THE EFFECT OF SENSOR MASS, SENSOR LOCATION, AND DELAMINATION LOCATION OF
DIFFERENT COMPOSITE STRUCTURES UNDER DYNAMIC LOADING

A Thesis

presented to

The Faculty of California Polytechnic State University,
San Luis Obispo

In Partial Fulfillment

Of the Requirements for the Degree of
Masters of Science in Aerospace Engineering

By

Albert Darien Liu

January 2013

© 2013

Albert Darien Liu

ALL RIGHTS RESERVED

COMMITTEE MEMBERSHIP

TITLE: The Effect of Sensor Mass, Sensor Location, and Delamination Location of Different Composite Structures under Dynamic Loading

AUTHOR: Albert D. Liu

DATE SUBMITTED: January 2013

COMMITTEE CHAIR: Faysal Kolkailah, Ph.D.

COMMITTEE MEMBER: Eltahry Elghandour, Ph.D.

COMMITTEE MEMBER: Eric Kasper, Ph.D.

COMMITTEE MEMBER: Kodi Rider, M.S.

Abstract

The Effect of Sensor Mass, Sensor Location, and Delamination Location of Different Composite Structures under Dynamic Loading

By: Albert D. Liu

This study investigated the effect of sensor mass, sensor location, and delamination location of different composite structures under dynamic loading. The study pertains to research of the use of accelerometers and dynamic response as a cost-effective and reliable method of structural health monitoring in composite structures. The composite structures in this research included carbon fiber plates, carbon fiber-foam sandwich panels, and carbon-fiber honeycomb sandwich panels. The composite structures were manufactured with the use of a Tetrahedron MTP-8 heat press. All work was conducted in the Cal Poly Aerospace Structures/Composites Laboratory. Initial delaminations were placed at several locations along the specimen, including the bending mode node line locations. The free vibration of the composite structure was forced through a harmonic horizontal vibration test using an Unholtz-Dickie shake system. A sinusoidal sweep input was considered for the test. The dynamic response of the composite test specimens were measured using piezoelectric accelerometers. Measurements were taken along horizontal and vertical locations on the surfaces of the composite structures. Square inch grids were marked on the surfaces to create a meshed grid system. Accelerometer measurements were taken at the center of the grids. The VIP Sensors 1011A piezoelectric accelerometer was used to measure vibration response. The measurements were then compared to response measurements taken from a PCB Piezotronics 353B04 single access accelerometer to determine the effects of sensor mass. Deviations in bending mode natural frequency and differences in mode shape amplitude became the criteria for evaluating the effect of sensor mass, sensor location, and delamination location. Changes in damping of the time response were also studied. The experimental results were compared to numerical models created using a finite element method. The experimental results and numerical values were shown to be in good agreement. The sensor mass greatly affected the accuracy and precision of vibration response measurements in the composites structures. The smaller weight and area of the VIP accelerometer helped to minimize the decrease in accuracy and precision due to sensor mass. The effect of sensor location was found to be coupled with the effect of sensor mass and

the bending mode shape. The sensor location did not affect the vibration response measurements when the sensor mass was minimized. Off-center horizontal sensor placement showed the possibility of measuring vibration torsion modes. The effect of delamination changed the bending mode shape of the composite structure, which corresponded to a change in natural frequency. The greatest effect of the delamination was seen at the bending mode node lines, where the bending mode shape was most significantly affected. The effect of delamination was also dependent on the location of the delamination and the composite structure type. The results of this study provided considerations for future research of an active structural health monitoring system of composite structures using dynamic response measurements. The considerations included sensor mass reduction, sensor placement at constraints and bond areas and the presence of damping material.

Keywords: aerospace engineering, composite structures, dynamic response, modal analysis, nondestructive inspection/evaluation (NDI/E), structural analysis, structural health monitoring (SHM)

Acknowledgement

This thesis is dedicated to my family and friends. They are everything to me and they are responsible for the person I am today.

I would like to thank my thesis advisors, Dr. Faysal Kolkailah and Dr. Eltahry Elghandour for giving me the opportunity to conduct research in the field of structural analysis of composite materials. Their support and guidance throughout this project has helped me gain experience and knowledge in the field. I would like to thank my thesis committee members, Dr. Eric Kasper and Kodi Rider, for their time and assistance in reviewing the work and providing feedback to propel this research paper to its fullest potential. I am also very thankful for the support provided by the Cal Poly Aerospace Engineering department.

I would like to thank my friends, my second family. Cameron Chan, Jason Cortez, Jay Lopez, Josiah Mayfield, and Etuate Varea, you are all brothers to me. Together, we all went through the struggles of many academic and personal experiences, yet we were always able to find ways to celebrate our lives to the fullest. I would also like to thank Sara Noii and Yvette Castillo. You two were the staple that held all of us together. I will miss the many home-cooked meals and desserts you ladies provided for us during times of hunger.

I would like to thank my mother, my father, and my sister. They always believed in me, even and especially at times when I didn't believe in myself. My family is the reason I am who I am today. Without my mother, aunts and grandmothers, I would not have found insight on love, care, and consideration for others. Without my father, uncles and grandfathers, I would be without earnest responsibility and a sense of work ethic. Without my sister and many cousins, I would not have been able to practice those mannerisms towards my own kin.

Lastly, I would like to thank God. Without God, I would not have found a remarkable life nor been through an incredible journey.

Table of Contents

List of Tables	x
List of Figures.....	xii
List of Equations	xviii
Nomenclature	xix
Chapter 1. Introduction	1
1.1 Overview of Composite Structures	1
1.2 Structural Health Monitoring	3
1.3 Structural Dynamic Response	4
1.4 Research of Active Sensor Use in Structural Health Monitoring	4
1.5 Main Objective and Scope of the Study	6
Chapter 2. Manufacture of Composite Test Specimens	8
2.1 Material Selection of Composite Structures.....	8
2.1.1 Carbon fiber plates and facesheets	8
2.1.2 Sandwich cores of sandwich panels	9
2.2 Manufacture of Composite Plates and Composite Sandwich Panels	11
2.3 Carbon Fiber Plates.....	15
2.4 Carbon Fiber-Foam Sandwich Panels.....	16
2.5 Carbon Fiber-Honeycomb Sandwich Panels.....	17
Chapter 3. Experimental Test & Methodology	19
3.1 Material and Mechanical Properties	19
3.1.1 LTM45-EL carbon fiber	19
3.1.2 FR-6710 polyurethane foam	23
3.1.3 HexWeb HRH-10 Nomex honeycomb	24
3.2 Determination of the Bending Mode Node Line Locations	24
3.3 Harmonic Horizontal Free Vibration Test	27
3.3.1 Vibration test equipment.....	27
3.3.2 Vibration test overview	29
Chapter 4. Experimental Test Results	31
4.1 Vibration Response Overview.....	31
4.2 Determining Change in Vibration Response.....	36
4.3 Effects of Sensor Mass on Vibration Response	37
4.3.1 1 st Vibration Bending Mode.....	38
4.3.2 2 nd Vibration Bending Mode.....	40
4.3.3 3 rd Vibration Bending Mode	41
4.3.4 Conclusion of the effects due to sensor mass	41
4.4 Effects of Sensor Location on Vibration Response	42

