. However, the thinness of the foam effectively increased the strain rate through the material which caused excessive element distortions. Since the foam was used and validated primarily for the large panel tiled projectile, the HOPK tests at 98 m/s were conducted with the foam layer removed which allowed the analysis to complete.
The force time history profile for the 38 m/s projectile impact onto specimen type 1435 is shown in Figure 6.23. The overall initial force loading response, the onset of yielding, and the pressure pulse duration of the FE model matched the actual test data. The drop in the FE force data between 0.43 – 0.65 ms was due to the specimen almost losing contact with the Hopkinson bar. In the actual test system, this was mitigated by the use of the foam pulse shaper to help maintain contact between projectile and specimen and a vicious grease (not included in FE model) to maintain contact between the specimen and strike face of the Hopkinson bar.
At 70 m/s, several differences between the FE predicted force transmission through the coupon sandwich specimen and the recorded data were visible (see Figure 6.24). The collapse stress in the material around 10 MPa was lower than the observed collapse occurring at 20 MPa, although the maximum contact stress through the specimen was the same for both simulation and recorded data. The low collapse stress was a necessary requirement of validating the same material model at both quasi-static and dynamic loading rates.

At impact velocities beyond 70 m/s, the foam pulse shaper experienced excessive distortion and caused the analysis to stop. Removing the foam pulse shaper caused the initial force loading to be higher and increased the force in pulse contact regime, but the overall loading profile remained the same (see Figure 6.24).
The largest discrepancy in the validation of the FE predictions with the HOPK test results occurred during the unloading portion of the force time history profile (see Figure 6.24). This was due to dynamic vibrations within the specimen causing a momentary separation \( (t = 0.62 \text{ ms}) \) from the Hopkinson bar where the contact forces were queried. A similar phenomena was also observed in the actual HOPK tests where the front loading cap separated from the remainder of the force measurement bar although this separation occurred at a much later time in the impact event (approximately 1.4 ms vs. 0.62 ms, see Figure 6.25).
Based on these results, was believed that the FE foam projectile’s plastic material properties were stiffer in comparison to the actual material and that contrary to earlier assumptions, the projectile’s crush characteristics were responsible for a much larger degree of the force response in the unloading regime. However, due to the nature of dynamic testing, especially at the NEM test levels, accurate modeling of the type 1435 aluminum foam core material was required primarily during the loading phase, which appeared to be replicated even at the 70 m/s FE simulations. Therefore, since the force response of the 1435 aluminum foam did correlate to the actual test data with respect to
initial load response, and both predicted and recorded crush behavior was similar (9.6% vs. 8.6%, in Figure 6.26), the aluminum foam core material was deemed acceptable and additional validation tests of the projectile itself were not pursued.

![Figure 6.26. FE comparison of specimen 1435 deformation behavior at 38 – 98 m/s, actual velocities per specimen as indicated](image)

The last force time history material test on the specimen 1435 material at 98 m/s showed much of the same trends at the 70 m/s impact, including a low yield stress and momentary specimen separation (see Figure 6.27). Maximum contact forces were higher for the FE simulations while the specimen height reductions were lower (13.6% vs. 36.5%) compared to the HOPK test results (see Figure 6.26). This was indicative of strain rate effects in the aluminum foam core of specimen 1435, which was required to
prevent element distortion under the extremely high strain rate FE blast loads. This requirement also led to replacing the polymer layer in specimen 1435 with the aluminum foam core material (negligible differences at 98 m/s) so that the FE blast events could be simulated without excessive element distortions in the layer (see Figure 6.28).

![Figure 6.27. FE validation of 98 m/s HOPK impact force time history on coupon specimen 1435](image)

![Figure 6.28. FE details of sandwich panel type 1435 with modified polymer layer for dynamic blast simulation](image)
Summarizing the coupon material validation tests, the FE polyurethane foam pulse shapers matched test results up to 28.8 m/s impact velocity. Since the dynamic simulation used a tetrahedral geometry and impact speed that was similar to the real tiled projectile, the tuned foam projectile was expected to impart a similar pressure pulse as the actual system. For the 38 and 70 m/s impacts onto specimen 1435, the FE simulation matched the dynamic force response of the HOPK test series. Since the NEM projectile impacts did not cause significant core crushing and used a lower velocity, this material model was expected to be sufficient for FE simulations of the NEM impact event. Simulating the higher strain rate of an ideal blast loading required using a modified 1435 sandwich specimen that behaved similar to the un-modified material at the highest dynamic impact velocities tested (98 m/s). However, as validation testing of the material alone at blast-like loading conditions was not available, comparison could only be made with the OBLT large panel data.

6.4 LARGE PANEL VALIDATION WITH NON-EXPLOSIVE TEST DATA

With material validation tests complete, large panel assemblies were constructed in order to compare the non-explosive and actual blast test methodologies. This required first validating the FE models for each test method using all of the test results presented in Chapter 4.

The FE models incorporated nearly every component (sans bolts and reaction block) on the impact side of the test setup, including the steel supports, gussets, window
frames, transmission plate, panel, and projectile. The purpose of including this level of
detail was twofold: to provide a more realistic finite element simulation capturing the
interaction between components and to examine the stress levels within certain
components when subject to high impulse loading conditions. This latter requirement
was part of the fixture design process and was the reason the window fixtures were
switched to the 4130 chromoly units prior to NEM2 and OBLT testing.

To reduce the amount of computation given the number of components, the
assembly used quarter symmetry about the X and Y planes (loading in the +Z direction).
Likewise, many of the test fixture components were modeled with reduced shell elements
(C4DR) as through-thickness stress distributions were not of primary concern. Parts
thicker than 6.35 mm were modeled with reduced integration solid elements (C8DR).
The RHA panel, while only 6.35 mm thick, initially used shell elements for examining
the fixture components, but was later redefined with a four layer solid mesh for obtaining
through thickness stresses (see Figure 6.29). The 1435 series sandwich panel used solid
elements (see Figure 6.30).

Boundary conditions were fixed at the base of the supports to replicate the actual
test conditions. Translations perpendicular to the planes of symmetry and rotations about
the X and Y axis were not permitted as required for symmetric boundary conditions. Tie
constraints for both RHA and sandwich panel were added to the panel and window
fixtures at the same location as the physical bolts (see Figure 6.31). As the real bolts
were only hand tightened, no preload was defined.
Figure 6.29. FE model of RHA panel and fixture for NEM and OBLT simulations

Figure 6.30. FE model of panel 1435 and fixture for NEM and OBLT simulations
The pressure pulse loading components were modeled as two separate systems to replicate the two different projectiles used in each NEM test series. For the NEM1 finite element simulation, each tile block was identical, with the exception of being partitioned depending on the block location within the quarter symmetric model (see Figure 6.32). The center block was also simplified to a solid block with no holes cut for the outgas ports or supporting shaft. All blocks were the same height which was consistent with the original NEM1 projectile. The thin aluminum backing sheet was created with shell elements and tie constrained to the back faces of the tile blocks.

The NEM2 finite element projectile model included both material and height variations to replicate the actual tiled projectile. Tile blocks further away from the center
were partitioned and assigned a higher ratio of aluminum to steel while the blocks closer to the center included more steel and also taller geometries (see Figure 6.33). As with the FE NEM1 model, the center tile was simplified as a solid block without the holes used in the real projectile system.

![FE model of NEM1 projectile impact with all-steel tile blocks and polyurethane foam shaper](image)

*Figure 6.32. FE model of NEM1 projectile impact with all-steel tile blocks and polyurethane foam shaper*

![FE model of NEM1 projectile impact with hybrid tile blocks and polyurethane foam shaper](image)

*Figure 6.33. FE model of NEM1 projectile impact with hybrid tile blocks and polyurethane foam shaper*
The non-explosive finite element analysis was performed on three panels: RHA, 1435, and the 9.53 mm thick aluminum monolithic plate. The impact sequence for the RHA and 1435 sandwich panel are shown in Figures 6.34 and 6.35 and include the three main points of interest: initial contact, transmission plate separation, and the maximum deformation when the projectile came to a complete stop (prior to reversing direction). The entire loading event took < 4 ms to complete. During this stage, the acceleration at the transmission plate’s CG was queried in order to compare to the recorded transmitted accelerations. For the two solid metal panels, strain gauges applied to the actual panel surface were also monitored during the loading process for comparison with FE results. After impact, the projectile reversed direction which unloaded the armor panel and allowed it to oscillate about a yielded plastic deformation state. This time period was used to validate the plastic deformations and deformed panel profiles recorded for the actual RHA, aluminum, and 1435 aluminum foam core sandwich specimen. Transmission plate velocity was also recorded at this stage since it would be in steady-state free flight.
Figure 6.34. FE details of NEM2 RHA impact loading sequence

- **t = 0.24 ms**
  - Projectile first contact

- **t = 2.08 ms**
  - Transmission plate separation

- **t = 2.98 ms**
  - Absolute maximum displacement
Figure 6.35. FE details of NEM1 panel 1435 loading sequence

- $t = 0.12$ ms
  - Projectile first contact

- $t = 2.00$ ms
  - Transmission plate separation

- $t = 3.30$ ms
  - Absolute maximum displacement
The transmission plate accelerometer data was one metric for validating the finite element models during the initial time period just after impact. As the period was very brief, the FE simulation time was reduced to 150 µs for the monolithic panels and 250 µs for sandwich panel 1435. The number of time increments was also increased from 200 to 1,200 in order to capture the high frequency acceleration signal. Both RHA and aluminum panel transmitted accelerations correlated with the finite element models with respect to the magnitude of signal and shape (see Figures 6.36 to 6.38). The average acceleration response for the transmission plate appeared to follow the actual test data for the monolithic panels.

