1-1-2004

Energy-absorbing sandwich structures under blast loading

Dong Kwan Lee
University of Nevada, Las Vegas

Follow this and additional works at: https://digitalscholarship.unlv.edu/rtds

Repository Citation
https://digitalscholarship.unlv.edu/rtds/1687

This Thesis is brought to you for free and open access by Digital Scholarship@UNLV. It has been accepted for inclusion in UNLV Retrospective Theses & Dissertations by an authorized administrator of Digital Scholarship@UNLV. For more information, please contact digitalscholarship@unlv.edu.
ENERGY ABSORBING SANDWICH STRUCTURES
UNDER BLAST LOADING

By

Dong Kwan Lee
Bachelor of Science in Mechanical Engineering
Indiana Institute of Technology, Fort Wayne, Indiana
May 2002

A thesis submitted in partial fulfillment
of the requirements for the

Master of Science Degree in Mechanical Engineering
Department of Mechanical Engineering
Howard R. Hughes College of Engineering

Graduate College
University of Nevada, Las Vegas
August 2004
INFORMATION TO USERS

The quality of this reproduction is dependent upon the quality of the copy submitted. Broken or indistinct print, colored or poor quality illustrations and photographs, print bleed-through, substandard margins, and improper alignment can adversely affect reproduction.

In the unlikely event that the author did not send a complete manuscript and there are missing pages, these will be noted. Also, if unauthorized copyright material had to be removed, a note will indicate the deletion.

UMI®

UMI Microform 1422870
Copyright 2005 by ProQuest Information and Learning Company.
All rights reserved. This microform edition is protected against unauthorized copying under Title 17, United States Code.

ProQuest Information and Learning Company
300 North Zeeb Road
P.O. Box 1346
Ann Arbor, MI 48106-1346

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
Thesis Approval
The Graduate College
University of Nevada, Las Vegas

June 11, 2004

The Thesis prepared by
Dong Kwan Lee

Entitled
Energy Absorbing Sandwich Structures Under Blast Loading.

is approved in partial fulfillment of the requirements for the degree of
Master of Science in Mechanical Engineering

Examination Committee Chair

Dean of the Graduate College

Examination Committee Member

Graduate College Faculty Representative
ABSTRACT

Energy Absorbing Sandwich Structures Under Blast Loading

by

Dong Kwan Lee

Dr. Brendan J. O'Toole, Examination Committee Chair
Associate Professor of Mechanical Engineering
University of Nevada, Las Vegas

An experimental study at the Army Research Laboratories shows that flat panels with foam or honeycomb faceplates transferred more energy to a structure under blast loading relative to a structure without an energy absorbing faceplate. The objective of this work is to simulate non-uniform response of sandwich panels subject to blast loading. This involves an investigation into the optimum design of a square-celled sandwich structure for energy absorption. Variables under investigation are the core and face sheet thicknesses of the sandwich structure. Results of a design of experiments study are attained, which evaluate the relative contribution of panel variables to energy absorption. Also, the results of an optimization study are discussed along with some of the problems faced during this study. The Armor Personnel Carrier vehicle is then modeled to compare the damages on the vehicle with and without the optimized sandwich structure.
TABLE OF CONTENTS

ABSTRACT .................................................................................................................................. iii

LIST OF FIGURES ........................................................................................................................ vi

LIST OF TABLES ........................................................................................................................ vii

ACKNOWLEDGEMENTS ............................................................................................................. viii

CHAPTER 1 INTRODUCTION .......................................................................................... 1
  1.1 Motivation and Objectives ................................................................................................ 2
    1.1.1 Study of Armoured Personnel Carrier (APC) Vehicle Model .............................. 4
  1.2 Energy absorption and plastic deformation characteristics of sandwich structure... 5
  1.3 Literature Review ............................................................................................................... 7
    1.3.1 Energy Absorption Characteristics of Structures .................................................... 8
    1.3.2 Explosive Blast ........................................................................................................... 10
      1.3.2.1 CONWEP ........................................................................................................... 12

CHAPTER 2 MODEL DESCRIPTION ................................................................................... 13
  2.1 Detail Description of Sandwich Model .......................................................................... 13
    2.2.1 Finite Element Model of Sandwich Structure ........................................................ 16
    2.2.2 CONWEP Blast Load Function .............................................................................. 19
    2.2.3 Design of Experiment (DOE) Study ....................................................................... 20
    2.2.4 Optimization Study .................................................................................................... 21
  2.2 Armoured Personnel Carrier (APC) Vehicle Model ................................................... 22
    2.2.1 Finite Element Model of APC Vehicle ................................................................... 24
      2.2.1.1 CASE I: Only APC Model Without Energy Absorbing Material ................ 24
      2.2.1.2 CASE II: APC Model with Flat Plate Attached at the Bottom ................... 26
      2.2.1.3 CASE III: APC Model with Sandwich Structure Attached at the Bottom 27
  2.3 Computing System and Software ................................................................................... 29
    2.3.1 Computing system configuration ............................................................................ 29

CHAPTER 3 RESULTS AND DISCUSSIONS ..................................................................... 30
  3.1 Deformation History of Sandwich Structure ................................................................. 30
  3.2 Design of Experiment (DOE) Result .............................................................................. 30
  3.3 Optimization Result .......................................................................................................... 32
    3.3.1 Dishing Effects ........................................................................................................... 35
    3.3.2 Uniform Pressure ....................................................................................................... 36
3.3.3 Peak Acceleration of Sandwich Structure ...........................................................38
3.4 Comparison of Three Cases in APC Model.................................................................39
  3.4.1 Results of 517.9-g of TNT and 26.13-cm of Ground Clearance (SET1) ..........40
  3.4.2 Results of 6,840-g of TNT and 40.60-cm of Ground Clearance (SET2) ........46

CHAPTER 4 COMPARISON OF LS-DYNA RESULTS TO EXPERIMENT ............48
  4.1 Parameters of Ballistic Pendulum Model.................................................................48
  4.2 Comparison of Results at the Center of Gravity Point ...........................................50

CHAPTER 5 PRELIMINARY ANALYSIS OF OTHER VARIABLES .......................52
  5.1 Number of Cells.........................................................................................................52
  5.2 Core Height .............................................................................................................53
  5.3 Additional Horizontal Layer(s) in the Core ............................................................53
  5.4 Vary the Material Properties ..................................................................................55
  5.5 Pre-Specified Dent of the Core ...............................................................................55
  5.6 Summary ..................................................................................................................56

CHAPTER 6 CONCLUSIONS AND RECOMMENDATIONS ..................................57
  6.1 Conclusions ..............................................................................................................57
  6.2 Recommendations ....................................................................................................58

APPENDIX A. VARIOUS CALCULATIONS..............................................................60
APPENDIX B. THE PEAK ACCELERATION COMPARISON OF NODES
OUTPUTTED IN VICINITY FROM THE CENTER NODE FOR
THREE CASES (SET1 & SET2).................................................................................66
APPENDIX C. SERIES OF DEFORMATION HISTORY: MAXIMUM
DISPLACEMENT AT CENTER NODE FOR ALL THREE CASES
(SET1 & SET2)........................................................................................................66
APPENDIX D. BOUNDARY CONDITIONS FOR SANDWICH STRUCTURE AND
APC VEHICLE WITH SANDWICH STRUCTURE ..............................................71
APPENDIX E. DESIGN OF EXPERIMENT RESULTS – ANALYSIS OF VARIANCE
(ANOVA) OF EACH RESPONSE ..........................................................................81
APPENDIX F. COMPARISON OF RESPONSES FOR UN-DENTED VERSES
DENTED SANDWICH CORE WITH 24 ELEMENTS PER CELL
EDGE......................................................................................................................84
APPENDIX G. ACCURACY OF RESULT COMPARISON WITH REFINEMENT OF
ELEMENTS ON VEHICLE MODEL.................................................................85
APPENDIX H. PEAK ACCELERATION COMPARISON AT DIFFERENT TIME
INTERVAL FOR CASE I & III..............................................................................86

REFERENCES............................................................................................................87

VITA..........................................................................................................................90
LIST OF FIGURES

Figure 1  Schematic Arrangement for Ballistic Pendulum Experiment ............................3
Figure 2  Illustration of Dishing Effect-Confines Pressure .................................................3
Figure 3  APC M113 Model ...................................................................................................5
Figure 4  Typical Blast Pressure Profile ...............................................................................11
Figure 5  (a) Configuration of Square-cell Sandwich Structure, (b) 1/4 Section Model ..........14
Figure 6  Finite Element Model of 1/4 section Sandwich Structure Model ...........................17
Figure 7  Convergence of Internal Energy with Refinement of Mesh Density ......................18
Figure 8  Geometry of APC M113 vehicle with modified dimensions (Unit: cm) .................24
Figure 9  Fully Meshed APC Model Divided into Top and Bottom Components .................25
Figure 10 APC Model with Flat Plate Attached at the Bottom .............................................27
Figure 11 APC Model with Sandwich Structure Attached at the Bottom ............................28
Figure 12 Predicted Deformation Histories for Sandwich Under Blast Load .......................31
Figure 13 DOE Result of the Internal Energy .....................................................................32
Figure 14 Total Energy Distributions For Iteration 1 of the Optimization Study ..................34
Figure 15 Illustration of Global and Local Dishing Effects .................................................35
Figure 16 Fully Deformed Shape of Sandwich Structure Under Uniform Pressure ............36
Figure 17 Comparison of Peak Acceleration of Sandwich Structure vs. Plain-Plate ............38
Figure 18 Selected Nodes from the Center for Peak Acceleration Comparison ....................40
Figure 19 Peak Accelerations of Nodes for Three Cases (SET1) .........................................41
Figure 20 Outputted Nodes to Find the Highest Peak Acceleration ..................................42
Figure 21 Peak Acceleration of All Nodes Inputted into a Single Graph for Case I .............43
Figure 22 Ten Nodes that Experience Higher Peak Acceleration than the Highest Peak Acceleration of Case III ...............................................................43
Figure 23 Peak Acceleration of All Nodes Inputted into a Single Graph for Case III .........44
Figure 24 The Highest Peak Acceleration in Case III ..........................................................44
Figure 25 Peak Accelerations of Nodes for Three Cases (SET2) .........................................46
Figure 26 Schematic Drawings of Ballistic Pendulum Model for ARL and LS-DYNA ...........49
Figure 27 Resultant Displacement of Ballistic Pendulum from LS-DYNA .........................51
Figure 28 Vary the Number of Cells – Doubled .................................................................53
Figure 29 Vary the Core Height ............................................................................................54
Figure 30 Additional Horizontal Layers in the Core .............................................................54
Figure 31 Pre-Dented Core with Dimensions .....................................................................56
LIST OF TABLES

Table 1  Accuracy of Results with CPU Time Cost............................................................. 17
Table 2  Material Properties for Sandwich Structure ........................................................... 19
Table 3  Material Property Section in LS-DYNA Input File .............................................. 19
Table 4  Apply CONWEP Blast Load in LSDYNA Input File for Shell Model ............. 20
Table 5  Size Design Variables with Initial Value and Bounds (unit: centimeters) ...... 21
Table 6  Material Properties of APC Model......................................................................... 26
Table 7  History of Optimization Iteration ........................................................................ 33
Table 8  Optimized Design Variable Values (unit: centimeters)...................................... 33
Table 9  History of Optimization Study with Uniform Pressure .................................... 37
Table 10 Percent Differences of Peak Acceleration for Three Cases (SET1)............... 45
ACKNOWLEDGEMENTS

I wish to express my sincere appreciation to Dr. Brendan J. O'Toole for his support, guidance, and patience in order to finish the thesis. Dr. O'Toole has provided an opportunity to publish a paper and present the work at the 8th International LS-DYNA Users Conference. It was an amazing experience and I appreciate for his encouragement and guidance to do so. I also wish to express my appreciation to Trevor Wilcox for his help using LSDYNA and HyperMesh software and fixing technical problems with my computer.