4.4.1 Carbon fiber plates	42
4.4.1.1 1 st vibration bending mode	44
4.4.1.2 2 nd vibration bending mode	46
4.4.1.3 3 rd vibration bending mode	48
4.4.2 Foam sandwich panels.....	51
4.4.2.1 1 st Vibration Bending Mode	52
4.4.2.2 2 nd Vibration Bending Mode	55
4.4.2.3 3 rd Vibration Bending Mode.....	57
4.4.3 Honeycomb sandwich panels	59
4.4.3.1 1 st Vibration Bending Mode	61
4.4.3.2 2 nd Vibration Bending Mode	63
4.4.3.3 3 rd Vibration Bending Mode.....	65
4.4.4 Conclusion of effects due to sensor location	68
4.5 Effects of Delamination Location on Vibration Response	69
4.5.1 Carbon fiber plates	69
4.5.1.1 1 st Vibration Bending Mode	70
4.5.1.2 2 nd Vibration Bending Mode	72
4.5.1.3 3 rd Vibration Bending Mode.....	74
4.5.2 Foam sandwich panels.....	75
4.5.2.1 1 st Vibration Bending Mode	76
4.5.2.2 2 nd Vibration Bending Mode	78
4.5.2.3 3 rd Vibration Bending Mode.....	79
4.5.3 Honeycomb sandwich panels	80
4.5.3.1 1 st Vibration Bending Mode	81
4.5.3.2 2 nd Vibration Bending Mode	82
4.5.3.3 3 rd Vibration Bending Mode.....	83
4.5.4 Conclusion of effects due to delamination location	84
Chapter 5. Numerical Analysis Results	86
5.1 Numerical Analysis Approach & Methodology	86
5.2 Validation of the Control Case Finite Element Models	88
5.2.1 Carbon fiber plate	88
5.2.2 Foam sandwich panel.....	89
5.2.3 Honeycomb sandwich panel	91
5.3 Finite Element Model of Delamination Cases.....	93
5.3.1 Carbon fiber plates	93
5.3.2 Foam sandwich panels.....	95
5.3.3 Honeycomb sandwich panels	98
Chapter 6. Comparison of Experimental Test Results and Numerical Analysis Results	101
6.1 Carbon Fiber Plate Comparison	101

6.1.1 1 st Vibration Bending Mode.....	101
6.1.2 2 nd Vibration Bending Mode.....	103
6.1.3 3 rd Vibration Bending Mode	105
6.2 Foam Sandwich Panel Comparison	106
6.2.1 1 st Vibration Bending Mode.....	107
6.2.2 2 nd Vibration Bending Mode.....	108
6.2.3 3 rd Vibration Bending Mode	110
6.3 Honeycomb Sandwich Panel comparison.....	111
6.3.1 1 st Vibration Bending Mode.....	112
6.3.2 2 nd Vibration Bending Mode.....	113
6.3.3 3 rd Vibration Bending Mode	115
6.4 Conclusion of Experimental and Numerical Comparison.....	116
Chapter 7. Conclusion.....	118
Chapter 8. Future Work.....	120
References	121
Appendix	124
A.1 Vibration Response of Delaminated Composite Structures (Left and Right Sensor Location)	124
A.1.1 Carbon fiber plates.....	124
i. Left	124
i. Right	125
A.1.2 Foam sandwich panels	126
i. Left	126
ii. Right	127
A.1.3 Honeycomb sandwich panels.....	128
i. Left	128
ii. Right	129
A.2 ABAQUS Tutorial	130
A.2.1 Carbon fiber plates.....	130
A.2.2 Foam sandwiches	130
A.2.3 Honeycomb sandwiches	130

List of Tables

Table 2-1. Delamination grid locations of delamination cases for carbon fiber plates.....	16
Table 2-2. Delamination grid locations of delamination cases for foam sandwich panels	17
Table 2-3. Delamination grid locations of delamination cases for honeycomb sandwich test specimens	17
Table 3-1. Node line locations with respect to grid system	26
Table 3-2. Comparison of VIP Sensors 1011A and PCB Piezotronics 353B04.....	29
Table 4-1. Natural frequencies for the composite structures	34
Table 4-2. Maximum response amplitudes for the composite structures	34
Table 4-3. Average sample rates for the composite structures	37
Table 5-1 Material properties used in ABAQUS model	86
Table 5-2 Comparison of natural frequencies for carbon fiber plate test specimens and ABAQUS model	89
Table 5-3 Comparison of natural frequencies for FR-6710 foam test specimen and ABAQUS model.....	90
Table 5-4 Comparison of natural frequencies for foam sandwich panels and ABAQUS model	91
Table 5-5 Comparison of natural frequencies for honey sandwich panels and ABAQUS model.....	92
Table 5-6 Natural frequencies of delamination cases for carbon fiber plate ABAQUS models.....	94
Table 5-7 Natural frequencies of delamination cases for foam sandwich panel ABAQUS models	96
Table 5-8 Natural frequencies of delamination cases for honeycomb sandwich panel ABAQUS models	99
Table 6-1 Comparison of 1 st bending mode natural frequencies of experimental results and ABAQUS model for carbon fiber plates.....	102
Table 6-2 Comparison of 2 nd bending mode natural frequencies of experimental results and ABAQUS model for carbon fiber plates.....	103
Table 6-3 Comparison of 3 rd bending mode natural frequencies of experimental results and ABAQUS model for carbon fiber plates.....	105
Table 6-4. Comparison of 1 st bending mode natural frequencies of experimental results and ABAQUS model for foam sandwich panels.....	107
Table 6-5. Comparison of 2 nd bending mode natural frequencies of experimental results and ABAQUS model for foam sandwich panels.....	109