Figure 6.36. FE validation of NEM1 RHA transmission plate accelerometer
Figure 6.37. FE validation of NEM2 RHA transmission plate accelerometer

Figure 6.38. FE validation of NEM2 ALUM transmission plate accelerometer
The sandwich panel had similar magnitude transmitted accelerations, but did not correlate to the shape of the initial pulse after projectile impact (see Figure 6.39). This may have been caused by the FE 1435 material not completely capturing the stress wave propagation through the sandwich material. The actual accelerometer data may have also had some errors since NEM1 tests used a different cable routing that was observed to have issues (see Chapter 4.6).

![Figure 6.39. FE validation of NEM1 panel 1435 transmission plate accelerometer](image)

The transmission plate velocity also provided means of validating the finite element models. Based on the NEM results in Chapter 4.5, it was expected that the velocities would be consistent for all three panel types (RHA, specimen 1435, and ALUM). Measuring the velocity in the finite element simulation involved a different
process compared to the actual test data. Whereas the actual test data only permitted a short time window when the transmission plate would simultaneously be not accelerating (steady-state flight) or decelerating (impacting the foam and rubber deceleration system), the finite element model would actually remain at the same velocity indefinitely in the absence of any friction, drag, or other energy losses. However, without those same energy losses, the transmission plate tended to oscillate once impacted which made comparing velocities more difficult.

The solution for extracting the transmission plate velocity was to use an extended simulation run time (25 ms) such that the transmission plate entered steady-state oscillations. Then the velocity was averaged over two periodic cycles and compared with the actual data: RHA from both NEM tests, the NEM2 aluminum plate, and the NEM1 1435 panel test (see Figures 6.40 to 6.43). It should be noted that the actual computed RHA transmission plate velocities did not have useable accelerometer data so the velocity computed using the high speed video was substituted instead. As an example, the high speed video showed RHA transmission plate velocity averaged 22.32 m/s with 1.82% standard deviation for all three NEM2 tests and the steady-state velocity in the finite element simulation oscillated about 22.65 m/s (see Figure 6.41). This represented a difference of less than 1.5% between the FE model and the actual results. For all four panel tests (NEM1 and NEM2 RHA, NEM2 ALUM, and NEM1 panel 1435), the FE predicted steady-state velocity showed good agreement with the actual test data. Values are shown on each graph representing the averaged FE and NEM data.
Figure 6.40. FE validation of NEM1 RHA transmission plate velocity

Figure 6.41. FE validation of NEM2 RHA transmission plate velocity
Figure 6.42. FE validation of NEM2 ALUM transmission plate velocity

Figure 6.43. FE validation of NEM1 panel 1435 transmission plate velocity
Validation of the FE models using the deformation profiles was done in two stages: the deformation profile and the plastic deformation at the panel center (see Figures 6.44 and 6.45). The deformation profile was a difficult metric to implement due to uncertainties in the exact yielded profile. Unlike the center deformation measured at one point in Figure 6.45, the deformation profile oscillated about every point along the panel centerline at different time intervals. Thus, while the center of the panel may have been at a minimum, points nearer the clamped boundaries edge may have been at a relative maximum (see “Osc. min” in Figure 6.44). A decision was made to use the average of the oscillating deformation profiles. This may not have been an actual yielded profile, but it provided a common metric for comparison to the actual panel deformations recorded post impact. Figures 6.46 and 6.47 show the FE deformation profiles for the solid aluminum and the type 1435 sandwich panel. Only front face deformations are plotted for the monolithic panels for clarity since the back face deformations are identical. Each plot includes the absolute maximum deformation when the projectile was at rest as well as the oscillating minimum and maximum values when the panel was unloaded post-impact. The FE predicted deformation for the sandwich panel was almost identical to the actual deformation profile. Also, examining the internal stress state at a single time point appeared to predict the location of stress concentrations that were manifested as core cracks in the waterjet section of panel 1435 (see Figure 6.48).
Figure 6.44. FE validation of NEM1 RHA panel deformation profile (front only) including absolute maximum, maximum oscillating, and minimum oscillating profiles.

Figure 6.45. FE comparison of NEM1 and NEM2 RHA displacement time history profiles.
Figure 6.46. FE validation of NEM2 ALUM deformation profiles (front only) including absolute maximum, maximum oscillating, and minimum oscillating profiles.

Figure 6.47. FE validation of NEM1 panel 1435 panel deformation profiles including absolute maximum, maximum oscillating, and minimum oscillating profiles.
Figure 6.48. FE comparison of NEM1 internal stress state of panel 1435 with impact damaged panel

With the FE predicted plastic deformation state available, the deformation at the panel center were compared to the measured post impact deformations (dent depth) of the actual test panels (see Table 6.7). For all panels, the FE predicted center deformations were higher than the actual values recorded, although this error was only 2.72% for the sandwich specimen.

<table>
<thead>
<tr>
<th>Test</th>
<th>Actual (mm)</th>
<th>FE (mm)</th>
<th>Difference (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>RHA NEM1</td>
<td>4.89</td>
<td>8.45</td>
<td>72.8</td>
</tr>
<tr>
<td>RHA NEM2</td>
<td>5.00</td>
<td>8.85</td>
<td>77.0</td>
</tr>
<tr>
<td>ALUM NEM2</td>
<td>21.53</td>
<td>26.02</td>
<td>20.9</td>
</tr>
<tr>
<td>1435 NEM1</td>
<td>18.00</td>
<td>18.49</td>
<td>2.72</td>
</tr>
</tbody>
</table>

As a final validation test on the aluminum and RHA panel, two strain gages were installed to the back face during NEM2 testing. One gage (SG1) was installed at the panel center and the other gage (SG2) installed 88.9 mm away from the side edge, along
the horizontal axis (see Figure 6.49). Only one panel of each monolithic material was tested and the results are shown in Figures 6.50 to 6.53. The RHA and ALUM panel showed some correlation at the side SG2 location, but did not exhibit any correlation at the center location SG1. Unfortunately, there was no explanation for this discrepancy in the results.

Figure 6.49. Strain gage locations (dimensions in mm)
Figure 6.50. FE validation of NEM2 RHA strain data SG1

Figure 6.51. FE validation of NEM2 RHA strain data SG2
Figure 6.52. FE validation of NEM2 ALUM strain data SG1

Figure 6.53. FE validation of NEM2 ALUM strain data SG2
In summary, finite element simulations of the non-explosive test methods were validated using one or more of the available NEM test data, including transmission plate velocities, transmission plate accelerations, panel deformations, and strain gages. There existed errors between the predicted and recorded values, but in almost all cases, the quantities of comparison followed the same trends observed in the test data and in some cases, e.g., the transmission plate velocities, the FE simulation was within a few percent of the actual values.

6.5 LARGE PANEL VALIDATION WITH EXPLOSIVE TEST DATA

Blast loading in Abaqus/Explicit was accomplished using the same NEM RHA and panel 1435 models in Figures 6.29 and 6.30 but replacing the projectile with idealized pressure pulses. The triangular pressure pulses were originally created based on ATBlast calculations to define the mass and height variations for the NEM2 projectile (see Figures 3.21 and 3.22 in Chapter 3.3). To maintain continuity with the mechanical impacts, the A – F tile block designations were used to load the corresponding location on the panel surface. Additional locations G – J filled in the 38.1 gap between the projectile and window fixture to better represent the actual blast pressure pulse that was applied to the full panel surface (see Figure 6.54).
Using the designation A – J, each 130 µs pressure pulse was tuned with respect to magnitude and time delay to replicate the 8,460 Pa-s impulse event (actual maximum impulse was 8,306 Pa-s). For example, the center of the panel (location F) used a 127.8 MPa pressure pulse magnitude that arrived with zero time delay while location G at the corner of the panel used a 69.9 MPa pressure pulse that arrived 40 µs after the center was loaded (see Figure 6.55 and Table 6.8). Unlike the NEM impacts, locations B/C and D/E were not combined but instead used discrete impulses.
Figure 6.55. FE details of idealized blast loading pressure time history profile

Table 6.8. FE idealized blast pressure pulse loading values

<table>
<thead>
<tr>
<th>Location</th>
<th>Impulse (Pa-s)</th>
<th>Max Pressure (MPa)</th>
<th>Time delay (µs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>5,811</td>
<td>89.4</td>
<td>-23.3</td>
</tr>
<tr>
<td>B</td>
<td>6,490</td>
<td>99.8</td>
<td>-11.4</td>
</tr>
<tr>
<td>C</td>
<td>6,809</td>
<td>104.7</td>
<td>-6.7</td>
</tr>
<tr>
<td>D</td>
<td>7,384</td>
<td>113.6</td>
<td>-4.0</td>
</tr>
<tr>
<td>E</td>
<td>7,816</td>
<td>120.2</td>
<td>0.0</td>
</tr>
<tr>
<td>F</td>
<td>8,306</td>
<td>127.8</td>
<td>0.0</td>
</tr>
<tr>
<td>G</td>
<td>4,540</td>
<td>69.9</td>
<td>-40.0</td>
</tr>
<tr>
<td>H</td>
<td>5,083</td>
<td>78.2</td>
<td>-30.0</td>
</tr>
<tr>
<td>I</td>
<td>5,567</td>
<td>85.7</td>
<td>-22.5</td>
</tr>
<tr>
<td>J</td>
<td>5,788</td>
<td>89.0</td>
<td>-20.0</td>
</tr>
</tbody>
</table>

Figure 6.56 shows the progression of core crush at impact and at t = 100 and 200 µs (detonation occurred at t = 10 µs). Figure 6.57 provides a visual comparison of the
internal stress state between the FE model and post impact OBLT tested panel 1435. The maximum stress limit was set to 20 MPa in both figures which coincided with the maximum densified stresses measured during the MTSQ tests for the 1435 material.