I would like to extend my appreciation to the committee members, Dr. Mohamed Trabia, Dr. Woosoon Yim, and Dr. Samaan Ladkany for their willingness to be my defense committee.

Finally, the most of appreciation goes to my wife, Joyce, for her support and patience through past two years that made possible to complete my study.
CHAPTER 1

INTRODUCTION

The sandwich structure can provide two major key functions throughout the industry applications. It can be designed to sustain severe load applied to the structure but also can be designed to absorb energies from the load by its plastic deformation, both with outstanding weight saving advantages. For example, the first landing of the Apollo on the moon on July 20, 1969 was the major achieve by the advancement of sandwich technology. It was feasible to construct the Apollo capsule with the help of sandwich technology that was light in weight and yet strong enough to sustain the stresses of acceleration and landing [1]. Conversely, automotive industries design the sandwich structures to absorb impact energy by its plastic deformation at the time of crash to transfer minimum energy to the passenger for both safety and weight reduction purposes. Numerous studies of energy absorbing characteristics with sandwich structure have been carried out intensely in the past few years from automobile industries, railways, and aerospace vehicles under the loading condition of, typically by impact. However, literature on sandwich structures under blast loading condition is somewhat limited indicating that more studies in such area are necessary. The significance of this study is to provide the protection to the army personnel from injuries as well as to build a robust and yet lightweight-armoured vehicle to increase the overall chance of success of the
mission. The Center for Defense Information\(^a\) stated that the landmines were responsible for about 35 percent of all U.S. casualties in Vietnam War and 20 percent of U.S. casualties during the Gulf War. Dealing with landmines during the war is an important issue to be faced. Hence, study of energy absorbing structure under land mine blast is essential especially for Armoured Personnel Carrier (APC) to be enhanced to become anti-vehicular blast mines. It is proven in automotive industry that sandwich structure can be effectively used as an energy absorbing material under impact loading. This structure is also necessary to be investigated under blast load so that it will be appropriately used as the shock-mitigating device as well.

1.1 Motivation and Objectives

The U.S. Army Research Laboratories (ARL) [2] and Hanssen et. al. [3] have experimentally investigated the effects of panel geometry and core material properties on the dynamic response of ballistic pendulums to blast loads. As shown in Figure 1, energy absorbing material (protection concept) is placed on the face of the pendulum and 1.0-lb of C4 charge is located at the standoff distance of 26.13-cm. The pendulum displacement is measured after the detonation to calculate the amount of energy being transferred to the supported structure. These displacement results are compared with the results of a base line flat rigid panel without energy absorbing material on the blast face. Unpredictably, the flat foam and honeycomb-faced panels transmitted more energy to the pendulum than the base line. As shown in Figure 2, Hanssen et. al. [3] have explained that the this phenomenon may be due to the non-uniform deformation (dishing) of the front face,

\(^a\) Center for Defense Information: http://www.cdi.org/terrorism/afghanistan-challenges.cfm

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
which may increase the overall pressure loading on the panel from the blast since it confines the blast pressure.

Figure 1 Schematic Arrangement for Ballistic Pendulum Experiment

In this way, dishing effects could be controlling the energy transfer to the supported structure. There were some variations in panel response depending on the type of foam or honeycomb used and it is not clear what the optimum material properties should be.

Figure 2 Illustration of Dishing Effect-Confines Pressure
Customized sandwich panels can also be designed with truss-like rods, vertical walls, or angled walls as the core structure. Can the sandwich structure be tailored to minimize the damages to the main structure from a blast load? The main objective involves an investigation into the optimum design of a square-cell shaped core subject to blast loading. The variables under investigation include the core and face-sheet thicknesses, number of cells, core height, and additional horizontal layer(s) in the core. However, this paper only presents the effects of core and face-sheet thickness variations. In order to find the optimum values of these variables, design of experiments (DOE) and the optimization studies are carried out to maximize energy absorption under blast load. Once the optimized sandwich structure is obtained, it is then attached at the bottom of the simplified Armoured Personnel Carrier (APC) Vehicle to study its responses.

1.1.1 Study of Armoured Personnel Carrier (APC) Vehicle Model

APC M113 is proved to be the United States’ most adaptable and longest lasting APC that has been converted into many different versions, and also been in service since the Vietnam War. The APC M113 model is shown in Figure 3. It has commander and a driver seats, and could carry eleven additional men and their equipments. Since it can carry thirteen men and can be converted into different versions; this vehicle is the best candidate to be studied the effect of energy absorbing materials under blast load.

\[\text{http://www.clash-of-steel.co.uk/gallery/pages/1GWGhost_Troop.html}\]
Three cases are studied and compared with this vehicle model. Case I considers APC model without any protection concept attached at the bottom, Case II considers APC model with flat panel attached at the bottom, and Case III considers APC model with sandwich structure attached at the bottom of APC. The purpose of this comparison is to study and validate the advantages of using sandwich structures to the blast load that reduces injuries to personnel and damages to the APC vehicle.

1.2 Energy absorption and plastic deformation characteristics of sandwich structure

Prior to a further study, the characteristics of sandwich structures should be identified as the energy-absorbing device. Why is sandwich structure known as energy absorbing device and how does it absorb energy through its plastic deformation? What are the

---

important characteristics of structures in plastic deformation? What are the important
factors that designer must consider while designing one?

Sandwich structures with crushing cores are broadly employed as the main load
bearing members of structures since they have a high-strength-to-weight-ratio and
excellent energy absorption capabilities under dynamic loading conditions [4, 5]. The
difference between crushing and buckling cores is that the crushing core deforms in
plastic region that cannot be return to its original shape whereas buckling core can return
to its original shape since it deforms merely in an elastic region. The energy-absorbing
characteristic of a sandwich structure is that the core can sustain large deformations
(strains) under a constant load. Additional energy is also absorbed by the face sheets if
significant bending or stretching occurs in the structure. Sandwich cores have a behavior
of perfectly plastic over a large displacement of buckling plateau, which is compatible for
applications where it requires large energy absorption by plastic deformation that will
transfer minimum load to the support structure. As energies are being absorbed into
structure by its plastic deformation, it is essential to understand the characteristics of
structure under plastic deformation.

The important characteristics of structures in plastic deformation are shape of
deformation, impulse transfer, energy absorption in plastic deformation, and collapse
space efficiency [6]. The shape of deformation is very important characteristic since all
other parameters are depending on it and it varies greatly with varying strength of loading
for many structural configurations. Hence, it is important to choose structural
configurations that have a consistent deformation shape throughout the applied loading.
Other characteristics of energy absorption ability and collapse space efficiency are
depending on the space of plastic region in the structure. Thornton, et. al.[7] discussed that the most important feature of the absorption devices is the collapse space efficiency, which can be expressed as,

\[ \eta = \frac{E_a}{MgD} \]  

(1)

where \( \eta \) is the collapse efficiency, \( E_a \) is the energy absorbed, \( M \) is the mass, \( g \) is the gravity, and \( D \) is the collapse distance. The core layer must be spread over a large area during plastic deformation to efficiently absorb energy and this deformation should maintain as long as the blast load lasts.

All these important energy-absorbing characteristics need to be considered to design the structure that can absorb maximum amount of energy through its plastic deformation. In the next section, numerous literatures that studied various geometries and materials under different loading conditions are briefly discussed to observe the characteristic of energy absorption.

1.3 Literature Review

An attempt to gather literature in the area of energy absorption of sandwich panel under blast and impact loadings has been carried out by searching through various Internet search engines, journals, and the Lied library at the University of Nevada Las Vegas. Various keywords includes ‘Material under blast loading’, ‘Sandwich structures under blast loading’, ‘Energy absorbing materials under blast loading’, ‘honeycomb energy absorption’, and so on. Compendex search engine helped finding electronic version of literatures (.PDF) where it is directly related to www.sciencedirect.com and
many other websites. Sciencedirect.com contains millions of electronic collection of science, technology, and medicine resources with full text provided instantly. However, UNLV library does not have subscription to all the necessary journals that many literatures required. Therefore hard copies are obtained through the Document Delivery Services. More than 100 pieces of literature were collected that are somewhat related to this project and portions of them are discussed in next section to provide some key studies that helped proceeding this project.

1.3.1 Energy Absorption Characteristics of Structures

Numerous literature has discussed energy absorption characteristics of structures under static and dynamic loading conditions. Bandak and Bitzer [5] studied different honeycomb material types and various cell configurations when crushing at both static and dynamic loads to prove that honeycombs are reliable, lightweight energy absorbing materials. They noted that several deformable materials are available which will absorb various levels of kinetic energy; however, honeycomb energy absorbers give highest crush strength to weight ratio among other deformable materials because of its perfectly plastic behavior over a large displacement. This behavior will make honeycombs to absorb as much energy as possible while crushing so that it transfers minimum load to the support structure.

Goldsmith and Sackman [4] experimentally studied the impact of blunt striker on both bare honeycombs and sandwich plates with honeycomb cores at static and dynamic loads. Static tests were run using a cylindrical punch and dynamic were run blunt cylindrical strikers were launched with an initial velocity ranging from 10 to 40 m/s. Among the several conclusions made from their study, they stated that for the sandwich
plates with honeycomb cores absorbs energy by core crushing, bending, stretching of top face (where load directly applied) plate, wrinkling, and punch-through of the upper facing under the certain conditions, which core crushed significantly more than the bare honeycombs. The faceplate final deflections in both static and dynamic tests were seven to fifteen times larger than the plate thickness. They also found that the energy absorbed per unit area is the best correlation of energy absorption capacity without considering areal density.