Table 6-6 Comparison of 3 rd bending mode natural frequencies of experimental results and ABAQUS	
model for foam sandwich panels.....	110
Table 6-7 Comparison of 1 st bending mode natural frequencies of experimental results and ABAQUS	
model for honeycomb sandwich panels.....	112
Table 6-8 Comparison of 2 nd bending mode natural frequencies of experimental results and ABAQUS	
model for honeycomb sandwich panels.....	114
Table 6-9 Comparison of 3 rd bending mode natural frequencies of experimental results and ABAQUS	
model for honeycomb sandwich panels.....	115

List of Figures

Figure 1.1. Lockheed Martin X-55A utilizing advanced composite materials ^[2]	1
Figure 1.2. Composite sandwich panel utilizing facesheets and core sandwich structure ^[3]	2
Figure 1.3. Abstract active sensors used to detect operational loading on an F-22 Raptor ^[5]	3
Figure 2.1. LTM45 pre-preg carbon fiber (black) selected for the composite structures	9
Figure 2.2. Last-A-Foam foam (a) and Hexcel Nomex honeycomb (b) selected as the sandwich core materials.....	10
Figure 2.3. Prepared set-up of materials used for lay-up process of honeycomb sandwich test specimen	11
Figure 2.4. Cure preparation lay-out of foam sandwich test specimen	12
Figure 2.5. Tetrahedron heat press in the Cal Poly Aerospace Structures/Composites laboratory	12
Figure 2.6. LTM45-EL woven bidirectional pre-preg carbon fiber cure cycle	13
Figure 2.7. An example of the delamination locations for each delamination case of a composite structure.....	15
Figure 2.8. Carbon fiber plate solid model of the D1 delamination case	15
Figure 2.9. Foam sandwich panel solid model of the D1 delamination case	16
Figure 2.10. Carbon fiber honeycomb sandwich solid model of the D3 delamination case	18
Figure 3.1. Tensile test of carbon fiber test coupons. Shown is the aftermath of a tensile test.	20
Figure 3.2. Test results of ASTM D3039 tensile test for LTM-45EL carbon fiber	20
Figure 3.3 Vishay strain gage mounted on a test coupon (left) and the resulting Poisson's ratio test (right)	22
Figure 3.4. Burn test of LTM-45EL carbon fiber using a Thermolyne 1300 furnace	22
Figure 3.5. Compressive testing for Last-A-Foam FR-6710 polyurethane foam conducted by Rider.....	23
Figure 3.6. MB magnetic shake table (right) and function generator, amplifier, and frequency meter (left)	25
Figure 3.7. Vertical vibration test to determine node line locations.....	25
Figure 3.8. Node line locations for 1 st natural frequency (left), 2 nd natural frequency (middle), 3 rd natural frequency (right)	26
Figure 3.9. Unholtz-Dickie shake table used for forced harmonic horizontal vibration test	27
Figure 3.10. The PCB 353B04 accelerometer (left) and the VIP 1011A accelerometer (right).....	28

Figure 3.11. Vibration test set-up with the PCB accelerometer (left) and VIP accelerometer (right) on	
04-02 grid location	30
Figure 4.1. Vibration response plot of control carbon fiber plate for various middle vertical grid points.....	32
Figure 4.2. 1 st , 2 nd , and 3 rd bending modes of a typical fixed-free vibration specimen (top to bottom).....	32
Figure 4.3. Vibration response plot of control honeycomb sandwich panel for various middle vertical	
grid points.....	33
Figure 4.4. Half-power bandwidth method used to determine damping ratios	35
Figure 4.5. 1 st mode natural frequencies and response amplitudes of foam sandwich panels for the VIP	
and PCB accelerometers.....	39
Figure 4.6. 1 st mode time response of foam sandwich panels for the VIP and PCB accelerometers.....	39
Figure 4.7. 2 nd mode natural frequencies and response amplitudes of foam sandwich panels for the VIP	
and PCB accelerometers.....	40
Figure 4.8. 3 rd mode natural frequencies and response amplitudes of foam sandwich panels for the VIP	
and PCB accelerometers.....	41
Figure 4.9. Vibration response plot of control carbon fiber plates for middle horizontal locations	43
Figure 4.10. Vibration response plot of control carbon fiber plates for left horizontal locations	43
Figure 4.11. Vibration response plot of control carbon fiber plates for right horizontal locations	44
Figure 4.12. 1 st bending mode natural frequencies and response amplitudes of the carbon fiber plates for	
vertical and horizontal grid locations	44
Figure 4.13. 1 st mode time response of carbon fiber plates for horizontal grid locations at 10" vertical	
grid locations.....	45
Figure 4.14. 1 st mode time response of carbon fiber plates for vertical grid locations	46
Figure 4.15. 2 nd bending mode natural frequencies and response amplitudes of the carbon fiber plates	
for vertical and horizontal grid locations	47
Figure 4.16. 2 nd mode time response of carbon fiber plates for horizontal locations at 10" vertical grid	
locations	48
Figure 4.17. 2 nd mode time response of carbon fiber plates for vertical grid locations	48

Figure 4.18. 3 rd bending mode natural frequencies and response amplitudes of the carbon fiber plates for vertical and horizontal locations.....	49
Figure 4.19. 3 rd mode time response of carbon fiber plate for horizontal locations at 10" vertical grid locations	50
Figure 4.20. 3 rd mode time response of carbon fiber plate for vertical grid locations.....	50
Figure 4.21. Vibration response plot of control foam sandwich panels for middle horizontal locations.....	51
Figure 4.22. Vibration response plot of control foam sandwich panels for left horizontal locations	52
Figure 4.23. Vibration response plot of control foam sandwich panels for right horizontal locations	52
Figure 4.24. 1 st bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal grid locations	53
Figure 4.25. 1 st mode time response of foam sandwich panels for horizontal grid locations at 10" vertical grid locations.....	54
Figure 4.26. 1 st mode time response of foam sandwich panels for vertical grid locations	54
Figure 4.27. 2 nd bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal grid locations	55
Figure 4.28. 2 nd mode time response of foam sandwich panels for horizontal locations at 10" vertical grid locations	56
Figure 4.29. 2 nd mode time response of foam sandwich panels for vertical grid locations	56
Figure 4.30. 3 rd bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal locations.....	57
Figure 4.31. 3 rd mode time response of foam sandwich panels for horizontal locations at 10" vertical grid locations	58
Figure 4.32. 3 rd mode time response of foam sandwich panels for vertical grid locations.....	59
Figure 4.33. Vibration response plot of control honeycomb sandwich panels for middle horizontal locations	59
Figure 4.34. Vibration response plot of control honeycomb sandwich panels for left horizontal locations	60
Figure 4.35. Vibration response plot of control honeycomb sandwich panels for right horizontal locations	60