$t = 0 \mu s$, detonation at $t = 10 \mu s$

$t = 100 \mu s$

$t = 200 \mu s$

Figure 6.56. FE details of OBLT stress distribution through panel at detonation ($t = 10 \mu s$) and select time intervals
Validating the finite element models for the actual blast test data (OBLT) was not as straightforward as the NEM simulations. Accelerometer data were saturated for most tests, high speed video capture was not available, and the deformation profiles were not very consistent. Therefore, validating the models became a more qualitative comparison of these criteria.

The transmission plate velocities were extracted for the RHA and type 1435 panel by computing the average oscillating value with an extended simulation run time (see Figures 6.58 and 6.59). The finite element simulation predicted much higher velocities than what was calculated from the hang time for either panel. This may be attributed to the material models, which were not validated for these strain rates.
Figure 6.58. FE validation of OBLT RHA transmission plate velocity

Figure 6.59. FE validation of OBLT panel 1435 transmission plate velocity
The FE transmitted accelerations measured at the transmission’s plate center of gravity appeared to follow OBLT accelerations prior to signal saturation (see Figures 6.60 and 6.61). The RHA accelerations, for example, showed the primary response with similar time duration (approximately 12 µs) as the OBLT data while the sandwich panel FE transmitted acceleration had the two maximum values that appeared in the real detonation. Further validation with the transmitted accelerations may not be possible due to uncertainties in the actual sensor response when loaded past its rated limit of 50,000 g.

Figure 6.60. FE validation of OBLT RHA transmission plate acceleration
As with the NEM tests, the deformation profile and dent depth provided another metric for comparison with the FE models. The center deflection time histories for the RHA and type 1435 panel are shown in Figures 6.62 and 6.63. The predicted displacement for the RHA panel was much higher than recorded, 13.75 vs. 2.97 mm, while the center point deflection for the sandwich panel (28.56 mm) was within 2.2% of the recorded value (29.21 mm).
Figure 6.62. FE validation of OBLT RHA displacement time history profile

Figure 6.63. FE validation of OBLT panel 1435 displacement time history profile
Deformation profiles across the panel centerline were extracted for both armor panels (see Figures 6.64 and 6.65). The front face deformation profile for the RHA simulation was much higher and more localized compared to the OBLT measured profile. When averaging the minimum and maximum oscillations, the deformed profile also exhibited some of the same wave-like characteristics that were present in the real panel. In contrast, the FE simulated deformation profile for panel 1435 exhibited a more global deformation while the actual test panel strike face had a localized deformation above the close-in blast. The deformation results may be explained by the use of material models for both RHA and aluminum foam that were not validated for the highly dynamic theoretical blast loads.

Figure 6.64. FE validation of OBLT RHA panel deformation profiles (front only) including absolute maximum, maximum oscillating, and minimum oscillating profiles
In summary, blasts are notoriously unpredictable and difficult to replicate exactly even under the most controlled circumstances. The finite element simulations were conducted with an idealized blast, but without knowing the full pressure distribution or proven material models at the strain rates in questions, the best simulation can only provide estimates for the structure’s behavior. Thus, the fact that the blast results are within the same magnitude of the real values using only relatively low-strain rate validated material models and unconfirmed theoretical loading profiles provides some validation of the FE methods and assumptions made to this point.
6.6 LARGE PANEL PREDICTED VALUES

Having demonstrated that the finite element models match the actual test data within a reasonable degree of accuracy, the simulations can now be used to explore the test methodologies, panel response, and validate certain assumptions made during this investigation. Topics of interest to this current investigation are:

i. Differences, if any, between the NEM1 all-steel projectile and the NEM2 hybrid steel/aluminum projectile with respect to loading (Chapter 3.3).

ii. Impulse matching to produce comparable contact stresses between explosive and non-explosive tests (Chapter 2.2).

iii. Effect of the transmission plate to internal damage of a sandwich panel (Chapters 4.7 to 4.8).

Additional topics of interest or more precise measurements may be investigated at a future time with more advanced FE models.

As detailed in Chapter 3.3, developing the tiled projectile required a significant amount of effort, especially to create the spatially and temporally varying steel and aluminum tile blocks used during the second phase of non-explosive testing. Since both projectiles were created in the FE environment, it was desired to compare and contrast the differences between the two systems. The two non-explosive projectiles were first compared with respect to the maximum deformation profiles. The FE deformation results for RHA in Figure 6.45 suggest that both NEM projectiles would cause similar panel behavior, despite the 9.9 kg difference in projectile mass and 1.68 m/s difference in
speed. This was confirmed through the profiles of maximum deformation for both RHA and the type 1435 sandwich panel (see Figure 6.66). Furthermore, the similarities in magnitude for the measured deformation, transmitted velocities, and transmitted accelerations in Chapter 4 suggested that the two projectiles created a similar impact event. Examining the FE results of these events directly for the two methods in Figures 6.66 to 6.69 indicates that indeed, the impact for the two projectiles should be similar.

Figure 6.66. FE comparison of NEM1 and NEM2, RHA and panel 1435 deformation profiles
Figure 6.67. FE comparison of NEM1 and NEM2 panel 1435 transmitted velocities

Figure 6.68. FE comparison of NEM1 and NEM2 RHA transmitted accelerations
An interesting observation was noted in the initial transmitted acceleration response of the transmitted acceleration of the NEM1 aluminum plate being similar to the NEM2 1435 panel (see Figure 6.70). Both panels had a similar thickness aluminum front face (10 mm for panel 1435 and 9.53 mm for ALUM) and the core of the sandwich panel appeared to significantly reduce the oscillations in the accelerometer data. This result seems to indicate that the initial behavior of the transmission plate depends on the facesheet material properties and that the FE crushable foam material model may be responsible for some stress wave attenuation.
Since the panel deformation was comparable for the two NEM methods, and both the transmitted velocity and acceleration measured at the transmission plate were also similar, it was expected that both projectiles would induce comparable pressure loads on the panel surface at locations A – F (see Figure 6.54). However, this did not occur. The NEM1 projectile showed the highest forces at the panel corner location A with the panel center F being the lowest initial contact force. This trend was observed for both RHA in Figures 6.71 and 6.72 and the sandwich panel type 1435 in Figures 6.73 and 6.74. While location A remained the highest initial contact force for the NEM2 projectile, the center location F was increased dramatically, with minor increases in the area surrounding the panel center (D/E), and a reduction in initial contact stresses for the outermost layer (A – C) that was closest to the clamped boundary.
Figure 6.71. FE predicted RHA contact pressures from NEM1 loading

Figure 6.72. FE predicted RHA contact pressures from NEM2 loading
Figure 6.73. FE predicted panel 1435 contact pressures from NEM1 loading

Figure 6.74. FE predicted panel 1435 contact pressures from NEM2 loading
The differences between NEM projectiles with respect to contact stresses and impulses in Figures 6.71 to 6.74 suggested that two very different loading scenarios may be possible even though all measured quantities, whether indirectly through the transmission plate acceleration and velocity or directly through the panel deformations indicated that the loading events should be the same.

Examining the specific impulses provides additional insight into the differences between NEM methods and the ideal blast tests as well. The idealized blast loading profiles should result in an identical contact pressure time history measured on the panel surface at the same locations. Both RHA and 1435 panel matched the applied idealized triangular loading to within 14.0%, with the error attributed to the oscillations in the contact stress time histories (see Figures 6.75 and 6.76). Using the same integration computations on the NEM projectile contact pressures in Figures 6.71 to 6.74 showed a different, higher impulse measured at the panel surface compared to the momentum calculation of the projectile (see Tables 6.9 and 6.10). This was not necessarily an error since impulse is defined as difference in initial and rebound mass and velocity and the tiled projectile was observed in the actual NEM tests to have a rebound velocity that was not calculated.
Figure 6.75. FE predicted RHA contact pressures from OBLT loading

Figure 6.76. FE predicted panel 1435 contact pressures from OBLT loading
Table 6.9. FE comparison of RHA calculated impulses

<table>
<thead>
<tr>
<th>Location</th>
<th>Ideal Blast</th>
<th>NEM1</th>
<th>NEM2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Impulse (Pa-s)</td>
<td>Error (%)</td>
<td>Impulse (Pa-s)</td>
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<tr>
<td>A</td>
<td>6,220</td>
<td>7.03</td>
<td>12,362</td>
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<td>B</td>
<td>7,400</td>
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<tr>
<td>C</td>
<td>7,521</td>
<td>10.5</td>
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<tr>
<td>D</td>
<td>7,100</td>
<td>-3.86</td>
<td>10,898</td>
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<tr>
<td>E</td>
<td>7,312</td>
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<tr>
<td>F</td>
<td>7,934</td>
<td>-4.49</td>
<td>12,637</td>
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Table 6.10. FE comparison of panel 1435 calculated impulses

<table>
<thead>
<tr>
<th>Location</th>
<th>Ideal Blast</th>
<th>NEM1</th>
<th>NEM2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Impulse (Pa-s)</td>
<td>Error (%)</td>
<td>Impulse (Pa-s)</td>
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<tr>
<td>B</td>
<td>6,737</td>
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<tr>
<td>C</td>
<td>6,699</td>
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<td>D</td>
<td>6,914</td>
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<tr>
<td>E</td>
<td>7,339</td>
<td>-6.11</td>
<td>10,470</td>
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<tr>
<td>F</td>
<td>7,816</td>
<td>-5.91</td>
<td>10,262</td>
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The impulses presented in Tables 6.9 and 6.10 indicate that neither NEM method could match the ideal loading profile with continually increasing impulses toward the panel center, e.g., impact at F should be higher than E, which in turn should be higher than D. In almost all instances (especially for the NEM simulations), locations B and C near the boundary showed higher contact forces than D near the panel center. This may not necessarily be a fault of the tiled projectile since even for the ideal blast, the measured impulse showed minor deviations from the expected trend of increasing impulse closer to the panel center. Based on these results, it is possible that the
individual stiffness of each panel must be compensated prior to loading, especially for the non-explosive projectile.