Hutchinson and Xue [8] have studied to answer the question if metal sandwich plates with tetragonal truss core would maintain considerably larger blast loads than monolithic solid plates with same material and total mass. They have applied uniformly distributed impulse load to the both solid and sandwich plates with clamped their edges. Conclusions were made that sandwich plates with sufficiently strong cores have potential to maintain considerably larger uniform impulses than solid plates of the same material and total mass. They have discovered plastic dissipation in the face sheets and core of the sandwich plate are the factors of considerably larger energy absorption in the sandwich plate relative to the corresponding solid plate. The thinner the face sheet towards the blast, the higher the initial kinetic energy imparted to the structure that entire plate must absorb this energy by plastic deformation. It may be possible to achieve more effective design for blasts in air are to increasing the thickness of the face sheet towards the blast. Hutchinson also had the Talbot Lecture at University if Illinois Urbana Champaign (2003) and mentioned that the optimal core density come out to be about 1/3 of the total mass of the sandwich structure. He have compared with various shapes of cores, foam, textiles, trusses, folded (corrugated), and square honeycombs, and stated that the cores of
tetragonal truss, folded, and square honeycombs are more reliable to the blast load application. He also mentioned that the structure is more susceptible to shear failure if blast load is more localized.

Guruprasad and Mukherjee [6] have carried out both numerical and experimental analyses to present the behavior of layered sacrificial claddings under blast loading. They have designed the sacrificial cladding that has three layers and a stiff non-sacrificial structure. Important aspects for effective energy absorption and predictable behavior of layers have been applied to their design and discussed for only absorbing blast energy in layered sacrificial claddings. First, there should be enough space for each layer to take large deformation. Second, the layers should not rupture during the blast pulse. Third, the shape or pattern of deformation should not be changed every time for expected blast load and the layers should crush effectively. As a result, numerical results were validated through experimental results that the sacrificial cladding was very efficient in dissipating blasts. The collapse behavior of cladding was consistent and impulse transfer to main structure was marginal that they conclude the layered sacrificial claddings were effective in design of blast resistant structure.

1.3.2 Explosive Blast

Description of blast wave pattern and the effects of blast loading on structures are presented in several references, [9-13]. Türkmen and Mecitoglu [9] discussed that the shock or blast wave is generated when the air surrounded by the explosion is forcibly pushed back by the hot gases produced from the explosion source. This causes a shock wave to spread out in the air with an instantaneous high-pressure pulse propagating along with wave front. This high-pressure pulse decreases rapidly as the shock is propagating.
away from the explosion as shown in Figure 4. The pressure is then drop below the ambient pressure and it creates partial vacuum and suck in the air (Note that positive and negative phase is indicate as $T^+$, $T^-$, respectively).

Several articles have used Friedlander’s equation to describe this approximate time variation of the blast pressure [9-17]. Details of equation (2) can be found in appendix B.

\[
p(t) = p_o + P_s \left(1 - \frac{t - t_o}{t_o}\right) e^{-b\frac{(t-t_o)}{t_o}}
\]

\[\text{(2)}\]

where, \(p\) = pressure

\(p_o\) = ambient atmospheric pressure

\(P_s\) = peak overpressure
\[ b = \text{decay coefficient} \]
\[ t = \text{time} \]
\[ t_a = \text{time arrival} \]
\[ t_o = \text{time of duration for positive phase} \]

The peak overpressure and positive phase duration determine the specific impulse of the blast wave. Both blast wave parameters influence the injury and damage that the blast wave can cause. Both parameters need to be specified as some materials can resist rapid high-level blast, but will fail as the duration is extended.

1.3.2.1 CONWEP

Armstrong et. al. [18] discussed that Kingery and Bulmash [19] have developed equations to predict airblast from the free air detonation of a spherical charge and the surface detonation of a hemispherical charge. Then Hyde [20] has programmed these equations into the computer program, CONWEP. Randers-Pehrson and Bannister [21] have incorporated this CONWEP model into DYNA2D and DYNA3D creating *LOAD_BLAST boundary condition card. CONWEP model accounts for the angle of incidence of the blast wave, but does not account for the shadowing or confinement effect. When front of blast pressure hits an object, it bounds back generating secondary pressure; however, CONWEP does not account for the secondary pressure. The airblast and surface detonation types are adequately predicting free-field pressures and loads on structures that these load functions are suitable for modeling vehicle response to land mines.
CHAPTER 2

MODEL DESCRIPTION

2.1 Detail Description of Sandwich Model

The consistently used units for modeling are grams (g) for mass, centimeters (cm) for length, microsecond (μs) for time, and mega-bar (Mbar) for pressure. These units are preferred to go with the units on *LOAD_BLAST card, where “IUNIT” is set to 4.

Several types of core geometries can be designed with truss-like rods and angled or vertical walls of triangular, rectangular, and hexagonal shapes. Among them, rectangular shape core is studied in this project for the simplicity of model. As shown in Figure 5, total of twenty-five square shaped cells are created with entire model length of 45.72-cm, cell length of 9.14-cm, and core height of 5.76-cm. The model is divided into four components: inner-core (t₁), outer-core (t₂), back-face (t₃), and front-face (t₄). Front face is generally referred to as ‘blast panel’ since blast load is directly strikes into this component. The blast panel is subject to an explosive blast that is located at fixed standoff distance from the center of the panel. The panel is free-floating in space to behave as a pendulum experiment where it will have final velocity after hit by the blast load. The panel is symmetric about its’ center so a quarter-symmetry model can be used for simulation. For the results presented in this paper, the overall dimensions of the panel are fixed, the number of core cells are fixed (6.25 cells in the ¼ section model) and the
height of the core is also fixed. A 517.9-g mass of TNT is used for the explosive, which is equivalent to a 1.0-pound C-4 charge assuming a 1.14 TNT/C-4 energy release ratio and a standoff distance of 26.13-cm is used.

Figure 5 (a) Configuration of Square-cell Sandwich Structure. (b) 1/4 Section Model
We ultimately want to determine the energy transmitted to the panel by determining its steady state velocity or kinetic energy. We are also interested in determining the peak acceleration of the panel. The total mass of the structure is constrained at 4000-g so that all panels have the same mass. An equation is generated that relates the thickness of the back face to all other dimensions in the model so that the mass is the same for all panels. Since the total mass of structure can be expressed as

\[ M_t = \rho_{mat} \cdot V_t = \rho_{mat} \left(4A_c \cdot t_1 + 2A_c \cdot t_2 + A_f \cdot t_3 + A_f \cdot t_4\right) \]  

(3)

where \( M_t \) = total mass

\( \rho_{mat} \) = material density

\( V_t \) = total volume

\( A_c \) = area of core

\( A_f \) = area of face

\( t_i \) = thickness of components (\( i = 1..4 \))

Rearranging equation (1) for \( t_3 \) yields

\[ t_3 = \left[ \frac{M_t - A_c \left(4t_1 + 2t_2\right)}{A_f} \right] - t_4 \]  

(4)

Thickness variable, \( t_2 \), is always a half of \( t_1 \) because of the symmetry conditions applied to the panel, which the equation simplifies to

\[ t_3 = \left[ \frac{M_t - 5t_1A_c}{A_f} \right] - t_4 \]  

(5)

Thickness variables \( t_2 \) and \( t_3 \) are defined as equations in the template file used for the Design of Experiment and Optimization studies. The template file is used to parameterize
the input deck that identifies changes in variable while the study processes. The following input deck is used in this project to make the total mass constant as well as changing variable thicknesses from the lower to upper values.

\{parameter (th1,"thick1", 0.1, 0.09, 0.11)\}
\{th2 = th1*0.5\}
\{th3 = (2.910394029 - (1.00915135608*th1) - (0.504575678*th2)) - th4\}
\{parameter (th4,"thick4", 0.3, 0.27, 0.33)\}

2.2.1 Finite Element Model of Sandwich Structure

The sandwich model is created with only shell elements. If solid elements are used to create the model, the thickness of core is very thin and it requires at least three elements through the thickness to observe the bending, which increases the total number of elements to about 1.5-million. This model could run for couple of weeks and it is inconceivable to perform design of experiment and optimization studies, which could run up to 16-iterations for each study. Therefore, shell elements are modeled to obtain results within a reasonable period.

As shown in Figure 6, the quad-shape of mesh is used and a 1:1 length-to-width aspect ratio for the elements is maintained as closely as possible.
In Table 1, the number of shell elements along a core cell edge is varied from 6 to 60 elements to determine the effect of mesh size on the accuracy of resultant output.

![Figure 6 Finite Element Model of 1/4 section Sandwich Structure Model](image)

Table 1 Accuracy of Results with CPU Time Cost

<table>
<thead>
<tr>
<th># of elm/cell</th>
<th>Total # of elm</th>
<th>CPU Time</th>
<th>Internal Energy</th>
<th>%Diff. of IE from elm60</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>810</td>
<td>0:00:57</td>
<td>0.022515</td>
<td>16.02</td>
</tr>
<tr>
<td>12</td>
<td>3240</td>
<td>0:03:45</td>
<td>0.024098</td>
<td>8.40</td>
</tr>
<tr>
<td>24</td>
<td>12600</td>
<td>0:16:57</td>
<td>0.025050</td>
<td>4.28</td>
</tr>
<tr>
<td>36</td>
<td>22400</td>
<td>0:39:44</td>
<td>0.026136</td>
<td>0.05</td>
</tr>
<tr>
<td>48</td>
<td>50400</td>
<td>5:21:50</td>
<td>0.025931</td>
<td>0.74</td>
</tr>
<tr>
<td>60</td>
<td>79200</td>
<td>17:02:50</td>
<td>0.026123</td>
<td>0.00</td>
</tr>
</tbody>
</table>

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
The results using 36 elements per cell-edge is approximately the same (0.05% error) as the results using 60 elements per cell-edge. Also CPU time can be saved 26 times from running 60 elements to the 36 elements per cell-edge. Therefore, 36 elements per cell-edge model are used for most of the analyses reported in this paper. The total number of elements and nodes in this model is 22400 and 22185, respectively. This model is constructed from Belytschko-Tsay (ELFORM=2) shell elements with 5 integration points.

The material model 3 (*MAT_PLASTIC_KINEMATIC) is used with the properties of Aluminum 5456-H116 for all components. The material properties of model are summarized in Table 2 and input file of the LS-DYNA is shown in Table 3.
Table 2 Material Properties for Sandwich Structure

<table>
<thead>
<tr>
<th>Property</th>
<th>Aluminum 5456-H116</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material Model</td>
<td>3</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>2630</td>
</tr>
<tr>
<td>Elastic Modulus (MPa)</td>
<td>72000</td>
</tr>
<tr>
<td>Yield Strength (MPa)</td>
<td>230</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.33</td>
</tr>
</tbody>
</table>

Table 3 Material Property Section in LS-DYNA Input File

```
*MAT_PLASTIC_KINEMATIC
$HMNAME MATS 1 Aluminum-5456
$-1-2-3-4-5-6-7
$ MID RO E PR SIGY ETAN BETA
 1 2.63 0.72 0.3 0.0023
$ SRC SRP FS VP
```

A contact type of *CONTACT_AUTOMATIC_SINGLE_SURFACE card is used with slave and master is set to zero, which includes all the part IDs to ensure the contacts between various components. *CONTACT_BULK_VISCOSITY card is used to treat shock waves. This card was advised for shock wave propagation.