Figure 4.36. 1 st bending mode natural frequencies and response amplitudes of the honeycomb sandwich panels for vertical and horizontal grid locations	61
Figure 4.37. 1 st mode time response of honeycomb sandwich panels for horizontal locations at 10" vertical grid locations.....	62
Figure 4.38. 1 st mode time response of honeycomb sandwich panels for vertical grid locations	63
Figure 4.39. 2 nd bending mode natural frequencies and response amplitudes of the honeycomb sandwich panels for vertical and horizontal grid locations	63
Figure 4.40. 2 nd mode time response of honeycomb sandwich panels for horizontal locations at 10" vertical grid locations.....	64
Figure 4.41. 2 nd mode time response of honeycomb sandwich panels for vertical grid locations	65
Figure 4.42. 3 rd bending mode natural frequencies and response amplitudes of the honeycomb sandwich panel for vertical and horizontal locations	66
Figure 4.43. 3 rd mode time response of honeycomb sandwich panel for horizontal locations at 10" vertical grid locations.....	67
Figure 4.44. 3 rd mode time response of honeycomb sandwich panel for vertical grid locations	67
Figure 4.45. 1 st bending mode natural frequency deviations and response amplitude differences of carbon fiber plates for delamination cases.....	71
Figure 4.46. 1 st mode time response of delamination cases for the carbon fiber plate at the 10" grid location.....	72
Figure 4.47. 2 nd bending mode natural frequency deviations and response amplitude differences of carbon fiber plates for delamination cases.....	73
Figure 4.48. 3 rd bending mode natural frequency deviations and response amplitude differences of carbon fiber plates for delamination cases.....	75
Figure 4.49. 1 st bending mode natural frequency deviations and response amplitude differences of foam sandwich panels for delamination cases	76
Figure 4.50. 1 st bending mode time response of foam sandwich panels for delamination cases.....	78
Figure 4.51. 2 nd bending mode natural frequency deviations and response amplitude differences of foam sandwich panels for delamination cases	79

Figure 4.52. 3 rd bending mode natural frequency deviations and response amplitude differences of foam sandwich panels for delamination cases	80
Figure 4.53. 1 st bending mode natural frequency deviations and response amplitude differences of honeycomb sandwich panel for delamination cases	81
Figure 4.54. 2 nd bending mode natural frequency deviations and response amplitude differences of honeycomb sandwich panels for delamination cases	82
Figure 4.55. 3 rd bending mode natural frequency deviations and response amplitude differences of honeycomb sandwich panels for delamination cases	83
Figure 5.1. ABAQUS model of 1 st , 2 nd , and 3 rd vibration bending modes of carbon fiber plates (left to right)	89
Figure 5.2. ABAQUS model of 1 st , 2 nd , and 3 rd vibration bending modes of foam sandwich panels (left to right)	91
Figure 5.3. ABAQUS model of 1 st , 2 nd , and 3 rd vibration bending modes of honeycomb sandwich panels (left to right)	92
Figure 5.4. Amplitude differences of delamination cases for the 1 st , 2 nd , and 3 rd bending modes of the carbon fiber plate ABAQUS models (top to bottom)	95
Figure 5.5. Delamination present in the 3 rd bending mode shape of the D3 delamination case of the foam sandwich panel ABAQUS model.....	96
Figure 5.6. Amplitude differences of delamination cases for the 1 st , 2 nd , and 3 rd bending modes of the foam sandwich panel ABAQUS models (top to bottom)	98
Figure 5.7. Amplitude differences of delamination cases for the 1 st , 2 nd , and 3 rd bending modes of the honeycomb sandwich panel ABAQUS models (top to bottom)	100
Figure 6.1. Comparison of 1 st bending mode amplitude differences of experimental results and ABAQUS model for carbon fiber plates.....	102
Figure 6.2. Comparison of 2 nd bending mode amplitude differences of experimental results and ABAQUS model for carbon fiber plates.....	104
Figure 6.3. Comparison of 3 rd bending mode amplitude differences of experimental results and ABAQUS model for carbon fiber plates.....	106

Figure 6.4. Comparison of 1 st bending mode amplitude differences of experimental results and ABAQUS	
model for foam sandwich panels.....	108
Figure 6.5. Comparison of 2 nd bending mode amplitude differences of experimental results and ABAQUS	
model for foam sandwich panels.....	109
Figure 6.6. Comparison of 3 rd bending mode amplitude differences of experimental results and ABAQUS	
model for foam sandwich panels.....	111
Figure 6.7. Comparison of 1 st bending mode amplitude differences of experimental results and ABAQUS	
model for honeycomb sandwich panels.....	113
Figure 6.8. Comparison of 2 nd bending mode amplitude differences of experimental results and ABAQUS	
model for honeycomb sandwich panels.....	114
Figure 6.9. Comparison of 3 rd bending mode amplitude differences of experimental results and ABAQUS	
model for honeycomb sandwich panels.....	116

List of Equations

Equation 3-1. Poisson's ratio	22
Equation 3-2. Fiber volume fraction	23
Equation 4-1. Damping ratio through half-power bandwidth method	35
Equation 4-2. Time-domain response amplitude	36

Nomenclature

	<u>Units</u>
D – node line delamination case	--
E – elastic modulus	psi
F – force	lb _f
G – shear modulus	psi
I – area moment of inertia	in ⁴
K – stiffness	lb _f /in
L – location delamination case	--
M – moment	lb _f *in
P – load	lb _f
R – crosshead loading rate	in/min
V – volume fraction	%
W – weight fraction	%
X – longitudinal direction	--
Y – transverse direction	--
Z – minimum outer fiber straining rate	in/in*min
a – acceleration	ft/s ²
f – frequency	Hz
l – length	in
m – mass	lb _m
t – time	s
t – face thickness	in
u – time domain amplitude	g
α – shear stress nonlinear term after yielding has occurred	--
β – weighing factor for shear term in tensile fiber mode	--
δ – displacement	in
ε – strain	in/in
ζ – damping ratio	--
ν – Poisson's ratio	--
ρ – density	lb _f /in ³
σ – stress	psi
ω – circular frequency	rad/s

Chapter 1. Introduction

The following section provides a brief overview of composite structures and structural health monitoring of composite materials. Structural dynamic response is also reviewed and applications to structural health monitoring are presented. Previous and current research efforts, including those at the Cal Poly Aerospace Structures/Composites laboratory, are also examined. Finally, the objective of the research is defined and the scope of the research is addressed.