Improvements for the NEM2 projectile compared to the NEM1 projectile are also readily apparent in Tables 6.9 and 6.10. The impulse distribution of the NEM2 projectile impact showed a more blast-like pressure pulse distribution with higher impulses towards the panel center compared to the NEM1 projectile. This trend was more pronounced for the RHA panel compared to panel 1435. Thus, while the FE predictions showed that the NEM methods did not achieve the expected impulse distribution profile, they were able to meet or exceed the impulse of a close-in detonation, with some of the same loading characteristics. NEM2 was a worthwhile endeavor in this regard as it created a more blast-like pressure pulse distribution.

As a final investigation into the FE predicted results, it was suggested in Chapter 4.8 that the inertial effect of the panel, made worse by the transmission plate located on the back face, was responsible for the high degree of core crush for the sandwich panels when subject to the actual blast loads. Finite element simulations of the ideal blast event for panel 1435 with respect to panel displacement, velocity, and acceleration time history plots initially showed little difference between the two test setups at detonation (see Figures 6.77 to 6.79). However, once the majority of the theoretical pressure pulses ceased loading the panel at 140 – 160 µs, the effects of the transmission plates were readily observed by the panel’s velocity and displacement measured at the center (see Figures 6.78 to 6.79).
Figure 6.77. FE predicted OBLT 1435 panel center point acceleration with and without transmission plate

Figure 6.78. FE predicted OBLT 1435 panel center point displacement with and without transmission plate
Since the front face of the panel was loaded much earlier than the back face, it appeared that the transmission plate played a significant role in the internal damage of the aluminum foam core specimen. The internal stress state showed that the transmission plate analysis experienced higher stresses > 20 MPa in more locations compared to the internal damage state of the non-transmission plate FE simulation (see Figure 6.80). The non-transmission plate FE analysis also had more stress < 5 MPa (corresponding to the collapse stress of the aluminum foam) in comparison to the transmission plate results. The amount of core crush at the panel center was estimated to be higher by 4.74 mm (22.1% of the available core) for the transmission plate analysis.

Figure 6.79. FE predicted OBLT 1435 panel center point velocity with and without transmission plate
The implications of these results were significant as they suggested that the current standard for measuring the transmitted impulse through monolithic RHA panels might not necessarily be ideal when testing new armor panels incorporating a sandwich construction. Furthermore, as was explored while comparing the NEM methods, the transmission plate and even direct measurement of deformation post impact may not provide discriminating insight into the actual impulse distribution profile.

Figure 6.80. FE predicted OBLT internal stress state of panel 1435 for non-transmission plate and transmission plate loading
7 CONCLUSIONS

A non-explosive methodology (NEM) has been established to simulate the wide area spherical pressure pulse distribution of a close-in blast detonation and create similar damage modes on flexible armor panels. The purpose was to develop a more consistent and controlled environment for assessing armor components and to advance the current state of armor development by reducing the dependency on actual blast tests for dynamic, high impulse, and wide area pressure loads. The non-explosive methodology centered on the concept of impulse matching, where a high pressure and short duration blast detonation event (1.37 kg C4 at 305 mm standoff) was replicated with a longer duration and lower pressure pulse projectile impact (50.4 – 60.3 kg at 23.0 – 24.6 m/s). As both events were dynamic in nature, it was expected that the common 8,460 Pa-s impulse would result in the same damage modes and extent of damage on conventional steel and prototype sandwich armor panels as an actual explosive detonation despite the estimated order of magnitude difference in both pulse time and maximum pressure. This motivated the current investigation and the contribution to the blast community to assess an equivalent non-explosive projectile impact for replicating the effects of close-in spherical blast pressures on flexible armor panels.

For the large panel RHA steel tests, the difference in the post-impact damage (deformation profile and extent of damage) between the actual blast tested panels and the non-explosive tested panels was virtually imperceptible—with the exception that the actual blast event was simply more dynamic (excited faster panel response). In all cases,
the available accelerometer data were an order of magnitude higher than the non-
explosive measured transmitted accelerations. While the effects of this intensely
dynamic load were not evident in the monolithic RHA panel, the actual blast tested
sandwich panels exhibited more extensive core damage due to the high strain rate
detonation event. However, all panels including the sandwich core specimens exhibited
the same external final deformation profiles for both test methods. Thus, an equivalent
lower pressure and longer duration projectile impact could cause the same facesheet
damage modes as an actual blast, but the internal damage to the core materials was
typically more extensive for the loading produced by high explosive detonations.

Compared to the actual blast tests, the non-explosive methodology was able to
achieve a much higher consistency and repeatability between tests. The actual blast tests
were incomplete in many respects: two of the five panels experienced eccentric loading,
three had saturated accelerometer data, none had high speed video captures of the
transmission plate free flight, and most importantly, no panel had confirmation of the
applied impulse. In fact, only one of the five blast tested panels had a complete data set
(accelerometer, centered loading, and symmetric deformation profile). By contrast, over
40 projectile impact tests on both steel and sandwich panels were conducted a year apart
with less than 1.9% standard deviation in applied impulse and less than 1% for the second
phase of 27 tests. Repeatability was such that the deformation profiles for a set of three
panel tests in some instances showed exactly the same post-impact yielding behavior.
Accelerometer data may have been in error for a few of the non-explosive tests, but high
speed video data were intact for all tests such that data was never completely lost for any
panel. Thus, the benefit to researchers requiring highly controlled loading to produce repeatable and consistent data for armor component design is apparent, and until recently, this type of close-in blast test was typically not performed on large components without the use of explosives.

The flexible tiled array projectile that was developed for the non-explosive methodology allows for a repeatable close-in simulated blast loads to investigate dynamic impacts on wide area (610 x 610 mm) armor panels. The non-explosive methodology enables isolating a few of the blast parameters of interest, e.g., the spatially and temporally varying pressure pulse of a close-in charge, and investigating how these parameters affect the target. For example, this could be used to quantify the subtle differences between using a 300 vs. 400 kg/m$^3$ aluminum foam core with respect to transmitted accelerations (5,741 g vs. 5,297 g mean peak acceleration), details which could easily be overlooked in an actual blast test due to an off-axis detonation or variability in the (unmeasured) applied impulse.

Combining this level of detail on the global panel behavior with material studies performed on small scale coupon specimens enabled developing finite element models with very good correlation. This was necessary to virtually examine component behavior under close-in explosive loading as was demonstrated on both conventional steel RHA panels and a sandwich specimen utilizing an aluminum foam core design. Using just the two non-explosive data sets (coupon and large panel), the FE simulations predicted through thickness stress distribution, damage modes, and extent of damage of an ideal blast test. The deformation profiles were modeled with good correlation, even
demonstrating the shortcomings of using a traditional transmission plate approach for measuring the response of a prototype sandwich panel. Additional material parameters such as material strain rate dependency would permit even better validations with the blast results. However, for simulations that were analyzed with commercially available FE software using standard desktop machines, the benefits of the non-explosive method to provide data that could be used to directly validate FE models was invaluable. The fact that the actual data sets could be also repeated within a very small margin was of particular utility for understanding component behavior in a highly variable explosive environment.

It should be mentioned that actual explosive blast testing will always be needed, and that the non-explosive techniques may never truly replace an explosive test on an armor component, especially at the full-structure level. However, in between the conceptual design and the completed structure, the benefits of the non-explosive methods can be realized. The tiled array projectile incorporating both varying mass and variable height tile blocks capable of conforming to a deforming flexible panel surface represents just the beginning in a series of tunable, scalable, and highly repeatable non-explosive pressure pulse generators, each tailored for a specific aspect of the actual detonation event. Future work in this field of non-explosive testing may consider the performance, design, and assessment of armor components through FE simulation (supported by limited testing for validation). These validation tests could be completed with the UCSD Blast Simulator and only the finalized designs would be tested with actual explosives, thereby expediting the development of the armor components.
REFERENCES


[33] L. K. Stewart, "Testing and analysis of structural steel columns subjected to blast loads," University of California, San Diego, La Jolla, Dissertation UMI 3404594,