2.2.2 CONWEP Blast Load Function

CONWEP blast function is used to apply simple blast loading rather than to explicitly simulate the shock wave from the high explosive, which is adequate for a case that investigates vehicle responses due to the blast from land mines. Following Table 4 shows the input data required for CONWEP model in LSDYNA.
Table 4 Apply CONWEP Blast Load in LSDYNA Input File for Shell Model

*LOAD_BLAST
$-+-1--2-3-4-5-6-7
$  WGT  XBO  YBO  ZBO  TBO  IUNIT  ISURF
   517.9  0  0  -26.13  0  4  2
$  CFM  CFL  CFT  CFP

*SET_SHELL_LIST_GENERATE
$-+-1--2-3-4-5-6-7
$  SID  DA1  DA2  DA3  DA4
     777
$  B1BEG  B1END  B2BEG  B2END
    20521  28620
$

*LOAD_SHELL_SET
$-+-1--2-3-4-5-6-7
$  SID  LCID  SF  AT
     777  -2  1  0

Weight of TNT equivalent mass is 517.9-grams and it is positioned at 26.13-cm in negative Z-direction from the origin, where the origin is specified on Figure 5. “2” is selected in ISURF so that blast load to be detonated away from the structure rather than on the surface of structure. B1BEG represents the first shell ID in shell block and the B1END represents the last shell ID in shell block, which defines the shell set for applied blast surface. The Load Curve ID (LCID) is set to “-2” for CONWEP function to determine pressure for the segments and load curve scale factor (SF) can be used to increase or decrease the pressure.

2.2.3 Design of Experiment (DOE) Study

DOE study is performed using Altair HyperStudy to evaluate the factors that significantly contribute the values of responses. Responses of the study are specified as kinetic energy (KE), internal energy (IE), total energy (TE), and rigid body velocity (velocity). Both full and fractional factorial of DOE type and controlled design variables are used to evaluate the factors that contribute the values of responses. Full factorial
investigates all possible combinations of the factor levels, which 3 levels in this project, and all possible interactions between factors, which two factors (t₁, t₄) in this project. Full factorial then runs $3^2 = 9$ iterations total to evaluate the contribution of each factor to system responses. Fractional factorial can be used to reduce the number of runs and in case where full factorial is difficult to use. Fractional factorial is used to screen design variables that influence significantly to the system responses. Controlled design variable indicates that design variables that can be changed in real world environment, which is thicknesses of cores and faces.

2.2.4 Optimization Study

Altair HyperStudy is used for optimization study and it is used in conjunction with LSDYNA solver. Optimization study is practical tool to develop the design in a well-structured manner and it is performed to find the optimal combination of design variables that satisfies the stated objective function. Design variables used in this project include thicknesses of all four-components: inner-core (t₁), outer-core (t₂), back-face (t₃), and front-face (t₄). Only t₁ and t₄ are inputted as design variables in HyperStudy. Table 5 shows the initial, lower, and upper values for all four of the design variables defined.

<table>
<thead>
<tr>
<th>Design Variable</th>
<th>Initial Value</th>
<th>Lower Value</th>
<th>Upper Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner-core, (t₁)</td>
<td>0.1</td>
<td>0.04</td>
<td>0.4</td>
</tr>
<tr>
<td>Outer-core, (t₂)</td>
<td>0.05</td>
<td>0.02</td>
<td>0.2</td>
</tr>
<tr>
<td>Back-face, (t₃)</td>
<td>2.48</td>
<td>2.66</td>
<td>1.61</td>
</tr>
<tr>
<td>Front-face, (t₄)</td>
<td>0.3</td>
<td>0.2</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Table 5 Size Design Variables with Initial Value and Bounds (unit: centimeters)
The design problem can be stated mathematically in the form of optimization problem as

Objective function: \( \psi_0 (IE) \Rightarrow \max \) (6)

Side constraints: \( t_i^l \leq t_i \leq t_i^u \) (7)

The objective of the optimization problem is to maximize the internal energy absorbed by the structure. Equation (5) keeps mass constant by increasing or decreasing the back-face thickness. The side constraint is defined to limit the component thicknesses at lower to upper bounds region.

2.2 Armoured Personnel Carrier (APC) Vehicle Model

A simplified APC is modeled to study the behavior of vehicle with sandwich structure attached at the bottom. Gupta et. al. [12] provides general specifications of APC designated as APC M113. Geometry of vehicle with dimensions is shown in Figure 8. The dimension of bottom face is slightly modified from the specifications that Gupta et. al [12] presented to make ideal fit of sandwich structure to the bottom of vehicle. Dimensions within the parenthesis (see Figure 8) are from Gupta et. al [12]. Thickness of vehicle is 3.175cm for throughout the structure and the total mass of vehicle model, as shown in Figure 8, is 1,955kg. In the sandwich structure model, mass of TNT and standoff distance are referenced from the ARL ballistic pendulum experiment since the motivation of this project is initiated from it. During the literature search; however, it is discovered that average mass of C4 charge and ground clearance for the APC vehicle are different, 6-kg and 40.6-cm, respectively [23, 24]. Two sets of results will be presented from this study, a set of results with 517.9-g of TNT and 26.13-cm of ground clearance applied (SET1), and also a set of results with 6,840-g of TNT and 40.60-cm of ground clearance applied (SET2).
clearance applied (SET2). A set of results implies that the three cases comparing the results and these cases are discussed below. Sandwich structure is designed to absorb maximum energy with SET1 condition; however, it is also interested to know whether it can absorb energies or do a counter effect that damages more to the vehicle with SET2 condition. In both sets, charge is exploded at the center of the vehicle in negative Z-direction.

APC model with three cases are considered and compared for their responses. Case I is the APC model without any protection concept attached at the bottom. It is solely APC model itself to observe the behavior under blast loading. Case II is the APC model with flat panel attached at the bottom. This flat panel has the same mass and material properties of sandwich structure. It is created to compare the responses between flat-plate and sandwich structure. Case III is the APC model with sandwich structure attached at the bottom. For each case, the response of vehicle is measured at the center node specified in Figure 8.
2.2.1 Finite Element Model of APC Vehicle

2.2.1.1 CASE I: Only APC Model Without Energy Absorbing Material

Figure 9 shows fully meshed APC model only. This model is divided into top and bottom components so that it is easier to assign blast loads and boundary conditions. Quad-shape of shell element mesh is used to create the vehicle and constructed from Belytschko-Tsay (ELFORM=2) shell elements with 5 integration points. The total number of elements and nodes in this model is 25,167 and 25,365, respectively. The length of each element is 2.95-cm by 2.95-cm that aspect ratio of element size is 1 to 1.
In this model, only one material is assigned to designate all elements, Aluminum-7039. This material properties, except yield strength, are also provided from Gupta et. al [12]. Yield strength is selected from matweb.com website of Aluminum 7039-O. Material model 3 (*MAT_PLASTIC_KINEMATIC) is used with the properties of Aluminum-7039 summarized in Table 6.

A contact type of *CONTACT_AUTOMATIC_SINGLE_SURFACE card is used with slave and master set to zero to ensure the contacts between top and bottom components.
Table 6 Material Properties of APC Model

<table>
<thead>
<tr>
<th>Property</th>
<th>Aluminum-7039</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material Model</td>
<td>3</td>
</tr>
<tr>
<td>Density (kg/m^3)</td>
<td>2700</td>
</tr>
<tr>
<td>Elastic Modulus (MPa)</td>
<td>68950</td>
</tr>
<tr>
<td>Yield Strength (MPa)</td>
<td>100</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.33</td>
</tr>
</tbody>
</table>

2.2.1.2 CASE II: APC Model with Flat Plate Attached at the Bottom

A flat plate is attached at the bottom of Case I model as shown in Figure 10. Flat face is assigned the same material and the total mass as the sandwich structure for the purpose of comparing results between them. An offset of 2.265-cm is placed between flat plate and bottom face of vehicle, which accounts for thicknesses of these two components. The thickness of flat plate is 1.355-cm and thickness of bottom face is 3.175-cm that adding half thicknesses of each components gives 2.265-cm. Element size of flat plate is 0.762-cm, which is the same as sandwich model that is shown in Case III. The total elements and nodes of flat face are 72,576 and 73,279, respectively.

Two contact cards are used in this case. The same contact card of Case I is used to ensure the contacts between top and bottom components of vehicle. Also *CONTACT_SURFACE_TO_SURFACE card is used to ensure the contacts between bottom face of vehicle and flat plate. Two *INTERFACE_COMPONENT_SEGMENT cards are used to set the element segments for each components where flat plate segment is set to 1 and bottom face segment is set to 2. Flat plate segment is set as slave and bottom face segment is set as master in *CONTACT_SURFACE_TO_SURFACE card.

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
2.2.1.3 CASE III: APC Model with Sandwich Structure Attached at the Bottom

Sandwich structure is attached at the bottom of the Case I model as shown in Figure 11. An offset of 1.737-cm is placed between back-face of sandwich structure and bottom face of vehicle, which accounts for thicknesses of these two components. The thickness of back-face is 0.3-cm and thickness of bottom face is 3.175-cm that adding half thicknesses of each components gives 1.737-cm. 12 elements per cell edge are used for sandwich structure to reduce the total CPU time. The total elements and nodes of sandwich structure with 12 elements per cell edge are 250,848 and 230,775, respectively and it runs for 147-hours (approx. 6-days) to reach 2000-μs with 50-μs increment. If 36 elements per cell edge are used, total elements of sandwich structure increases to 752,544 and it runs for 407-hours (approx. 17-days) for one run. Using 12 elements over 36
elements will provide about 8% of inaccurate results, although sacrificing 8% accuracy in results is tolerable for lowering total run time by one-third.

The same contact cards from Case II are used in this case. Only difference from Case II is that the segments of back-face of sandwich structure is set to 1 on *INTERFACE_COMPONENT_SEGMENT card and it is set as slave on *CONTACT_SURFACE_TO_SURFACE card. Thicknesses of sandwich structure are 0.08-cm for cores, 0.385-cm for back-face, and 0.243-cm for front-face components. These thicknesses are from the results of optimization study at iteration 4 where minimum peak acceleration is attained. Notice that outer-core is not created since this attached structure is the full structure. Total mass of the sandwich structure is 81.78-kg (180-lb) and of course bulk of mass is from the front and back panels. The mass of core is only 12-kg (26-lb).