1.1 Overview of Composite Structures

Composite materials exist in every aspect of the world. Every manufacturing industry around the world utilizes some type of composite material. From plywood in shutter boxes to carbon fiber on aircraft tooling, composite materials are everywhere. Advanced composite materials are described as advanced materials composed of two or more substances with overall high strength-to-weight characteristics. Many advanced aircraft are constructed of fiber-reinforced polymer (FRP) composites not only for their relatively high strength and stiffness to weight characteristics, but also for their low-cost for building complex structures. [1] A program manager from the Lockheed Martin X-55A Advanced Composite Cargo Aircraft was once quoted, "The design approach to [the] new fuselage resulted in a 90% reduction in the number of parts and fasteners

Fig. 1.1. Lockheed Martin X-55A utilizing advanced composite materials.[2]

required [along with a] reduction in production fabrication and assembly hours of more than 70% [and] weight savings on the order of 20-30%.”^[2]

One of the main differences between composite aircraft and traditional metallic aircraft is the frequent use of composite sandwich structures to form rigid surfaces instead of reinforced metallic skins. A composite sandwich structure, shown in Fig. 1.2, usually comprises of thin facesheet skins bonded around a lightweight core. Composite sandwich structures are known for excellent bending performance, where the facesheet skins take the compressive and tensile forces and the takes the shear loading through the thickness.

This configuration presents an efficient cross-sectional moment of inertia and results in higher bending stiffness and strength per mass. Though composite sandwich structures may introduce a larger structure, less reinforcement is needed to maintain rigid surfaces, producing a lower part count, fewer connections, and overall a more efficient structure.

Fig. 1.2. Composite sandwich panel utilizing facesheets and core sandwich structure.^[3]

Although composite structures present greater strength, weight, and manufacturing benefits over traditional metallic structures, the higher brittleness of composite structures introduce complexity in damage detection. Full acceptance of composite structure use is hindered by the tendency of internal damage modes to propagate with little or no outward indication. Impacts to the surface of composite structures can cause disbanding between plies or between skin-core interfaces. The damage is usually unseen at the surface and can create catastrophic failures. The availability of damage detection in composite structures is important for the confidence of use in all industries.

1.2 Structural Health Monitoring

Systems utilizing composite structures in the aerospace industry currently require regular nondestructive inspection for damage detection. Structural health monitoring (SHM) is a growing field dedicated to system structural integrity. SHM techniques consist of passive SHM and active SHM. Passive SHM techniques involve various mapping techniques to obtain information about usage and possible structural damage. Acoustic emission is the one of the best known passive SHM technique. Acoustic emission relates sound wave production to elastic waves generated by mechanical stress. Passive SHM techniques have been studied for a longer period of time; however are limited to continuous monitoring.^[4]

Active SHM techniques utilize actuation or excitation of the monitored structure and measure the resulting response. Ultrasonic testing is one of the most widely used techniques in damage detection of composite structures. However the requirement of point-by-point measurements in ultrasonic testing can become time-consuming and expensive for large composite structures. The integration of active piezoelectric sensors on aircraft to detect operational loading is currently being explored. Fig. 1.3 shows an example of the technique. The use of active sensors on aircraft in the field of operation may present a low-cost and efficient method of detecting damage. A robust method of damage detection with the use of existing accelerometers on an aircraft can introduce lower life cycle cost through longer periods between structural health monitoring.

Fig. 1.3. Abstract active sensors used to detect operational loading on an F-22 Raptor.^[5]

1.3 Structural Dynamic Response

Vibration is a mechanical phenomenon where dynamic oscillations occur about points of equilibrium. Dependent on the magnitude of vibration and the mechanical properties of the materials, vibration can be a very dangerous dynamic loading case. Vibration analysis provides a much more difficult problem when compared to static loading analysis, as the analysis of the structure becomes less simple and degrees of freedom can become coupled. Free vibrations can be implemented to study the response characteristics of a given mechanical system.

Mechanical resonance is one of the behaviors studied in the vibration response of a mechanical system. At a resonance vibration frequency, also known as the natural frequency, large deviations in response amplitude occur. When high response amplitudes and high oscillation rates are coupled in the mechanical system, catastrophic failures can occur. Resonance frequencies are studied to obtain the response behaviors and damping properties of a structure. ^{[6] [7]}

Accelerometers are the main devices used in most methods of dynamic response measurement. An accelerometer presents a sensor to receive a stimulus from the dynamic response and a transducer to convert the stimulus into a form of known form of measurement. Piezoelectric materials are widely used in accelerometers. Piezoelectric materials present the ability to relate mechanical strain from an external force to an electric charge. With the charge measured, further relations can be made to calculate the dynamic response of a system with the measurements from the piezoelectric material. Piezoelectric sensors provide a means of measuring the dynamic response of a structure and relating the measurements to possible forms of failure mode detection. ^[8]

1.4 Research of Active Sensor Use in Structural Health Monitoring

Previous research has shown prevalence in the use of accelerometer measurements as a means of structural health monitoring. Within the past decade, research efforts have been made to create a structural health monitoring system for use in field operation of aerospace structures. At the Massachusetts Institute of Technology, Kessler and Spearing surveyed various experimental and analytical methods for in-situ damage detection of composite materials using piezoceramic sensors. ^[9] The methods included modal

analysis, Lamb wave measurements, acoustic emission, and strain monitoring techniques to detect various damage modes. The modal analysis and Lamb wave techniques implemented conformable sensors on the surfaces and could be coupled with other methods. A development of an in-flight structural health monitoring system for composite unmanned aircraft is ongoing at the University of California, San Diego. Kosmatka and Oliver conducted research in the use of piezoelectric patches and fiber optics strain gauges to measure operational loads and structural damage.^[10] With both low frequency and high frequency modal data, measurements could be made to find regions needed further inspection. Most recently, the use of active piezoelectric-wafer sensors was applied to structural health monitoring of composite structures. A study of the use was conducted by Giugliutiu and Santoni-Bottai at the University of South Carolina in 2011.^[11] An ability to detect damage was correlated with the measurements of the active sensors. Damage detection was found in 40cm x 5cm carbon fiber strips when a minimum hole diameter of 0.8mm was present. However, delamination detection was unsuccessful in the study.

A case can be made for the design of a simple method of structural health monitoring in composite structures. The design may not require high accuracy and higher order methods of detection; however robustness would be required. With the ability to detect rough estimates of damage, the design can reduce lifecycle cost of composite structures by better estimating the need and schedule of structural inspection. Preliminary research must first be completed before a possible design can be organized.