APPENDICES

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A. Blast Simulator Settings

The UCSD Blast Simulator requires a Mathcad file to check valve open times and nitrogen fill pressures for safe operation. Due to the stroke limitations placed on the BG25 actuator, among other factors, the maximum velocities reported by the worksheet are higher by about 6 m/s compared to the actual projectile velocities for each phase of testing. The parameters listed in Tables A.1 and A.2 are sufficient to replicate the impacts for the projectile, actuator, and reaction block detailed in this manuscript. High speed camera representing the typical usage are also presented. The 2011 tests relied on ambient lighting conditions which limited the exposure and frames per second (fps). The 2012 tests used four 250 W halogen photography lamps which allowed for higher frame rates and lower exposure.
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</tr>
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<td>Impactor mass</td>
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</tr>
<tr>
<td>Impact distance (x = 0)</td>
<td>in 35</td>
</tr>
<tr>
<td>Initial VG position</td>
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</tr>
<tr>
<td>Decel pressure (x = 0)</td>
<td>psi 1,400</td>
</tr>
<tr>
<td>Accumulator N2 pressure</td>
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</tr>
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<td>Oil charge pressure</td>
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</tr>
<tr>
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</tr>
<tr>
<td>Anti cavitation interval</td>
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</tbody>
</table>

| Spread sheet inputs (ref) | |
| Elastomer coef of restitution | 0.5 |
| time span for reference | s 0.15 |
| Guess max pd | psi 800 |

| Results expected | |
| maximum velocity | m/s 27.2 |
| velocity at impact | m/s n/a |
| energy at impact | kJ n/a |
| acceleration before impact | g n/a |
| protrusion into test space | in n/a |
| time to impact | s n/a |
| margin of stroke available | in 20 |
| max oil pressure in actuator | psi 3,312 |
| max decel pressure | psi 5,985 |
| decel pressure at start | psi 1,736 |

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</tr>
<tr>
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<td>-</td>
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Table A. 2. NEM2 Blast Simulator settings, hybrid projectile (2012 test series)

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<td>Impactor mass</td>
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<td>Initial VG position</td>
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<td>Decel pressure (x = 0)</td>
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<td>Fps</td>
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</table>
B. LARGE PANEL DAMAGE AND DEFORMATION PROFILES

Panel test information, internal damage, and deformation profiles are included here for all tests. Tables B.1 to B.3 are the measured quantities. Figures B.1 to B.16 are the sectioned panels ordered in by panel ID (NEM first, then blast tested panels) and Figures B.17 to B.34 are the deformation profiles also ordered by panel numeric ID.

Table B.1. NEM1 applied and transmitted impulses

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<th>Acceleration</th>
<th>Dent depth (mm)</th>
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<td></td>
<td></td>
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<td>Trans. (m/s)</td>
<td>Init.-Avg. (g)</td>
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<td>19.13</td>
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* Excluded from acceleration average
** Method for computing transmission plate velocity, HS camera (CAM) or integration of accelerometer data (ACL)
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* Excluded from acceleration average
** Method for computing transmission plate velocity, HS camera (CAM) or integration of accelerometer data (ACL)
Table B.3. OBLT applied and transmitted impulses

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<th>Trans. Plate Vel (m/s)</th>
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Figure B.1. NEM1 panel 1435.0.3 showing mostly intact core with some cracks and debonding at the center and shear cracks near the clamped boundary
Figure B.2. NEM1 panel 1436.0.2 showing intact core with shear cracks near clamped boundary
Figure B.3. NEM2 panel 1436.1.2 with front face bonded to polymer layer (top) and panel 1436.1.5 with front face debonded from polymer layer (bottom)
Figure B.4. NEM1 panel 1437.0.3 with intact core at panel center and shear effects near clamped boundary
Figure B.5. NEM2 panel 1437.1.2 showing intact core at panel center with some shear effects near clamped boundary
Figure B.6. NEM1 panel 1438.0.2 with intact core at panel center and shear effects near clamped boundary
Figure B.7. NEM2 panel 1568.0.3 showing front face core separation, intact core at panel center, and shear effects near clamped boundary
Figure B.8. NEM2 panel 1569.0.3 showing front face core separation, intact core, and minimal shear effects near clamped boundary
Figure B.9. NEM2 panel 1570.0.3 showing intact core, some facesheet debonding, and minimal shear effects near clamped boundary
Figure B.10. NEM2 panel 1571.0.4 showing some core buckling near panel center and core buckling near clamped boundary
Figure B.11. NEM2 panel 1572.0.3 showing intact core, cell buckling at panel center, and crushed core near clamped boundary
Figure B.12. NEM2 deformation for panel 1574.0.2 showing intact core at panel center and core crush near the clamped boundary
Figure B.13. OBLT panel 1435.0.4 showing extensive core crushing at center and cracks near the boundary
Figure B.14. OBLT panel 1436.1.3 showing extensive core crush, shear cracking near clamped boundary, front face debonding, and front layer charring
Figure B.15. OBLT panel 1437.1.5 with extensive core crush throughout
Figure B.16. OBLT panel 1438.1.2 with extensive core crush throughout
Figure B.17. NEM1 deformation profiles for panel 1435

Figure B.18. NEM1 and NEM2 deformation profiles for panel 1436
Figure B.19. NEM1 and NEM2 deformation profiles for panel 1437

Figure B.20. NEM2 deformation profiles for panel 1438
Figure B.21. NEM1 deformation profiles for panels 1469.0, 1477.0, and 1478.0

Figure B.22. NEM2 deformation profiles for panel 1568
Figure B.23. NEM2 deformation profiles for panel 1569, front only curve for 1569.0.4 due to back face damage

Figure B.24. NEM2 deformation profiles for panel 1570
Figure B.25. NEM2 deformation profiles for panel 1571

Figure B.26. NEM2 deformation profiles for panel 1572
Figure B.27. NEM2 deformation profiles for panel 1574

Figure B.28. NEM2 deformation profiles for ALUM panel
Figure B.29. NEM1 and NEM2 deformation profiles for RHA panel

Figure B.30. NEM1 average deformation profile, OBLT deformation profile for panel 1435
Figure B.31. NEM1 and NEM2 average deformation profiles, OBLT deformation profile for panel 1436

Figure B.32. NEM1 and NEM2 average deformation profiles, OBLT deformation profile for panel 1437, 1437.1.5 only back profile due to front panel damage
Figure B.33. NEM2 average deformation profile, OBLT deformation profile for panel 1438

Figure B.34. NEM1 and NEM2 average deformation profiles, OBLT deformation profile for RHA panel
C. SMALL COUPON SPECIMEN FORCE VS. DISPLACEMENT PROFILES

A summary of the collapsing stress for the MTSQ and MTSF coupon specimens is provided in Table C.1. Force vs. displacement plots are provided for all coupon specimen uniaxial compression tests in Figures C.1 to C.11.

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Figure C.1. MTSQ and MTSF force vs. displacements for specimen 1435

Figure C.2. MTSQ and MTSF force vs. displacements for specimen 1436
Figure C.3. MTSQ and MTSF force vs. displacements for specimen 1437

Figure C.4. MTSQ and MTSF force vs. displacements for specimen 1438
Figure C.5. MTSQ and MTSF force vs. displacements for specimen 1568

Figure C.6. MTSQ and MTSF force vs. displacements for specimen 1569
Figure C.7. MTSQ and MTSF force vs. displacements for specimen 1570

Figure C.8. MTSQ and MTSF force vs. displacements for specimen 1571
Figure C.9. MTSQ and MTSF force vs. displacements for specimen 1572

Figure C.10. MTSQ and MTSF force vs. displacements for specimen 1573
Figure C.11. MTSQ and MTSF force vs. displacements for specimen 1574
D. Hopkinson Bar Aluminum Tipped Projectile Casting

The aluminum tipped foam projectiles were cast individually in a three stage process. The first stage was designed to center the aluminum disk in the projectile, the second stage cast main body of the projectile, and the final stage added the tail fins for flight stability. Total time required was two hours per projectile batch (2 molds available). The resulting projectiles were 125 – 140 g, capable of stable flight as low as 25 m/s and high velocities greater than 160 m/s. A variation of this casting process using a higher density foam and an RHA-reinforced tip produced 285 – 305 g projectiles. The general supplies, dimensions, and finished projectile are shown in Figures D.1 and D.2.

Figure D.1. Projectile casting supplies
The mold requires the stainless steel mold bodies, the stainless steel extension tubes, the precision steel centering rings, the stainless caps, the aluminum o-ring mold components, mold release, and the mold compressor (see Figure D.3). The projectile requires 76.2 mm diameter, 3.175 mm thick aluminum 6061-T6 disks, waterjet cut (not milled), Smooth On brand 5 PCF (80 kg/m³) foam, glue gun, and EastPoint Sports brand shuttlecocks.
To center and cast the aluminum front disk, first spray mold release on the cap and centering rings. Place centering ring inside the cap and the aluminum disk inside the ring (see Figure D.4). Do not spray mold release on the aluminum disk. Spray mold release on the interior wall of the extension tube and fit over the cap. Repeat process for second cap. If casting RHA-backed aluminum tips, first use a hot glue gun to attach RHA square to aluminum disk. Then follow the same process. Place extension tube on top of centering ring (see Figure D.5).
Figure D.4. Inserting centering ring into mold cap (left) and disk inside centering ring (right)

Figure D.5. Fit mold extension tube to cap

Measure 15 mL of Smooth On Foam-It 5 Part A with 15 mL Part B. Combine in a stirring cup and mix thoroughly until a uniform tan color. Pour 15 mL into each open
mold (see Figure D.6). Use stirring rod to push down on aluminum disk inside mold until foam starts to rise. Use a weight or mold compressor to keep parts from moving during the foam curing stage. Wait a minimum of one hour for foam to cure. If using Foam-IT 10, use 30 mL Part A and 30 mL Part B and fill each mold with 30 mL of mixture. After one hour, remove cured foam casting “pucks” from the extension tube (see Figure D.7).