Figure 11 APC Model with Sandwich Structure Attached at the Bottom
2.3 Computing System and Software

2.3.1 Computing system configuration

Models are analyzed using a dual processor of AMD Athlon™ MP 2400+ AT/AT COMPATIBLE with 4GB of RAM. Running models with dual-cpu, model with 6 elements per cell edge (810 elements total) runs for 57-seconds, with 36 elements per cell edge (28620 elements total) runs for about 40-minites, and half-vehicle model with sandwich structure attached (280,687 elements total) runs for 147-hours. It is recommended to use cluster to run the vehicle model with sandwich structures attached since the elements crush so tiny that bulk of running time are used at this crushing period.

2.3.2 Commercial software

Commercial software of Pro/Engineer Wildfire, Altair Hypermesh 6.0, Altair HyperStudy 6.0, LS-DYNA, and LSPOST 2.0(Beta) are used in this project from the creation of geometry of sandwich structure to the examination of resultant dynamic response of the structure. Pro/Engineer Wildfire is used to create 3-D geometries of sandwich structure and Altair Hypermesh is used as a preprocessor to generate LS-DYNA keyword files. LS-DYNA v.960 is used to analyze the sandwich structure and LSPOST 2.0(Beta) is used to observe the dynamic behavior of structure. HyperStudy 6.0 is used to study design of experiment (DOE) and optimization of the sandwich structure.
CHAPTER 3

RESULTS AND DISCUSSIONS

3.1 Deformation History of Sandwich Structure

A typical series of deformation history of a sandwich structure is shown in Figure 12. As discussed on section 1.2, shape of deformation is the most important characteristic of energy absorption and the core of structure must have consistent deformation shape throughout the expected loading to absorb maximum amount of energy. In Figure 12, the core is indeed deformed consistent folded-like shape throughout its complete plastic deformation. The core is completely crushed without rebound at 700-microseconds, at which point the kinetic and internal energies become steady state with time.

3.2 Design of Experiment (DOE) Result

The DOE study ran through nine-iterations of varying the two thickness values and measuring changes in the internal energy. The internal energy represents the amount of energy being absorbed by plastic deformation of sandwich structure. Therefore, the response of interest that is used for the DOE is the internal energy, which denoted as IE. It is desired to identify which design variable contributes significantly to the internal energy. Figure 13 shows the graph of percent contribution by each design variables for the internal energy.
In the internal energy graph, it is indicated that varying inner-core thickness influences about 7% of internal energy absorption to the structure and varying front-face thickness influences about 93%. This graph is not an indication of percentage that each component has absorbed the internal energy. It is, however, used to indicate the sensitivity of the internal energy absorption to changes in each design variable.
The results of DOE can be verified from the optimization result shown in Table 7. For iterations 1 and 3, when thick1 stays constant and thick4 has varied 22%, internal energy has changed about 21%, which indicates that internal energy changes by almost the same percentage amount as the changes in thick4. Equally, for iteration 7 to 9, when thick4 stays constant and thick1 has varied 31%, internal energy has changed only about 4%, which indicates that internal energy changes fairly small amount to the changes in thick1. Therefore, the DOE results are verified from the result of optimization study.

3.3 Optimization Result

The HyperStudy optimization results for maximum internal energy were also attained after nine iterations. Table 7 shows the design variables and model responses for each iteration. Table 8 shows the optimum values of variables (over the range prescribed) that maximize internal energy of the structure.
Table 7 History of Optimization Iteration

<table>
<thead>
<tr>
<th>Iteration</th>
<th>Objective</th>
<th>thick1</th>
<th>thick4</th>
<th>MASS</th>
<th>KE</th>
<th>IE</th>
<th>TE</th>
<th>Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.042274</td>
<td>0.10000</td>
<td>0.20000</td>
<td>3999.660</td>
<td>0.0059163</td>
<td>0.042274</td>
<td>0.0481076</td>
<td>0.0016852</td>
</tr>
<tr>
<td>2</td>
<td>0.0397610</td>
<td>0.12200</td>
<td>0.30000</td>
<td>4000.690</td>
<td>0.0057983</td>
<td>0.0397610</td>
<td>0.0455843</td>
<td>0.0016772</td>
</tr>
<tr>
<td>3</td>
<td>0.0332014</td>
<td>0.10000</td>
<td>0.36600</td>
<td>3999.660</td>
<td>0.0057963</td>
<td>0.0332014</td>
<td>0.0389959</td>
<td>0.0016722</td>
</tr>
<tr>
<td>4</td>
<td>0.0572231</td>
<td>0.09300</td>
<td>0.24300</td>
<td>3999.340</td>
<td>0.0060300</td>
<td>0.0572231</td>
<td>0.0634022</td>
<td>0.0017065</td>
</tr>
<tr>
<td>5</td>
<td>0.0723142</td>
<td>0.06480</td>
<td>0.20000</td>
<td>4000.480</td>
<td>0.0062698</td>
<td>0.0723142</td>
<td>0.0815323</td>
<td>0.0017337</td>
</tr>
<tr>
<td>6</td>
<td>0.0612276</td>
<td>0.05184</td>
<td>0.23800</td>
<td>4000.000</td>
<td>0.0062340</td>
<td>0.0612276</td>
<td>0.0702117</td>
<td>0.0017236</td>
</tr>
<tr>
<td>7</td>
<td>0.0745510</td>
<td>0.05248</td>
<td>0.20000</td>
<td>3999.750</td>
<td>0.0063379</td>
<td>0.0745510</td>
<td>0.0846609</td>
<td>0.0017425</td>
</tr>
<tr>
<td>8</td>
<td>0.0761175</td>
<td>0.04199</td>
<td>0.20000</td>
<td>3999.420</td>
<td>0.0064220</td>
<td>0.0761175</td>
<td>0.0874756</td>
<td>0.0017495</td>
</tr>
<tr>
<td>9</td>
<td>0.0778678</td>
<td>0.04000</td>
<td>0.20000</td>
<td>4000.090</td>
<td>0.0064396</td>
<td>0.0778678</td>
<td>0.0880312</td>
<td>0.0017503</td>
</tr>
</tbody>
</table>

Table 8 Optimized Design Variable Values (unit: centimeters)

<table>
<thead>
<tr>
<th>Design Variable</th>
<th>Optimum Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner-core, (t1)</td>
<td>0.04</td>
</tr>
<tr>
<td>Outer-core, (t2)</td>
<td>0.02</td>
</tr>
<tr>
<td>Back-face, (t3)</td>
<td>2.66</td>
</tr>
<tr>
<td>Front-face, (t4)</td>
<td>0.2</td>
</tr>
</tbody>
</table>

All of the response values were taken at the termination time (at 2000-microsecond) where energies and velocity had reached a steady state. Table 7 clearly indicated that internal energy increased from 0.042 to 0.078, about 86% from iteration 1 to 9. The inner-core (thick1) decreased 60% and the blast-face (thick4) decreased 33% from iteration 1 to 9, which are at lower bound values. The optimized values indicate that the internal energy increases as the wall thickness decreases for the core and the blast face.

Other energy values were also checked for consistency. LSDYNA calculates total energy in GLSTAT by adding six different energies: internal, kinetic, contact (sliding), hourglass, system damping, and rigidwall. Figure 14 shows all the energies encountered from the model. Adding energies from A to E gives a value of F at any given time.
Figure 14 Total Energy Distributions For Iteration 1 of the Optimization Study

One problem observed in the optimization results is that the total energy changes significantly throughout the iterations even though the blast load applied to the structure remains the same for all iterations. Ideally, we expected the total energy to be constant since the applied load is the same. So, even though the internal energy increased by 86% from iteration 1 to 9, the kinetic energy also increased by 8.5%. This is not a desirable result but it also corresponds to some experimental data found from ballistic pendulum experiments.
3.3.1 Dishing Effects

One possible explanation for the increase in total energy from iteration 1 to 9 is related to the deformation pattern of the blast face. Hanssen et. al.[3] explain the deformation pattern of blast face in detail.

Figure 15 Illustration of Global and Local Dishing Effects.

As shown in Figure 15, the core of the panel crushes more in the center than at the edges, forming a bowl or dish shape, since the pressure from the blast is higher in the
center. As the panel deforms in this manner, the normal direction of each element on the blast face is more closely oriented towards the blast center. The pressure from the blast on each element increases as the elements become more perpendicular to the radially expanding blast wave. The increased pressure on the blast face would account for the increase in total energy to the panel.

3.3.2 Uniform Pressure

As shown in Figure 16, a uniform pressure pulse is applied to each element on the blast face to investigate this phenomenon further. This uniform pressure is applied to at least eliminate the global dishing effect to observe the change in energies.

Figure 16 Fully Deformed Shape of Sandwich Structure Under Uniform Pressure
Friedlander’s decay function [9-17] is used to generate the pressure profile of blast load and its equations can be seen in appendix B. *DEFINE_CURVE is used to apply blast pressure and time.

Under this pressure loading, the panel crushed uniformly for all iterations of different cell wall and face sheet thicknesses. The applied load in this case is identical at each iteration and the kinetic energy decreased slightly as the internal energy increased as shown in Table 9. Final rigid body velocity is decreased due to decrease in kinetic energy; however, total energy is still increased due to the combination of local dishing effect and higher energy absorption in the core. Higher energy absorption in the core is attained since more core walls are crushed absorbing more energy while uniform pressure is applied.