This research investigates behavior of dynamic response measurements in the presence of damage and can be applied to the design of a structural health monitoring system for composite structures. This research uses the continuation of previous research effects at the California Polytechnic State University (Cal Poly), San Luis Obispo in the design of composite structures. Previous research efforts were thoroughly studied to obtain further information of experimental mechanical behaviors. An initial study of vibration response of composite structures was conducted at Cal Poly by Lopez, who studied the free vibration of aluminum plates and composite laminated plates with initial damage. The effect of cracks and discontinuities in dynamic loading was determined for the change in resonance and amplitude. The nondestructive tests provided substantial changes in the modal parameters and a capability of fault detection. The initial damages produced high effects and presented small errors in the damage detection capabilities.^[12]

Saguibo determined the change in vibration characteristics of composite laminated plates in the presence of a delamination. Initial delaminations were placed at several locations to provided difference delamination cases and delamination location cases. Significant findings occurred in the 2nd and 3rd bending modes of the plates. The delamination was correlated with the changes in resonance frequency and amplitude. A resulting change in damping characteristics was also found.^[13]

Tremaine developed an analytical solution for the modal analysis of orthotropic composite plates. The analytical solution provided a means of correlating a change in damping characteristics with a change in numerical model. The damping materials implemented in the composite plates provided changes in damping characteristics and the coupled resonance response.^[14]

Rider discovered the effects of damage arrestment devices (DADs) on the structural response of composite plates and composite sandwiches. The DADs greatly improved vibration and flexural characteristics of delaminated and impacted composite specimens. The resonance frequencies, amplitudes, and resultant damping ratios in impacted composites greatly improved with the use of DADs. Rider also developed an improved data acquisition system for the Dynatup impact machine in the Cal Poly Aerospace Structures/Composites Laboratory and helped open the opportunity for further research.^[15]

1.5 Main Objective and Scope of the Study

The objective of this study was to investigate the effects of sensor mass, sensor location, and delamination location on the dynamic response of composite structures. The free vibration of the composite structures was forced with a harmonic horizontal vibration test, employing a sinusoidal sweep input. Carbon fiber plates, carbon fiber-foam sandwich panels, and carbon fiber-honeycomb sandwich panels were manufactured as the composite structures. Initial delaminations were placed at several locations along the composite structure. A piezoelectric accelerometer was used to measure the dynamic response of the composite structure at several locations along the vertical and horizontal dimensions of the surface area. Deviations in bending mode natural frequency and differences in mode shape amplitude were the criteria for evaluating the effect of sensor mass, sensor location, and delamination location.

This paper discusses the manufacturing methods, experimental test methodologies, experimental test results, and numerical results, and results comparison of the study. Chapter 2 discusses the manufacture method used to create the composite test specimens and justification for the selected method. The selection of materials is also discussed.

Chapter 3 provides details of the experimental test and methodology in this study. Selection of equipment and corresponding specifications are mentioned. The Unholtz-Dickie shake system and experimental test procedures are described in this chapter.

Chapter 4 lists the experimental test results for all composite specimen types. The effects due to sensor mass, sensor location, and delamination location in the experimental test specimens are reasoned in this chapter.

Chapter 5 discusses the numerical analysis method selected to create numerical models. The results of the numerical model are detailed. Justification for the assumptions in the numerical models is discussed.

Chapter 6 compares the results of the experimental test specimens and numerical model. Similarities and correlations between the experiment and numerical models are highlighted.

Chapter 7 addresses the conclusions of this research and presents the results of all trends.

Chapter 2. Manufacture of Composite Test Specimens

This section discusses the method used to manufacture the composite structures. Three specimen types were created: one composite plate and two types of composite sandwich panel. Carbon fiber was the selected material for the plates and the facesheets of the sandwich panels. Rigid polyurethane foam and Nomex honeycomb were selected as core material for the sandwich panels. A Tetrahedron heat press was utilized to cure the composite structure at the recommended manufacture cure cycle. A 3"x10" mesh grid system was placed on the surface of the composites structures to detail the locations of the accelerometer placement. Initial delaminations were placed at specific locations in the composite structure prior to specimen cure. Careful attention was placed in the manufacture of the composite structures to ensure high precision and accuracy and to minimize manufacturing error.

2.1 Material Selection of Composite Structures

2.1.1 Carbon fiber plates and facesheets

Carbon fiber was selected as the material for the composite plates and for the face sheets of the composite sandwich plates. Carbon fiber was selected for its high strength-to-weight ratio and application to the aerospace industry. Carbon fiber-resin cloth and pre-impregnated carbon fiber were the two types considered. Pre-impregnated carbon fiber appeared to be a more viable choice because of its ease of manufacture and higher quality. A consistent fiber-to-matrix ratio would provide for regularity in quality of carbon fiber, as well as reduction in human error during the lay-up process. The pre-impregnated nature also decreases manufacture time, since the face sheets can be cut to specific dimensions and simply layered as plies.

Advanced Composites Group (ACG) pre-impregnated LTM45-EL matrix with CF1803 bidirectional woven carbon fiber was the type of carbon fiber selected.^[16] Fig. 2.1 shows the carbon fiber selected. The LTM45EL carbon fiber was selected in the 0.015" thickness range. The pre-impregnated carbon fiber was readily available in the Cal Poly Aerospace Structures/Composites laboratory. The LTM45 carbon fiber was expiring material generously donated from Boeing. Because the donated material was no longer able to meet

Fig. 2.1. LTM45 pre-preg carbon fiber (black) selected for the composite structures.

aerospace production quality standards, the material properties and mechanical properties of the carbon fiber did not match the specifications provided by the supplier. The deviations in material properties and mechanical properties were found and will be discussed in the next chapter.

2.1.2 Sandwich cores of sandwich panels

The selection of the sandwich material dictated the application of the research. Polyurethane foam and Nomex honeycomb were selected as the sandwich core materials because of the high feasibility and practicality of the two materials. The foam was selected to provide a rigid core with relatively high damping properties. The foam core also provided full bond surface contact with the face sheets. The honeycomb was selected for its relatively lighter weight and higher flexibility. The honeycomb core did not provide same surface contact as the foam core due to the cell geometries. Both rigid polyurethane foam and Nomex honeycomb are widely used as core material for composite sandwiches in the aerospace industry. For aircraft that utilize composite material, foam and honeycomb sandwiches are manufactured as control surfaces and panels.