![Figure D.6. Foam mixture divided into two molds](image-url)
Spray mold release to the inside wall of the mold body and place the foam puck inside, aluminum side facing downward (see Figure D.8). Push puck to the very bottom of the mold. Spray mold release onto the inside wall of the extension tube and place over the mold body. Spray mold release liberally on the aluminum o-ring mold components and place inside the extension tube (see Figure D.9 and note the top aluminum hat is omitted for clarity). Make sure that the o-ring mold can easily slide inside the extension tube before tightening the aluminum hat. Fit mold inside mold compressor and tighten screws (see Figure D. 10).
Figure D.8. Inserting foam puck into mold body (left) and puck at bottom of mold body (right)

Figure D.9. Inserting aluminum o-ring mold inside the extension tube; top aluminum hat holding o-ring mold is omitted for visibility
Figure D.10. Mold compressor
Mix 30 mL Part A with 30 mL Part B if using Foam-It 5 and pour the entire amount into one mold tube. Repeat process for second mold tube. If using Foam-It 10, use 60 mL Part A with 60 mL Part B. Pour through fill port in the o-ring mold components (see Figure D.10). Wait a few seconds for the foam to begin rising, then spin entire mold assembly inside the mold compressor so that some foam coats the aluminum mold pieces. This process creates a better casting but must be done when the foam just starts to rise. If attempted too early (when foam is still liquid), the mix will enter the seams and voids in the aluminum mold components and will make removal extremely difficult. Place mold compressor on level surface and allow a minimum of one hour to cure.

Remove mold from the mold compressor. Disassemble the aluminum top plate, then remove the lower mold body. With slight pressure, or even light taps with a rubber mallet, push down on the aluminum o-ring halves (the extension tube moves up, see Figure D.11). Do not push the extension tube down as doing so will destroy the small foam lip created in the gap between the body mold and the extension tube. Once the projectile is free from the tube molds, use a screwdriver to pry apart the aluminum o-ring halves (see Figures D.12 to D.13).
Figure D.11. Removing foam projectile from mold; note the lack of flash on the aluminum o-ring components.
Figure D.12. Foam projectile and aluminum o-ring components removed from mold

Figure D.13. Removing aluminum o-ring components from projectile
Saw off all foam above the o-ring lip. Remove rubber tip from shuttlecock, and place the fins in the center of sawed off surface (see Figure D.14). With a tube and hammer, tap fin into foam to make an indent. Remove fin, fill indent with hot glue, and push fin back into indent. Apply hot glue to the outer fin-foam joint seam. Projectile is now ready to accept outer o-ring seal and be loaded into the gas gun. For better results, allow foam to cure for 24 hours before use.

Figure D.14. Projectile casting progression (from left): aluminum disk, foam puck, cast projectile, completed projectile with stabilizer fins
E. Gas Gun Velocity vs. Pressure Plots

Gas gun projectiles are tailored to provide a specific pressure pulse loading profile. The use of aluminum-faced projectiles, for example, was born out of the necessity to excite high magnitude, short time duration pressure pulses in the intended specimens. Modifying the projectile often affects the projectile mass and subsequently, the required gas gun pressures necessary to achieve a desired velocity. To minimize the number of “blind” trials, especially for new projectile designs, it is useful to reference earlier tests with projectiles of a similar mass. Several pressure vs. velocity plots are presented here along with the projectiles (see Figures E.1 to E.6).
Figure E.2.  Velocity vs. pressure plot for aluminum tipped, 5 pcf projectile, 100 - 130 g

Figure E.3.  Velocity vs. pressure plot for aluminum tipped, RHA backed, 10 pcf, 200 – 360 g projectiles with Mylar option
Figure E.4. Aluminum tipped, 5 pcf, aluminum wasted body projectile; RHA tipped similar

Figure E.5. Velocity vs. pressure plot for aluminum tipped, 5 pcf, aluminum wasted body, 600 – 650 g projectiles
Depending on the velocity, the Hopkinson bar must be dropped to account for the parabolic flight path of the projectile. This is especially of concern at velocities < 80 m/s. At velocities > 80 m/s, the drop is less than 1 mm, which may be neglected. Computing the drop $D$ is based on the velocity $V$ and distance $L$ from the end of the barrel to the target surface, as shown in Equation E.1.

$$D = \frac{gL}{2V^2}$$ (E.1)
The distance from barrel to Hopkinson bar with the shock absorber extension tube is 1.45 m. Without the shock absorber extension, the length is 1.69 m. Drop amounts for both distances are presented in Table E.1.

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F. MATLAB SCRIPT FOR INTERFACING WITH ABAQUS V6.10 EF1

Abaqus/Explicit v6.10-EF1 is used for finite element analysis in conjunction with Matlab version 7.5.0.243 (R2007b) to provide increased functionality including:

i. Batch analysis for running several jobs in sequence without user intervention.

ii. Modifying and rewriting.inp files.

iii. Automatic .odb to .rep file conversions with graphing.

These tasks are accomplished using the B_ABQRW script, designed to interface with Abaqus through command line and the Matlab toolboxes. Details of the theory of operation and the actual code are included in this section.

At its core, Abaqus relies on a plaintext input file (.inp) containing all of the material and simulation parameters. During analysis, results are stored in a large .odb file that can only be opened within Abaqus. The input file is created automatically during the job submission process even when using the CAE graphical user environment. It may also be created without invoking the Abaqus solver engine in the “Job” context menu. In many cases, the input file must be manual edited since certain parameters are not accessible from the CAE environment. Once analysis is complete, the .odb file can be converted to a plaintext report (.rep) file. The report contains only the history outputs designated in the input file and not the full contents of the .odb results.

The process of modifying and input file, running a job, and converting .odb results can be scripted—a feature that is actually included within Abaqus using the
Python programming language. For this project, Matlab R2007b was used instead to create the B_ABQRW function for interacting with the Abaqus command line.

The original environment for the Matlab and Abaqus/Explicit software packages was a 32-bit Microsoft Windows XP desktop machine using standard installation directories which may require some changes for newer installation/computing platforms. The B_ABQRW assumes that the Abaqus default directory is 'c:\temp\'. Update all instances in the script to the actual directory used by Abaqus. If, after submitting a job in Abaqus using either B_ABQRW or command line, an error is shown stating: “abaqus is not recognized as an internal or external command ...” the environment variable needs to be set in the Microsoft Windows operating system. To do this, navigate (in XP) to Start, My Computer (right click, Properties), Advanced tab, Environment Variables. Scroll in the System variables task pane until reaching “Path”, click “Edit”, and add this line (not including quotes and separated from other commands with a semi-colon): "C:\ABAQUS\Commands;" or wherever the Abaqus commands directory is located (see Figure F.1). Press ok and restart the operating system. Also, in the Commands directory, there may exist an "abaqus.bat" file if multiple versions of Abaqus are installed. To use a specific version of the Abaqus engine, edit this .bat file by changing the command line to point to the desired Abaqus version .bat file.
Once Matlab is started and the Abaqus commands are available, the script can be invoked with a supplied filename, e.g. `B_ABQRW('FILE')`. The script searches the working directory (in this example, ‘C:\temp’) for the .inp file and calls the Microsoft Windows XP command line interface from within Matlab to run the Abaqus analysis. If only the filename is supplied, the script will default to single precision (PRSN = 1), no automatic .obd conversion (OBDC = 0), no user defined real time plotting (RLTM = 0), and no .inp editing.
function B_ABQRW(varargin)

CDR=cd; cd('c:\temp\');
A=varargin; [RW,CL]=size(A); PRSN=1; OBDC=0; RLTM=0;
if isempty(A); disp('Need input arg. Type help B_ABQRW'); cd(CDR);
    return; end;
%% If single cell varargin, calls abaqus in cmd
if CL==1; tic; ABQCMD(FILE,PRSN,OBDC,RLTM); toc;
%% OPTIONAL DELETE _N FILE %%%%%%%%%%%%%%%%%%%%%%%%%%
  fprintf('Deleting %s input file\n',FILE); delete([FILE '.inp']);
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
cd(CDR); end

Calling the function with a cell format that includes more variables, e.g.
B_ABQRW({'FILE','PRSN','OBDC','RLTM'}) enables more advanced settings
with the same .inp file job submission. Replace PRSN with 2 sets double precision,
OBDC = 1 turns on automatic .rep file if history outputs are defined within the .inp file,
and RLTM = 1 turns on real-time monitoring/graphing of a user defined parameter within
the history output. For most job submissions, real time monitoring is best turned off
(RLTM = 0) which enables verbose logging within the Matlab command window.
Otherwise, set RLTM = 1 and use as few history outputs as possible for faster plotting.

%%% Copies values from varargin, and assigns depending if varargin is cell
if iscell(A{1}); [RC,CC]=size(A{1});
  if CC<4; disp('Need more input parameters. Type help B_ABQRW');
    return; cd(CDR); end;
    FILE=A{1}{1}; PRSN=A{1}{2}; OBDC=A{1}{3}; RLTM=A{1}{4};
  else FILE=A{1}; end;

To turn on .inp file modification, call B_ABQRW(FI,'run',[S1
Vi],…,[Sn Vk]), where FI is created from either filename instructions previously
mentioned, S1 to Sn are the search values in order of appearance within the .inp file, and
Vi to Vk are the appended lines to write immediately after finding S1 to Sn, respectively. The appended lines can be of any length and any number in format \( V_i = [\text{'V1'} \text{'V2'} \text{'V3'} \ldots \text{'Vn'}] \). A new .inp is created with “_N” appended, which can be turned off such that the original .inp file is overwritten. The function then calls itself with the modified .inp file and the pre-defined parameters to run the Abaqus solver.