<table>
<thead>
<tr>
<th>Iter...</th>
<th>Objective</th>
<th>thick1</th>
<th>thick2</th>
<th>thick3</th>
<th>mass</th>
<th>KE</th>
<th>IE</th>
<th>TE</th>
<th>Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0144361</td>
<td>0.100000</td>
<td>0.300000</td>
<td>3999.6600</td>
<td>0.0051862</td>
<td>0.0144361</td>
<td>0.0198509</td>
<td>...</td>
<td>0.0016052</td>
</tr>
<tr>
<td>2</td>
<td>0.0121662</td>
<td>0.122000</td>
<td>0.300000</td>
<td>4000.6900</td>
<td>0.0052045</td>
<td>0.0121662</td>
<td>0.0175842</td>
<td>...</td>
<td>0.0016051</td>
</tr>
<tr>
<td>3</td>
<td>0.0103791</td>
<td>0.100000</td>
<td>0.366000</td>
<td>3999.6600</td>
<td>0.0052301</td>
<td>0.0103791</td>
<td>0.0157646</td>
<td>...</td>
<td>0.0016106</td>
</tr>
<tr>
<td>4</td>
<td>0.0229887</td>
<td>0.080000</td>
<td>0.243000</td>
<td>3999.3400</td>
<td>0.0052508</td>
<td>0.0229887</td>
<td>0.0288957</td>
<td>...</td>
<td>0.0015980</td>
</tr>
<tr>
<td>5</td>
<td>0.0325934</td>
<td>0.064800</td>
<td>0.200000</td>
<td>4000.4800</td>
<td>0.0051600</td>
<td>0.0325934</td>
<td>0.0394942</td>
<td>0.0016106</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>0.0294244</td>
<td>0.077600</td>
<td>0.200000</td>
<td>3999.5800</td>
<td>0.0052067</td>
<td>0.0294244</td>
<td>0.0357838</td>
<td>...</td>
<td>0.0015987</td>
</tr>
<tr>
<td>7</td>
<td>0.0363177</td>
<td>0.052480</td>
<td>0.200000</td>
<td>3999.7500</td>
<td>0.0051294</td>
<td>0.0363177</td>
<td>0.0438809</td>
<td>...</td>
<td>0.0015924</td>
</tr>
<tr>
<td>8</td>
<td>0.0401370</td>
<td>0.0419904</td>
<td>0.200000</td>
<td>3999.4200</td>
<td>0.0051491</td>
<td>0.0401370</td>
<td>0.0481333</td>
<td>...</td>
<td>0.0015967</td>
</tr>
<tr>
<td>9</td>
<td>0.0408688</td>
<td>0.040000</td>
<td>0.200000</td>
<td>4000.0900</td>
<td>0.0051613</td>
<td>0.0408688</td>
<td>0.0496068</td>
<td>...</td>
<td>0.0015975</td>
</tr>
</tbody>
</table>

The results discussed above imply that a sandwich structure used for blast mitigation can be tailored to maximize energy absorption, but this may also result in an increase in kinetic energy (or final velocity) applied to the structure in back of the panel. In general, this is not desirable but one other result to consider is how fast the back plate is accelerated to its final velocity.
3.3.3 Peak Acceleration of Sandwich Structure

Damage to a human body due to landmines is from the effects of the explosive shock front impacting the body. Boyd [22] stated that blast event causes two types of acceleration. The first is extremely elevated acceleration level with small displacement. The second is a much lower acceleration with a large displacement. The second type of acceleration is similar to that experienced in car crash. The first type of acceleration can cause severe injury to the human body since the shock wave passes through the body at a rate greater than it can absorb the energy. Decreasing this extremely elevated (peak) acceleration can reduce the injury to personnel, which then can increase the chances to complete the mission successfully.

![Figure 17 Comparison of Peak Acceleration of Sandwich Structure vs. Plain-Plate](image)

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
Figure 17 shows the peak acceleration of each component at all nine iterations along with the peak acceleration for a plain (flat) plate model with the same total mass of 4.0-kg. These peak accelerations are attained from the MATSUM outputs where average of nodes in each component is obtained. The front face and core walls accelerate very fast as the core crushes. But in all iterations, the back face accelerates slower than the flat plate. The lowest peak acceleration, $6.02E-06 \text{ cm/\musec}^2$, occurs during iteration 4. This is about a 73% reduction in peak acceleration compared to the flat plate case, which had a value of $2.25E-05 \text{ cm/\musec}^2$. This percent difference indicates that sandwich structure can be used to reduce the significant amount of peak acceleration to the main structure. Then it is important to know how the bottom face of APC vehicle experiences the peak acceleration. The charge is detonated at the center of the vehicle so that the behaviors of peak acceleration are mainly observed in the surrounding areas from the center. This study should provide the location of the maximum peak acceleration that the bottom face of vehicle experiences and also descending of the peak acceleration as it is measured further from the center.

3.4 Comparison of Three Cases in APC Model

A set of nodes are outputted using *DATABASE_NODOUT card as shown in Figure 18 for all three cases. These nodes are selected from vicinity of the center node since blast pressure is applied to the center node and to the surrounding areas intensely. A set of nodes is selected in three directions: left, right, and bottom from the center node. The purpose of selecting node in such way is to observe how the peak accelerations die down as it gets away from the center node.
When peak accelerations of three cases are compared along these three directions, it will clearly show which case is more beneficial to reducing the overall peak acceleration.

3.4.1 Results of 517.9-g of TNT and 26.13-cm of Ground Clearance (SET1)

Figure 19 shows the peak acceleration comparison of nodes outputted in vicinity of the center nodes for three cases. Actual graphs of these nodes and series of deformation history of APC vehicle with all three cases can be seen at appendix E and F respectively. As seen in top view of vehicle model, white lines indicate the nodes that are outputted in left, right, and bottom directions from the center node. The first point of each case in three graphs represents the peak acceleration at the center node. Twelve nodes in three directions are compared and several comments can be made from this figure:

- Peak accelerations of three cases declines as it moves further from the center
- Peak accelerations are not symmetric in three directions since the vehicle geometry is not symmetric at the center.
- The highest peak acceleration does not occur at the center node except for the vehicle with flat plate case. Also, the highest peak acceleration of all cases within the graphs occurred at most two nodes away from the center.

- The vehicle with flat plate case experiences highest peak acceleration and largest up and downs among three cases, which is an unexpected result.

- Over all, APC vehicle with sandwich structure experiences the lowest peak accelerations among three cases.

Figure 19 Peak Accelerations of Nodes for Three Cases (SET1)
Since the highest peak acceleration for three cases is occurred within three nodes away from the center, all the nodes within seven nodes are outputted for three cases to find the highest peak acceleration that bottom face of vehicle experiences.

![Figure 20 Outputted Nodes to Find the Highest Peak Acceleration](image)

The peak acceleration of all nodes in case I is input into a single graph as shown in Figure 21. In zoomed view of several nodes at their peak accelerations, ten nodes in the vicinity of center are experienced higher peak acceleration than the highest peak acceleration of case III.

Figure 22 shows the location of ten nodes from the center and the center node is also one of ten nodes. Note that in case II, thirteen nodes are experienced higher peak acceleration than the highest peak acceleration of case III. Figures of case II are not shown in this paper since main purpose of this work is to show the differences of peak acceleration between case I and case III. In Figure 23, however, have only one node experience much higher peak acceleration than rest of nodes in case III. Figure 24 shows the location of this highest peak acceleration in case III.

42
Figure 21 Peak Acceleration of All Nodes Inputted into a Single Graph for Case I

Figure 22 Ten Nodes that Experience Higher Peak Acceleration than the Highest Peak Acceleration of Case III
Figure 23 Peak Acceleration of All Nodes Inputted into a Single Graph for Case III

Figure 24 The Highest Peak Acceleration in Case III
The percent differences of peak acceleration among these cases are shown in Table 10. These differences are based on the highest peak acceleration that the bottom face of the vehicle experiences in each case.

**Table 10 Percent Differences of Peak Acceleration for Three Cases (SET1)**

<table>
<thead>
<tr>
<th>Case</th>
<th>Peak Accel. (cm/μsec²)</th>
<th>%Δ I vs. II</th>
<th>%Δ I vs. III</th>
<th>%Δ II vs. III</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case I</td>
<td>6.3206E-05</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Case II</td>
<td>9.3107E-05</td>
<td>- 47.3%</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Case III</td>
<td>3.4129E-05</td>
<td>-</td>
<td>46.0%</td>
<td>63.3%</td>
</tr>
</tbody>
</table>

An attached thick flat plate on the bottom of an APC vehicle is expected to have lower peak acceleration than the vehicle without any energy absorbing materials, since thicker plates absorb more energy or lowers the acceleration. However, case II experiences 47.3% higher peak acceleration than Case I. The cause of result is uncertain at this point; however, further study will be carried out to locate the reasons for it. APC vehicle with sandwich structure experiences 46% lower peak acceleration than APC vehicle without energy absorbing materials. If average of peak accelerations are compared from using the MATSUM results, case I experiences 5.0504E-7 cm/μsec² and case III experiences 2.7278E-7 cm/μsec² that still case III experiences 46% lower peak acceleration than case I. However, case II experiences 4.1495E-7 cm/μsec² of peak acceleration that is 17.8% lower than case I and 36.3% higher than case III when MATSUM results are compared. Case II may experience some concentrated peak

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
accelerations in the vicinity of the center, but it actually reduces 17.8% of peak acceleration over case I when averaged peak accelerations of all nodes are measured.

3.4.2 Results of 6,840-g of TNT and 40.60-cm of Ground Clearance (SET2)

Figure 25 shows the peak acceleration comparison of nodes outputted in vicinity of the center nodes. Actual graphs of these nodes and series of deformation history of APC vehicle with all three cases can be seen at appendix G and H respectively.

![Figure 25 Peak Accelerations of Nodes for Three Cases (SET2)](image)

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
When the highest peak accelerations are compared from Figure 25, APC vehicle with sandwich structure experiences 1010.7% higher peak acceleration than APC vehicle without energy absorbing materials. If average of peak accelerations are compared from this figure, case III experiences 237.7% higher peak acceleration than case I. Case I experiences the lowest peak acceleration among three cases. The blast pressure applied to these cases is excessively elevated that sandwich structure does not have time to absorb energy or lower the acceleration. In fact, sandwich structure is operating as a bulk of mass, striking the bottom face of vehicle that results in extreme peak acceleration than case I and case II. However, if MATSUM results of peak acceleration are compared, case I, II, and III experiences 4.0689E-6, 2.7963E-6, and 4.1792E-6, respectively that case II experiences the lowest peak acceleration among three cases and case III experiences 2.8% higher peak accelerations overall.
CHAPTER 4

COMPARISON OF LS-DYNA RESULTS TO EXPERIMENT

The results of ballistic pendulum experiment from the Army Research Laboratories are compared with the LS-DYNA results. The purpose of this comparison is to examine how accurate the LS-DYNA result represents the experimental results. The parameters and results of ARL experiment model are presented by Skaggs [2]. Skaggs [2] have done the experiments with various geometries and materials of energy absorbing concepts; however, only the baseline (without any protection concepts) model is compared with LS-DYNA for the simplicity of comparison in this chapter.

4.1 Parameters of Ballistic Pendulum Model

Figure 26 shows the schematic drawings of ballistic pendulum model for ARL experiment and LSDYNA with dimensions. A full model is created in LS-DYNA with 403 elements in total. The pendulum arm is created with beam elements and the pendulum face with shell elements. Dimensions of ARL and LS-DYNA are the same as well as the mass of the pendulum bob and arm, 451.16-kg, 350.27-kg, respectively. The material cards for both beam and face of the pendulum in LS-DYNA are “MAT_PLASTIC_KINEMATIC with a very high density to match the mass from the experiment. The top of the beam element is constrained in the x, y, and z-directions but free to rotate in any direction so that it will act as a hinge. 453.6-g of C4 charge is used
in ARL experiment and 517.9-g of TNT charge is used in LS-DYNA, both with 26.13-
cm of standoff distance from the center of pendulum face. ARL have determined the

Figure 26 Schematic Drawings of Ballistic Pendulum Model for ARL and LS-DYNA

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
energy transfer to the main structure by measuring the displacement at the center of gravity point; thus, node# 447 is outputted in LS-DYNA model to measure the displacement after the blast load has been applied.