General Plastics Last-A-Foam FR-6710 rigid polyurethane foam was selected as the core material for the foam sandwich panels.^[17] Fig. 2.2(a) shows the selected foam. The FR-6710 foam belongs to the FR-6700 aerospace grade series of foam and has a corresponding density of 10lb/ft³. The 0.5" thickness range was the class of FR-6710 foam used in manufacturing the foam sandwiches. The foam material was readily available in sheets in the Cal Poly Aerospace Structures/Composites laboratory. Each foam sheet was inspected to

ensure quality of the sandwich panel. Some foam sheets had previous cracks or abrasions from mishandling or general wear and tear. Only foam sheets without these imperfections were selected and in the foam sandwich panels.

Fig. 2.2. Last-A-Foam foam (a) and Hexcel Nomex honeycomb (b) selected as the sandwich core materials.

Hexcel HexWeb HRH-10-1/8-3.0 Nomex Aramid Fiber/Phenolic Resin Honeycomb was selected as the core material for the honeycomb sandwich test specimens.^[18] Sections of honeycomb core can be seen in Fig. 2.2(b). The honeycomb was manufactured with Nomex aramid fiber sheets bonded at the nodes with phenolic resin to create the cellular geometry. The name of the HexWeb honeycomb was designated as X-Y-Z, where X is the type of HexWeb honeycomb, Y is the cell size in inches, and Z is the nominal density in pounds per cubic foot. The HRH-10 honeycomb is one of the most common and basic aerospace grade honeycomb. The hexagonal cell class of the HRH-10 honeycomb was selected. The HRH-10 is a very common honeycomb used in the aerospace industry and was readily available in the Cal Poly Aerospace Structures/Composites laboratory. The 0.5" thick class honeycomb sheets were selected to match the thickness of the foam sheets. Similar to the foam, some honeycomb sheets came with imperfections, such as cracks and abrasions. Only quality sheets of honeycomb were used for the sandwich panels. Because of the weakness due to shear in the cell thickness direction, special care was taken in the manufacture of the honeycomb sandwich panels.

Material and mechanical characteristics of the foam and honeycomb material were found through testing and compared to the manufacturer's specifications. The results are discussed in the next chapter.

2.2 Manufacture of Composite Plates and Composite Sandwich Panels

The composite structures were lay-up by hand. This resulted in the need for careful accuracy and precision during the lay-up process. The carbon fiber was cut to 3.5"x12.5" plies to allow for proper margins in obtaining the desired 3"x11.5" cross section area of each composite structure. Four plies of carbon fiber were used for the composite plates. The composite sandwich panels also utilized four plies of carbon fiber, with two plies on each side of the core to create a symmetric sandwich panel. Drafting triangles and rulers were used to ensure that each ply of carbon fiber was cut with correct dimensions and perpendicular corners. The sandwich cores were also cut to 3.5"x12.5" cross sections. Level rulers were used to ensure flat surfaces and flat edges on the foam cores. The edges of the honeycomb were cut in the middle of the cells to keep symmetric cell grids throughout the core. Once all the plies and cores were cut, the composite structures was laid up by hand. Fig. 2.3 shows a prepared set-up of the lay-up process of a honeycomb sandwich panels. The carbon fiber plies were laid onto each other and evenly pressed with a pin-roller. For the sandwich panels, each two ply face sheet of carbon fiber was laid onto each side of the sandwich core and again pressed evenly. Initial delaminations were placed in the composite structures to create the delamination cases. A 1"x3" strip of Airtech Release Ease 234 TFNP non-porous peel ply release fabric was carefully placed to create a delamination. For the plate specimen, the delamination was placed symmetrically between the plies. For the

Fig. 2.3. Prepared set-up of materials used for lay-up process of honeycomb sandwich test specimen.

sandwich panels, the delamination was placed between the core and face sheet interface on one side of the sandwich panel. The specific locations of the delaminations will be discussed at the end of this section.

Once the composite test specimen lay-up was finished, peel-ply release fabrics were placed on each side of the test specimen to prepare for the cure cycle. Fig. 2.4 shows a diagram of the cure prep set-up for the

Fig. 2.4. Cure preparation lay-out of foam sandwich test specimen.

composite sandwich. Airtech Release Ease 234 peel ply release fabrics were selected as the mold release material. Both porous and non-porous types of peel ply were used. A layer of the porous peel ply was placed on each side of the composite test specimen. Non-porous peel ply was placed on each side of the test specimen, outside of the porous material. This release system allowed the resin to catch onto the porous material without touching any tooling or molds and seeping through the non-porous material. The peel ply release system provided for a user-friendly mold release system.

Several methods of curing were considered for the composite plates and sandwiches. Because pre-preg carbon fiber was selected, heat was necessary in the cure cycle. The temperature ranges of the cure cycle can restrict the selection of the sandwich core material; however, for this research, the temperature ranges of the cure cycle were well within the temperature limits of both the selected foam and honeycomb material.

A Tetrahedron MTP-8 heat press, shown in Fig. 2.5, was selected to manufacture the test specimens.^[19] The heat press was available in the Cal Poly Aerospace Structures/Composites laboratory. The Tetrahedron heat press is a simple and reliable machine used to manufacture composite specimen with flat cross

Fig. 2.5. Tetrahedron heat press in the Cal Poly Aerospace Structures/Composites laboratory.

sectional areas. The main restriction of the heat press was that only flat composite pieces could be manufactured. This restriction was actually beneficial to the manufacture of the rectangular sectioned test specimens. The MTP-8 heat press had the ability to reach cure temperatures of 600°F and compression forces of 1000lb. The vacuum chamber model of the heat press was not available for use and the test specimen were cured without vacuum. The MTP-8 heat press included an 8"x8" platen size for the cure cycle; however tooling was developed to create a 12.5"x12.5" platen size. Previous research had shown that the created tooling has a very minimal effect on quality of part.

Two 12"x14"x0.375" aluminum 6061 plates were manufactured to use as tooling for the heat press. The plates allowed for the heat press to manufacture a specimen surface area of 12"x14", rather than the given 8"x8" area. The test specimen lay-up was placed between the two aluminum plates and the two plates were lined up and placed in between the fixtures of the heat press. The plates were aligned to be centered with the fixtures to ensure uniform force distribution.

The manufacturer's cure cycle of the LTM-45 pre-preg carbon fiber was used to cure the composite test specimen. The recipe was set as a cure cycle program on the heat press. Shown in Fig. 2.6 is the specified cure cycle. During the entire cycle, the test specimen was pressed with a compressive force of 1000lb. The compressive force ensured that the interface between each carbon fiber ply and between each carbon fiber face and foam face would bond accordingly. The test specimen was heated from ambient room temperature

Fig. 2.6. LTM45-EL woven bidirectional pre-preg carbon fiber cure cycle.

to 150°F at a rate of 5°F/minute. Once the test specimen reached the cure temperature of 150°F, the temperature was held constant for 14 hours. At the end of the cure time, the material was brought down to a temperature of 70°F at a rate of 2°F/minute. Once the material and tooling reached 70°F, the temperature was held for 10 minutes and the cure cycle was finished. The test specimen manufacture process was repeated for each specimen type and each delamination case. Five test specimens were manufactured for the control specimen types to allow for variance in the test data.