To stop a job, first stop the script in Matlab by pressing “Ctrl + C”. Then in the command window, type \texttt{B\_ABQRW(FILE,'stop')}, where \texttt{FILE} is the name of the .inp job, which may include “_N” if a search and append operation was performed. A sample job submission is shown in Figure F.2 using verbose logging (RLTM = 0).

```plaintext
%% Finds, appends lines to input file if varargin has 3+ variables
if CL>=3; tic;
  EXT=['.inp']; IN=[FILE EXT]; OT=[FI EXT];
  fid=fopen(IN,'r'); pid=fopen(OT,'w');
  for i=3:CL+1
    if i<=CL; S=A{i}(1); V=A{i}(2:end); L=length(V);
      else S={'eof'}; end;
    fprintf('Writing file. '); R=filewrtr(fid,pid,S{:});
    if R==1; fprintf('Adding strings. '); for i=1:L; fprintf(pid,'%s
',V{i}); end;end;
    if R==0; break;end;end;
  fclose(fid); fclose(pid); fprintf('Done.
'); toc;

%% Replace original file, delete _N file. %%%%%% FILE CMT OPT %%%%%
% fprintf('Replacing original file:\n');
% copyfile(OT,IN); delete(OT); FI=FILE;
% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%% FILE CMT OPT %%%%%
% fprintf('Replacing original file:\n');
% % CALL Matlab-ABAQUS interface %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% B\_ABQRW({FI,PRSN,OBDC,RLTM}); cd(CDR); end;

%% Kills job %%%
if CL==2
  fprintf('Stop command issued for %s
',FILE);
  dos(['abaqus j=' FILE ' terminate']); cd(CDR); end;
end
```
The subfunction \texttt{ABQCMD(FILE,P,B,R)} is the actual command line calling script. Under normal usage, the script should not need any modifications. However, for long \texttt{.inp} files (>1e7 lines), the \texttt{maxit} variable in the \texttt{filewrtr} subroutine should be changed. By default, the main subfunction writes to the Matlab command window the status of the Abaqus analysis including the current time step. The RLTM monitoring parameter, on the other hand, checks for the existence of a lockfile (which Abaqus uses only during analysis), and if found, runs the \texttt{.obd} converter, captures the history output value, and graphs on screen. This process repeats every 30 seconds until the lockfile is no longer present (analysis completed) and for large history outputs, consumes
computation resources. Thus, under normal usage, real-time monitoring is not advisable as it slows the analysis process.

A sample job completion screen is shown in Figure F.3 with verbose logging provided from the Abaqus job submission process. This is the same information available by monitoring an analysis within CAE.
Figure F.3. Abaqus status and job completion through Matlab
G. Calculating Dynamic Stress-Strain Profile Using High Speed Camera

Creating the stress-strain profiles for dynamic strain rates using the high speed camera requires several steps. First, the camera must be positioned perpendicular to the target, with ample lighting, such that the highest possible frames per second (fps) and the lowest exposures possible are captured prior to analysis (see Figures G.1 and G.2). Poor quality video or using a low frame rate cannot be corrected and thus, strain measurements are not possible.

Second, a known reference length must be visible at some time frame within captured video if distances or velocity is required (strain calculations can be done using pixels). This should be a precision machined part with measureable distances that do not change during the dynamic impact event. The measurement must be also in the same vertical plane as specimen. Using a reference point that is physically located a few meters behind the target may not provide accurate results. Also, the specimen for strain calculations should be uniformly crushing within the desired region of interest. Otherwise dividing the change in length by the original length will not be an accurate representation of strain.

Finally, the process requires considerable patience. A single stress-strain curve may take up to several hours to complete depending on the frame rate and amount of strain.
Figure G.1. Example of poor high speed video capture for strain analysis

- Non-perpendicular perspective
- Wasted horizontal and vertical resolution
- Non-uniform crushing
- Deformable reference length
- Dim lighting

Figure G.2. Example of good high speed video capture for strain analysis

- Optimized horizontal and vertical resolution
- Perpendicular capture
- Good lighting
- Fixed reference lengths
- Uniform crushing

Figure G.2. Example of good high speed video capture for strain analysis
Assuming good quality high speed video has been captured, measure the reference points to set the scaling factor. This step is required only if displacements or velocities are required. Open the .cine file in the Phantom software and click on each end of a known length (see Figure G.3). The x/y coordinates are displayed for position 1 (e.g. [94, -34]) and also at position 2 (e.g. [137, -34]). For the example presented in this section the projectile flight (and subsequently, the specimen crush) is in the x direction only. Subtracting the two pixel positions (137 – 94) yields a distance of 43 pixels. The component is exactly 5.08 mm so the reference length is 5.08 / 43 = 0.118 mm/pixel. Several reference points at different times and using different references should be taken, if possible, and averaged to provide a more accurate reference length.

Next, record the length of the target at every time frame that the specimen experiences a change in length (per the example, x direction only). This requires two position references measured at the front and back of the specimen along with the time stamp (see Figure G.4). The high speed video is then advanced by one frame and the next two position measurements are recorded. This process is repeated until the desired strain level is achieved. In some instances, direct measurement of the front or back surface is not possible due to specimen deformation obscuring the loading surface or other objects entering the frame of view. During these instances, it may be possible to use a reference on a hard, non-deforming surface at a known distance away from the front or back face of the specimen. Monitoring this reference point and then adding in the distance to the respective loading surface can recover the specimen deformations (see Figure G.5).
Figure G.3. Finding pixel position of known reference

Figure G.4. Length measurement at time $t = 283 \, \mu s$
Subtracting the front and back face positions at every time point from each other provides the pixel length of the specimen as a function of time, which can be converted into strain as a function of time by Equation G.1.

\[
\varepsilon(t_i) = \frac{L_0 - |x_1(t_i) - x_2(t_i)|}{L_0}
\]  \hspace{1cm} (G.1)
Here, $t_i$ is the time point, $x_1$ is the front face position, $x_2$ is the back face position, and $L_0$ is the initial length of the specimen, in consistent units. Notice that the strain measurement itself does not require the pixel length scaling, however, for measuring strain rate and velocity of impact, the pixel length scaling is critical.

With the discrete strain as a function of time established, numerical operations must be performed to match the force time history increments, which can be several orders of magnitude smaller. For example, capturing 1 ms of force data at 1 MHz equates to 1,000 discrete data points at 1 µs intervals. Setting the high speed camera to 100,000 fps means that 100 data points are captured in the same 1 ms time period at 10 µs intervals. Since there are ten times less strain data points, the only options to synchronize the two vectors are to either downsample the force data or upsample the strain data. Upsampling is the preferred method here to preserve as much original data as possible. The process of upsampling the strain data is fairly routine:

i. Curve fit the strain data with a higher order polynomial (see Figure G.6).

ii. Sample the fitted-curve at the same time intervals as the force data.

iii. Offset both force and strain time histories to start at a common time point.

Once both curves are on the same time scale and at the same time offset, then the two quantities can be plotted against the other. Force is divided by the cross-sectional dimensions of a pristine specimen to recover the stress and plotted against strain to create the stress-strain plot (see Figures G.7 and G.8).
Figure G.6. Up-sampled strain time history with discrete data points for 38.1 mm diameter foam

Figure G.7. Stress time history for 38.1 mm diameter foam
Since strain is not captured directly, the sampling method is not always representative of the real strain in the material. This can be minimized by visually confirming that the specimen is experiencing a uniform crush (which is facilitated by drawing straight lines on the specimen and monitoring them) and by having as large of a strain data set as possible. Achieving the high strain data set requires maximizing the number of pixels in the direction of the strain measurements, high frame rates, and low exposures, all of which involve a compromise to achieve best results. For example, good quality high frame rates are only possible with lower resolutions and exposures. Low exposures may demand significant lighting (limiting frame rate if not achieved) while the region of interest may dictate larger resolutions. Assuming an ideal configuration of the 80 – 200 mm Nikon zoom lens at the minimum focus distance of 1.3 m, 100,000 fps and
320 pixels in the deformation direction, a sample can be up to 40 mm in length with a reference length of 0.12 mm/pixel. An error of just one pixel in measuring deformation can cause 0.25 % error in the calculated strain.

In addition to the strain errors, identifying the time offset to the start of the force and displacement loading is based on visual observation as there is currently no common trigger between the high speed camera and the force measurements. Since the stress-strain plot is based on both force and displacement time histories, correct time offset is necessary to ensure that the force at one time point corresponds exactly with the displacement at the same time point. In this case, an error of just 10 µs (one frame of the high speed camera) is sufficient to shift the entire stress-strain plot by 1% (assuming 1 ms of force measurement data). For some materials, the entire elastic loading regime is less than 1%, so an error in time shift or an error in strain can completely change the results.
H. Dynamic High Strain Test Overview of a Modified Hopkinson Bar Setup

Validating finite element simulations or generating material stress-strain plots at high strain rates and high strains presents several challenges. For coupon-sized specimens on the order of 100 mm, dynamic testing is often accomplished with a gas gun and either a split Hopkinson bar setup (SHPB) or a projectile and Hopkinson force transmission bar. The benefit of the SHPB is the parallel loading surfaces but at the expense of high impact velocities. By contrast, a projectile launched via gas gun and impacting a Hopkinson force transmission bar can achieve impact velocities > 100 m/s and even > 1,000 m/s. However, achieving good quality dynamic impacts, i.e., high parallelity of the loading surfaces and uniform crushing in the specimen, requires careful tuning, pre-test projectile flight tests, and in some cases, repeated tests.