4.2 Comparison of Results at the Center of Gravity Point

ARL carried out 15 individual baseline tests and had an average displacement of 16.36-cm at the center of gravity with one-sigma error of 3.68% (15.76-cm < 16.36-cm < 16.96-cm). LS-DYNA results are shown in Figure 14. The resultant displacement at the bottom-center of the pendulum face is 20.28-cm and the resultant displacement at node# 447 (center of gravity point) is 16.22-cm, which 16.22-cm gives 0.86% error from the results of ARL, 16.36-cm. The time to reach 16.22-cm is 700000-μs. The LS-DYNA results are validated since it produces accurate result (<1% error) compared to ARL experiments in ballistic pendulum model.
Figure 27 Resultant Displacement of Ballistic Pendulum from LS-DYNA
CHAPTER 5

PRELIMINARY ANALYSIS OF OTHER VARIABLES

Numerous variables can be applied to the sandwich structure that reduces peak accelerations. The variables include the number of cells, core height, additional horizontal layer(s) in the core, material properties of sandwich structure, and pre-specified dent of the core. Only one case of each variable is carried out to examine the behavior of peak acceleration. Please note that the peak accelerations of each case are obtained from the MATSUM outputs.

5.1 Number of Cells

The number of cells is doubled from 6.25 to 12.5 as shown in Figure 28. The mass of core in each case is the same so that the core thickness of 12.5-cell model is reduced to a half of the core thickness of 6.25-cell model. The result shows that 12.5-cell model reduces 16% of peak acceleration over 6.25-cell model. It is a reasonable result since 12.5-cell model have reduced local dishing effect and it also deformed plastically using additional spreading area, which is an important factor to efficiently absorb energy. Varying number of cells is a promising parameter that can lead to lowering peak accelerations; therefore, attain the optimum number of cells is necessary for such result.
5.2 Core Height

The core height of sandwich model is increased from 5.76-cm to 15.00-cm as shown in Figure 29. The mass of core in each case is the same so that the core height of 15.00-cm has thickness of core of 0.03845-cm. The result shows that 15.00-cm core height model reduces 62% of peak acceleration over 5.76-cm core height model. This great reduction in peak acceleration is mainly due to the plastic deformation using almost three times of spreading area that is an important factor to efficiently absorb energy. Varying core height is another promising parameter that can lead to lowering peak accelerations; therefore, more study is needed to obtain the optimum core height.

5.3 Additional Horizontal Layer(s) in the Core

Horizontal layers are added in the core as shown in Figure 30. The properties of each layer are the same as blast-face, 0.3-cm thickness and material properties of Aluminum-5456. The location of a layer is approximately one-third from the blast face and approximately two-third for the second layer. Total mass of structures are the same at 4.0-kg and adding these layers did not changed core and face thicknesses except for the
back-face. The back-face thickness is 2.4842-cm without any layers added, 2.1842-cm with a single layer added, and 1.8842-cm with two layers added to make the total mass of structure constant.

Addition of a single layer in the core reduces 13% of peak acceleration and double layers reduce 9% of peak acceleration over the base model. So adding double layers in the core may not be as effective as with a single layer lowering the peak acceleration. Other factors that need to be considered with this variable are the location, thickness, and
material properties of a layer. Additional horizontal layer in the core is one more promising parameter that can lead to lowering peak accelerations; therefore, more study is needed to obtain the optimum number of layers, location, thickness, and material properties.

5.4 Vary the Material Properties

The APC vehicle with sandwich structure model is used to vary the material properties of sandwich structure. In this case, Aluminum-7039 (the material of APC vehicle) is assigned to the entire sandwich components since it has significantly lower yield strength than Aluminum 5456. As a result, bottom face of vehicle experienced decrease in average peak acceleration by 5% over the original model. In this parameter, design of experiment and optimization studies is necessary to attain the optimum material properties. Applying the optimum material properties to the structure can be an additional factor that might be effective in reducing the peak acceleration.

5.5 Pre-Specified Dent of the Core

The pre-specified dent of core can also be a factor to lower the peak acceleration. As shown in Figure 31, the core is dented 0.38-cm in x-direction at 1.44-cm in Z-direction. This dent is arbitrarily chosen to compare the results between dented and un-dented model. As a result, the dented model experiences average of 45.7% lower peak acceleration than the un-dented model. This result indicates the pre-specified dent can also be the beneficial parameter to lower the peak acceleration. It might be more effective if dent is specified at the lower position in Z-direction and much smaller size of dent.
The dent of the core shown in Figure 31 is not appropriate for the actual study since it is globally dented; however, the idea of lower peak acceleration can be obtained from this model. Merely small dent is needed to initiate the crushing of core, which is an appropriate method to be utilized.

Figure 31 Pre-Dented Core with Dimensions

5.6 Summary

Many variables can be used to lower the peak acceleration that can minimize the damages to both human body and as well as the vehicle structures. The variables discussed in this chapter can be the effective methods to minimize the damages. First, systematical study of each variable to attain the optimum values of lowering the peak acceleration needs to be carried out. Then combinations of all the optimized variables are also need to be studied to attain the lowest peak acceleration that the bottom face of APC vehicle can experience.
CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

Computational analysis of non-uniform dynamic response for the sandwich structure under blast load is successfully carried out. It is observed that the non-uniform deformation pattern (dishing) tends to increase the total energy applied to the structure, which increases its final velocity. These computational results are in agreement with experimental data for ballistic pendulum experiments. However, the optimum design of sandwich structure significantly reduces the peak acceleration, average of 73% over the flat plate. The benefits of reduced peak acceleration may outweigh the drawbacks of increased kinetic energy depending on the particular structural application.

The optimum design of sandwich structure is then attached at the bottom of APC vehicle with SET1 loading condition and it showed 46% reduction in peak acceleration over the APC vehicle without any energy absorbing materials. Based on this reduction of peak accelerations, sandwich structures can be tailored to minimize the damages to the vehicle from a blast load. However, when sandwich structure with SET2 loading condition is applied, sandwich structure behaves as a bulk of mass striking the bottom face of vehicle that resulted in extreme peak acceleration. Therefore, sandwich structure can be beneficial with the SET1 loading condition but destructive with SET2.
6.2 Recommendations

- Sandwich structure is designed to perform the best with SET1 loading condition. However, sandwich structure may need to be re-designed to minimize the damages to the vehicle with SET2 loading condition, since SET2 is the common loading condition that APC vehicle experience.

- Full investigation of numerous variables discussed in chapter 4 needs to be carried out to attain the optimum sandwich structure that transfers the minimum peak acceleration to the APC vehicle.

- Critical locations such as driver and commander seats and attachment points of secondary systems need to be studied under blast loading to minimize the damages that can cause. Secondary systems under blast load can be damaged and misaligned to sensitive equipments that can reduce the overall performance of vehicle.

- Blast load should be applied at various locations of the vehicle to study the effects of critical locations.

- Different types of core such as hexagonal shape or truss-like rods are needed to be studied to compare the performances under blast load.

- Refined mesh of vehicle as well as sandwich structure should be applied to obtain the accurate results.

- The acceleration data is collected every 25-μs in this project. However, the question has been raised if 25-μs of time interval is actually capturing the highest peak acceleration that the main structure experiences. To answer this question, Case I and Case III models are re-run with smaller time intervals to collect the
acceleration data: 10, 1, 0.1, and 0.01-μs. The results are shown in appendix H. At the time interval of 10-μs, the peak acceleration between cases I and III are almost identical; however, at 1-μs case III experiences 70% higher peak acceleration than case I. By only looking at case III data, at 1-μs of time interval captures the highest peak acceleration among different time intervals and at 0.1 and 0.01-μs captures the lower peak accelerations than at 1-μs. Assuming data collected at 1-μs is correct, then data from 0.1 and 0.01-μs should be higher or equal, if not lower, but only a little, to the data at 1-μs; however, that is not the case in the result that inconsistent data are obtained. By only looking at case I data, the peak acceleration is converged from 10-μs to the lower time intervals. Therefore, at 10-μs of time interval captures the highest peak acceleration that the main structure experiences according to case I data, which shows consistency of results. Since case III outputs inconsistent data but case I outputs consistent data that more careful study needs to be carried out to determine the correct time interval that can capture the most accurate peak acceleration. Some of the things to check first would be the time steps that are small enough to capture in such lower time intervals; or it may have some problems on contact cards with penalty factors. No matter what may be the problem, the correct time interval to collecting data can be a critical factor that needs to be determined prior to any experimental or computational work.
APPENDIX A. VARIOUS CALCULATIONS
A.1 Unit Conversion

[kg, m, s] → [g, cm, micro-sec]

Pressure:

\[ \text{1 GPa} = 1 \frac{g}{\mu s} \cdot 10^{-2} \frac{m}{cm} = 10^{-2} \frac{g}{cm \mu s} \]

Density:

\[ \text{1 kg/m}^3 = 10^3 \frac{g}{m^3} \cdot 10^{-6} \frac{m^3}{cm^3} = 10^{-3} \frac{g}{cm^3} \]

Velocity:

\[ \text{1 km/s} = 10^3 \frac{cm}{km} \cdot 10^{-6} \frac{s}{\mu s} = 10^{-3} \frac{cm}{\mu s} \]

Normal Energy:

\[ 1 J = 1 Nm = 1 \frac{kgm}{s^2} \cdot \frac{1}{1} = 1 \frac{kgm^2}{s^2} = \frac{1000g}{kg} \cdot \frac{10000cm^2}{m^2} \cdot \left( \frac{1 \cdot 10^{-6} s}{\mu s} \right)^2 = 1 \cdot 10^{-5} \frac{gcm^2}{\mu s^2} \]
A.2 Friedlander’s Decay Function

Friedlander’s decay function provides blast wave profile of the peak overpressure – time history curve as shown in Figure 4. The blast wave profile can be expressed as

\[ p(t) = p_o + P_s \left(1 - \frac{t - t_a}{t_o}\right) e^{-b(t-t_a)} \]  

(A.2-1)

where,

- \( p \) = pressure (absolute)
- \( p_o \) = referenced ambient atmospheric pressure (absolute)
- \( P_s \) = peak overpressure
- \( b \) = decay coefficient
- \( t_a \) = time arrival
- \( t_o \) = time of duration for positive phase
- \( t \) = time
- \( e \) = base natural logarithms

Peak overpressure for the chemical explosion can be expressed as with function of \( Z \),

\[ P_s = \frac{808 \left[1 + \left(\frac{Z}{4.5}\right)^2\right]}{\sqrt{1 + \left(\frac{Z}{0.048}\right)^2} \sqrt{1 + \left(\frac{Z}{0.32}\right)^2} \sqrt{1 + \left(\frac{Z}{1.35}\right)^2}} P_o \]  

(A.2-2)

\( Z \) is the scaled distance,

\[ Z = \frac{f_d R}{3W} \]  

(A.2-3)
where

\[ R = \text{distance from the center of the explosive to a given location} \]
\[ W = \text{weight of explosive} \]

\[ f_d = \text{transmission factor for distance} = \left( \frac{p}{p_o} \right)^{1/3} \times \left( \frac{T_o}{T} \right)^{1/3} \quad (A.2-4) \]

where \( o \) is for the reference atmosphere

\[ T = \text{atmospheric temperatures} \]
\[ T_o = \text{absolute temperature in a reference temperature} \]

Time of arrive can be expressed as

\[ t_a = \frac{1}{a_x} \int_{r_c}^{r} \left[ \frac{1}{1 + 6 p_o / 7 P_a} \right] dr \quad (A.2-5) \]

where

\[ r = \text{distance} \]
\[ r_c = \text{charge radius} \]
\[ a_x = \text{speed of sound in the undisturbed atmosphere} \]

Time of duration for the chemical explosion can be expressed as

\[ t_d = \frac{980 \left[ 1 + \left( \frac{Z}{0.54} \right)^{10} \right]}{\left[ 1 + \left( \frac{Z}{0.02} \right)^3 \right] \left[ 1 + \left( \frac{Z}{0.74} \right)^6 \right] \sqrt{1 + \left( \frac{Z}{6.9} \right)^2} W^{1/3}} \quad (A.2-6) \]
A.3 Calculation to Vary the Number of Cells within the Given Mass

\[
\rho_{\text{Area}} = \frac{\rho_{\text{Mat}} \left(1 - k^2 \right)(L^2 \lambda T) + (2TL^2)}{L^2}
\]  \hfill (A.3-1)

where

\(\rho_{\text{Area}}\) = area density \(\left(\frac{\text{lb}}{\text{ft}^2}, \frac{\text{g}}{\text{cm}^2}\right)\)

\(\rho_{\text{Mat}}\) = material density \(\left(\frac{\text{lb}}{\text{ft}^3}, \frac{\text{g}}{\text{cm}^3}\right)\)

\(L\) = length of plate (ft, cm) = \(\sqrt{n} \times x\)

\(x\) = length of unit cell (ft, cm)

\(l\) = length of square (ft, cm)

\(T\) = thickness of the plate (ft, cm)

\(\lambda = \frac{h}{n}\), scale factor of height of core to the plate thickness

\(h\) = core height (ft, cm)

\(n = \text{number of cells in honeycomb core} = \left(\frac{l}{x}\right)^2\)

\(k = \text{ratio of length of square over length of unit cell} = \frac{1}{x}\)

Solve for \(\lambda\) from equation C-1 and then core height divided by \(\lambda\) will give the number of cells with the constant mass.
A.4 Calculation to Vary the Face-Thickness and Core-Height

\[ \rho_{\text{Area}} = \frac{\rho_{\text{Mat}}(x^2 - l^2)(n\lambda T) + (2TL^2)}{L^2} \]  

(A.4-1)

where \( \rho_{\text{Area}} \) = area density \( \frac{\text{lb}}{\text{ft}^2 \cdot \text{cm}^2} \)

\( \rho_{\text{Mat}} \) = material density \( \frac{\text{lb}}{\text{ft}^3 \cdot \text{cm}^3} \)

\( L \) = length of plate (ft, cm)

\( x \) = length of unit cell (ft, cm)

\( l \) = length of square (ft, cm)

\( T \) = thickness of face (ft, cm)

\( \lambda \) = \( \frac{h}{n} \), scale factor of height of core to the plate thickness

\( h \) = core height (ft, cm)

\( n \) = number of cells in honeycomb core

Solve \( T \) for the face-thickness or solve for \( \lambda \) from equation D-1 and then multiply it by the number of cells will give the height of core.
APPENDIX B. THE PEAK ACCELERATION COMPARISON OF NODES
OUTPUTTED IN VINCINITY FROM THE CENTER NODE FOR THREE CASES

(Set1 & Set2)
B.1 The Peak Acceleration Comparison for three cases. To the Left from the Center Node (SET1)

Comparison of Three Cases: To the Left from the center

Veh_O = Vehicle Only = Case I
Veh_F = Vehicle with Flat-Plate = Case II
Veh_S = Vehicle with Sandwich Structure = Case III

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
B.2 The Peak Acceleration Comparison for Three Cases in Three Different Directions (SET2)

Comparison of Three Cases: To the Left from the Center

- Veh_O = Vehicle Only = Case I
- Veh_F = Vehicle with Flat-Plate = Case II
- Veh_S = Vehicle with Sandwich Structure = Case III

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
APPENDIX C. SERIES OF DEFORMATION HISTORY: MAXIMUM
DISPLACEMENT AT CENTER NODE FOR ALL THREE CASES (SET1 & SET2)
C.1a Series of Deformation History – Maximum Displacement at Center Node Case I (SET1)

- Time = 200-μsec, Max. Disp. = 0.2081-cm
- Time = 1000-μsec, Max. Disp. = 0.9753-cm
- Time = 3000-μsec, Max. Disp. = 1.8043-cm
C.1b Series of Deformation History – Maximum Displacement at Center Node Case II (SET1)

- Time = 200-µsec, Max. Disp. = 0.1664-cm
- Time = 1000-µsec, Max. Disp. = 0.7451-cm
- Time = 3000-µsec, Max. Disp. = 1.3867-cm
C.1c Series of Deformation History – Maximum Displacement at Center Node Case III (SETI)

- **Time = 200-μsec, Max. Disp. = 0.0383-cm**
- **Time = 1000-μsec, Max. Disp. = 0.7959-cm**
- **Time = 3000-μsec, Max. Disp. = 1.6116-cm**
C.2a Series of Deformation History – Maximum Displacement at Center Node Case I (SET2)

Time = 200-μsec, Max. Disp. = 0.6465-cm

Time = 1000-μsec, Max. Disp. = 7.8166-cm

Time = 3000-μsec, Max. Disp. = 14.1230-cm
C.2b Series of Deformation History – Maximum Displacement at Center Node Case II (SET2)

- Time = 200-μsec, Max. Disp. = 0.4866-cm
- Time = 1000-μsec, Max. Disp. = 5.7027-cm
- Time = 3000-μsec, Max. Disp. = 10.5160-cm
C.2c Series of Deformation History – Maximum Displacement at Center Node Case III (SET2)

Time = 200-μsec,
Max. Disp. = 0.4479-cm

Time = 1000-μsec,
Max. Disp. = 8.5109-cm

Time = 2000-μsec,
Max. Disp. = 12.966-cm
APPENDIX D. BOUNDARY CONDITIONS FOR SANDWICH STRUCTURE AND
APC VEHICLE WITH SANDWICH STRUCTURE
D.1 Boundary Condition for Sandwich Structure

<table>
<thead>
<tr>
<th>Boundary Condition</th>
<th>Tx</th>
<th>Ty</th>
<th>Tz</th>
<th>Rx</th>
<th>Ry</th>
<th>Rz</th>
</tr>
</thead>
<tbody>
<tr>
<td>x-z symmetry plane</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>y-z symmetry plane</td>
<td>✓</td>
<td>✓</td>
<td></td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>y-direction</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
</tbody>
</table>
D.2 Boundary Condition for APC Vehicle with Sandwich Structure

<table>
<thead>
<tr>
<th>Boundary Condition</th>
<th>Tx</th>
<th>Ty</th>
<th>Tz</th>
<th>Rx</th>
<th>Ry</th>
<th>Rz</th>
</tr>
</thead>
<tbody>
<tr>
<td>y-z symmetry plane</td>
<td>√</td>
<td></td>
<td></td>
<td>√</td>
<td>√</td>
<td>√</td>
</tr>
</tbody>
</table>

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
APPENDIX E. DESIGN OF EXPERIMENT RESULTS – ANALYSIS OF VARIANCE (ANOVA) OF EACH RESPONSE
E.1 Kinetic Energy – Each percentage indicates the sensitivity of the internal energy absorption to changes in each design variable.
APPENDIX F. COMPARISON OF RESPONSES FOR UN-DENTED VERSES

DENTED SANDWICH CORE WITH 24 ELEMENTS PER CELL EDGE

Result Comparison: Un-dented vs. Dented (e24)

(Units = g, cm, μs, megabar)

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
APPENDIX G. ACCURACY OF RESULT COMPARISON WITH REFINEMENT OF ELEMENTS ON VEHICLE MODEL

Accuracy of Result with Refinement of Elements on Vehicle Model

![Graph showing the accuracy of result with refinement of elements on vehicle model](image-url)

- Peak Acceleration

Element Size (cm) vs. Peak Acceleration (cm/microsec^2)

- 0.5
- 0.75
- 1
- 1.75
- 2
- 3

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
APPENDIX H. PEAK ACCELERATION COMPARISON AT DIFFERENT TIME INTERVAL FOR CASE I & III

<table>
<thead>
<tr>
<th>Time Interval (micro-sec)</th>
<th>0.01</th>
<th>0.1</th>
<th>1</th>
<th>10</th>
<th>25</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_{peak}$ Case I</td>
<td>7.37E-05</td>
<td>7.37E-05</td>
<td>7.37E-05</td>
<td>7.37E-05</td>
<td>6.32E-05</td>
</tr>
<tr>
<td>$a_{peak}$ Case III</td>
<td>9.30E-05</td>
<td>9.30E-05</td>
<td>1.25E-04</td>
<td>3.82E-05</td>
<td>3.30E-05</td>
</tr>
<tr>
<td>%Δ I → III</td>
<td>-26.2 %</td>
<td>-26.2 %</td>
<td>-70.1 %</td>
<td>48.1 %</td>
<td>47.8 %</td>
</tr>
</tbody>
</table>
REFERENCES


87


VITA

Graduate College
University of Nevada, Las Vegas

Dong Kwan Lee

Local Address:
3851 South Wynn Road. #2038
Las Vegas, Nevada 89103

Home Address:
Kyungki-do SungNam-city YiMea-dong
SamSung Apt. 1004-dong 302-ho
South, Korea

Degree:
Bachelor of Science, Mechanical Engineering, 2002
Indiana Institute of Technology, Fort Wayne, Indiana


Thesis Title: Energy Absorbing Sandwich Structure Under Blast Loading

Thesis Examination Committee:
Chairperson, Dr. Brendan J. O'Toole
Committee Member, Dr. Mohamed Trabia
Committee Member, Dr. Woosoon Yim
Graduate Faculty Representative, Dr. Samman Ladkany

90

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.