To address these issues, a Hopkinson bar was modified by mounting an intermediate shaft in front of the strike surface (see Figure H.1). The projectile would first contact this intermediate shaft, which would then transfer the momentum to crush the test specimen. This configuration is similar in operation to the incident and transmission bar of a split Hopkinson pressure bar. An optional contact switch could be installed to activate a light source visible in the high speed camera when the projectile impacted the transmission shaft.
Figure H.1. Modified Hopkinson bar setup for dynamic high strain testing

With this modification, the projectile served only as a source of energy thereby eliminating the requirements of achieving nearly perfect free flight since the intermediate shaft would provide a plane and parallel loading surface with respect to the Hopkinson bar. A basic momentum calculation using the mass and velocity of the projectile as shown in Equation H.1 can be used to estimate the shaft velocity.

\[ V_{SH} = \frac{M_{SH}}{M_{SH} + M_{PRJ}} V_{PRJ} \]  \hspace{1cm} (H.1)

Based on Equation H.1, it is desirable to use a high mass for the projectile and low mass for the intermediate shaft in order to extract the highest strains and strain rates possible.
The shaft also appeared to produce uniform crushing in low impedance materials such as polyurethane foam (see Figure H.2).

Figure H.2. High speed video capture of projectile impacting intermediate shaft and crushing prismatic 18.8 x 18.8 mm foam specimens
Test specimens were initially limited to the diameter of the intermediate shaft, which was 25.4 mm based upon available components. A temporary solution for testing larger diameter specimens was to use another component to spread the load from the shaft over a much wider diameter (see Figure H.3).

Figure H.3. High speed capture of projectile impact on 38.1 mm diameter foam specimen
For high energy impacts on more robust materials such as metal foams, the lightweight aluminum intermediate shaft was often deformed at the projectile strike surface which required frequent machining that delayed testing. This occurred despite using 7075 high strength aluminum for the intermediate shaft. A solution was found by stacking low strength aluminum washers in between a wide loading disk and the actual intermediate shaft (see Figure H.4). The loading disk prevented the foam body projectiles from deforming around the 25.4 mm diameter shaft, while the aluminum washers absorbed some of the high frequency impact and deformed to prevent damage in the shaft. The aluminum washers were discarded after each test.

![Figure H.4. High speed capture of projectile impact on 38.1 mm diameter foam specimen](image)

This test setup has been used to excite strain rates in 25.4 mm diameter metal foam specimens up to 42% strain at 2,000 1/s and 83% strain at 1,700 1/s.
I. MACHINE DRAWINGS

BG LOADING SIDE COMPONENTS OVERVIEW

1.0 PUSHER PLATE SYSTEM
  1.1 FRONT PLATE
  1.2 TUBING
  1.3 BACK PLATE
  1.4 BG ADAPTER

1.5 SUPPORT SHAFT

2.0 PROJECTILE SYSTEM ALL STEEL

3.0 PROJECTILE SYSTEM HYBRID

2.1 TILE SHEET

3.1

2.2-2.3 TILE BLOCKS

3.2-3.5

2.4 JIG FOR CUTTING FOAM
2" CLEAR FOR 3/4-20 FREE FIT
USE DRILL SIZE H

1 SET = 4X PIECES

2x4x1/8 W.T. x16" TUBING

16.0C

3.250

3.250

1.500

2.0C

322
1.3 BACK PLATE

DRAW NO. RAM_SYS_DRAW_v1

MATERIAL AL 6061

TOLERANCES: PLUS OR MINUS UNLESS OTHERWISE NOTED

QUANTITY

NOTES

SCALE 1/4

SHEET 4 OF 6
DO NOT MAKE. WE WILL SUPPLY SHAFT.
3.2 STEEL TILE BLOCK

QTY DEPENDS ON DIMENSION Y:
Y = 1.00  QTY = 12
Y = 1.50  QTY = 36
Y = 1.75  QTY = 24
TOTAL QTY = 72

NOTE: TAP FROM OPPOSITE FACES

1 x 1/4-20 TAP
Ψ 1/2" OK TO BREAK FACE FOR EASE

2 x 1/4-20 TAP
Ψ 1.0" OK TO BREAK FACE FOR EASE

MATERIAL: STEEL

TOLERANCES: PLUS Otherwise noted

QUALITY:

SEE NOTE

YY3 BAR
STOCK OK
ROUND EDGES AND CORNERS

SCALE:
SHEET 3 OF 4
TITLE: 3.3 ALUM TILE BLOCK

DRAW NO: FLYER_SYS_DRAW_v3

MATERIAL: AL 6061

TOLERANCES: ± less otherwise noted

QTY DEPENDS ON DIMENSION Y:

\[ y = 1.00 \quad QTY = 12 \]
\[ y = 0.50 \quad QTY = 36 \]
\[ y = 0.25 \quad QTY = 24 \]

TOTAL QTY = 72

NOTE: SEE 3.3 BAR STOCK OK ROUND EDGES AND CORNERS
REACTION SIDE COMPONENTS OVERVIEW

4.0 PANEL MOUNTING
4.1 FRONT WINDOW FIXTURE
4.2 REAR WINDOW FIXTURE
4.3 TUBULAR SUPPORT

5.0 TOP/BOTTOM SUPPORTS
6.0 SIDE SUPPORTS

7.0 TRANSMISSION PLATE COMPONENTS
8.0 TRANSMISSION PLATE SPLIT RAIL

7.1-7.3 TRANS SHAFT SYSTEM
8.1-8.3 SPLIT RAIL SYSTEM

7.4 TRANSMISSION PLATE
7.5 LOCATING PLATE
4.1 FRONT WINDOW FIXTURE

TO ERANCES:
X.XXX = 0.002
X.XX = 0.02
X.X = ±0.2

MATERIAL: 4130 CHROMOLY
QTY: 1
5.0 TOP/BOTTOM SUPPORTS OVERVIEW

MATERIAL: STEEL
QTY: 2
5.3 SUPPORT TUBE

MATERIAL: STEEL RECT. TUBE
3 X 2 X 0.25W IN.

QTY: 4
DO NOT MAKE
WE WILL SUPPLY
EXISTING FRAME
3 X 2 X W.T. 1/4"

4X CLEAR HOLES
THROUGH BOTH WALLS
BOTH SIDES

1.000

4X CLEAR
FOR 3/8-16 CLOSE FIT
USE DRILL SIZE W

2.000

2.25

4.3 FRAME MACHINE WORK

MOUNT_REIF_V3

MATERIAL: STEEL

QUANTITY: X.XXX ≥ 0.002
           X.XX ≥ 0.02
           X.X  ≥ 0.2

TOLERANCES (UNLESS OTHERWISE NOTED:)

SCALE 1:1

SHEET 7 OF 7
WE WILL SUPPLY SHAFT
φ 1" X 9" MCMASTER
# 6649k73

MATL:
51200 STEEL CASE HARD
ROCKWELL C60-C64

LATHE THIS END 0.250"
(NOT TAPPED END)

0.250 +0.005
-0.000

TAPPED END

WE WILL SUPPLY SHAFT
φ 1" X 9" MCMASTER
# 6649k73

MATL:
51200 STEEL CASE HARD
ROCKWELL C60-C64

LATHE THIS END 0.250"
(NOT TAPPED END)

0.250 +0.005
-0.000

TAPPED END
7.4 TRANSMISSION PLATE

TRANSACTION DRAW

AL 6061

MATERIAL: AL 6061

TOOLTOLERANCES, UNLESS OTHERWISE NOTED:

X.XXX ± 0.002
X.X ± 0.02
X.X ± 0.2

QUANTITY: 3

NOTES:

SHEET # OF 3

SCALE 1:2

SECTION C-C

PREPARE FLAT SURFACE

MIN Ø0.375 WITH FINISH

OF OR BETTER FLAT TO 0.001 TIR

DETAIL D

SCALE 1:2

TAP M6x1.0

DEPTH 0.25” BREAK SURFACE

BOLT CIRCLE

2 3/8”

4X TAP FOR 3/8-16

1.625

1.46 X 0.07

2X TAP FOR 3/8-16 TO DEPTH 2.0"
2X FREE FIT FOR 3/8-16 USE DRILL SIZE X

1X TAP FOR 3/8-16 MIN 1.0" OK TO BREAK SURFACE

8.1 TRANS GUIDE RAIL

TRANS_GUIDE_ASM_V3

MATERIAL: AL 6061

QUANTITY: 12

NOTES:
BREAK EDGES AND CORNERS
USE 1X2.5 BAR STOCK

TOLERANCES: UNLESS OTHERWISE NOTED:
X.XXX ± 0.002
X.XX ± 0.02
X.X ± 0.2
\( 1.000 \pm 0.002 \)

1X TAP FOR 3/8-16 THRU ALL

0.45

8.3 TRANS GUIDE PLUG

TRANS_GUIDE_ASM_V3

MATERIAL: STEEL

TOLERANCES: +/- 0.002

QUANTITY: 2

NOTES: BREAK EDGES AND CORNERS
EXISTING Ø 1.5 HOLE
WJ LARGER TO Ø 4.0

DO NOT MAKE
WE WILL SUPPLY
EXISTING PLATE
24 X 30 X 0.5"

TITLE: 7.4 LOCATING PLATE
        MACHINE WORK

DRAWN BY:
TRANS_GUIDE_ASM_V3

MATERIAL:
AL 6061

QUANTITY:
1

NOTES:
WJ OK

TOLERANCES UNLESS OTHERWISE NOTED:
X.XXX ± 0.002
X.XX ± 0.02
X.X ± 0.2

SCALE 1:8
SHEET 5 OF 5