Once the composite piece was manufactured, a 3"x11.5" test specimen was cut from the composite piece with the use of a tile saw. Again, drafting triangles and level rulers were used to ensure for accuracy and precision in cutting. For the delamination cases, the dimensions were measured and cut dependent on the location of the delamination strip. This allowed for minimal error in the delamination location accuracy. When the test specimen was cut to the specified dimension, a 10"x3" grid system with 30 discretized 1"x1" grid sections was marked on the surface of the cured plates. For the sandwich test specimen, the grid system was marked on the face with the delamination. Center points were marked inside each grid section for the accelerometer placement locations. The start of the grid was located 1.5" from one end of the plate. That end was the clamped end of the test specimen. A 0.5" clearance between the clamped end and the start of the grid system was left to account for the interference of the accelerometer wire. Syntax used for grid sections followed as given: vertical location–horizontal location. For example, a grid section at a 4" vertical grid and a left side horizontal grid was referred to as 04-01. Vertical 01 grid sections were also known as left horizontal grids, 02 as middle horizontal grids, and 03 as horizontal right side grids.

The delaminations were placed at specified locations along the vertical length of the composite structure. Six delamination cases were created: a delamination at the 1st vibration bending mode node line (D1), a delamination at the 2nd vibration bending mode node line (D2), delaminations at the 3rd vibration bending mode node lines (D3), a delamination at the 2" horizontal grids (L2), a delamination at the 3" horizontal grids (L3), and a delamination at the 5" horizontal grids (L5). Fig. 2.7 shows a depiction of the delamination locations.

Fig. 2.7. An example of the delamination locations for each delamination case of a composite structure.

The delamination cases were referred to by the node line locations or grid line locations (ie: the delamination at the 2nd node line is referred to as Delamination D2 and the delamination at the 2" horizontal grids is referred to as Delamination L2). The node line location determination is discussed in the next chapter. For the higher order bending modes, delaminations were not placed at the 1st node line. For example, the D2 case included only one node line delamination at the 2nd node line and did not include a delamination at the 1st node line. The D3 case was the only case to include two node line delaminations (at the 2nd and 3rd node lines) and also did not include a delamination at the 1st node line.

2.3 Carbon Fiber Plates

The carbon fiber plates were laid-up using 4 plies of woven bidirectional carbon fiber weave for each test specimen. Fig. 2.8 shows a diagram of a delamination case of the carbon fiber plate. All delamination cases included a 1" delamination strip across the horizontal. Six delamination cases were created: a delamination at the 1st vibration bending mode node line, a delamination at the 2nd vibration bending mode node line, delaminations at the 3rd vibration bending mode node lines, a delamination at the 2" horizontal grids, a delamination at the 3" horizontal grids, and a delamination at the 5" horizontal grids. The delamination cases were referred to by the

Fig. 2.8. Carbon fiber plate solid model of the D1 delamination case.

node line locations or grid line locations (i.e.: the delamination at the 2nd node line is referred to as Delamination D2 and the delamination at the 2" horizontal grids is referred to as Delamination L2). The node line location determination is discussed in the next chapter. For the higher order bending modes, delaminations were not placed at the 1st node line. For example, the D2 case included only one node line delamination at the 2nd node line and did not include a delamination at the 1st node line. The D3 case was the only case to include two node line delaminations (at the 2nd and 3rd node lines) and also did not include a delamination at the 1st node line. Table 2-1 shows the locations of the delaminations for the plates test specimens.

Table 2-1. Delamination grid locations of delamination cases for carbon fiber plates

Delamination case	D1	D2	D3		L2	L3	L5
Grid location	0.5"	8"	5"	9"	2"	3"	5"

2.4 Carbon Fiber-Foam Sandwich Panels

The carbon fiber-foam sandwich panels were laid-up using 2 plies of woven bidirectional carbon fiber weave on each side of the foam core for each test specimen. Fig. 2.8 shows a diagram of a delamination case of a foam sandwich panel. All delamination cases included a 1" delamination strip. Six delamination cases were created: a delamination at the 1st node line, 2nd node line, 3rd node line, 2" horizontal grids, 3" horizontal grids, and 5"

Fig. 2.9. Foam sandwich panel solid model of the D1 delamination case.

horizontal grids. The 1st node line referred to the node line that is created during the 1st mode of vibration. The delamination cases were referred to by the node line locations or grid line locations (i.e.: the delamination at the 2nd node line is referred to as Delamination D2 case and the delamination at the 2" horizontal grids is referred to as Delamination L2 case). Node line location determination is discussed in the next chapter. For the higher order bending modes, delaminations were not placed as the 1st node line. The D3

case was the only case to include two node line delaminations and also did not include the delamination at the 1st node line. Table 2-2 shows the locations of the delaminations for the foam sandwich panels.

Table 2-2. Delamination grid locations of delamination cases for foam sandwich panels

Delamination case	D1	D2	D3		L2	L3	L5
Grid location	0.5"	7.5"	5"	9"	2"	3"	5"

2.5 Carbon Fiber–Honeycomb Sandwich Panels

The carbon fiber-honeycomb sandwich panels were laid-up using 2 plies of woven bidirectional carbon fiber weave on each side of the honeycomb core for each test specimen. Fig. 2.10 shows a diagram of a delamination case of a honeycomb sandwich panel. All delamination cases included one delamination strip. Six delamination cases were created: a delamination at the 1st node line, 2nd node line, 3rd node line, 2" horizontal grids, 3" horizontal grids, and 5" horizontal grids. The 1st node line referred to the node line that is created during the 1st mode of vibration. The delamination cases were referred to by the node line locations or grid line locations (i.e.: the delamination at the 2nd node line is referred to as Delamination D2 case and the delamination at the 2" horizontal grids is referred to as Delamination L2 case). Node line location determination is discussed in the next chapter. For the node line delaminations, 1st node line delaminations were not repeated for higher order bending modes. The D3 case was the only case to include two node line delaminations and did not include the delamination at the 1st node line. Table 2-3 shows the locations of the delaminations for the honeycomb sandwich panels.

Table 2-3. Delamination grid locations of delamination cases for honeycomb sandwich test specimens

Delamination case	D1	D2	D3		L2	L3	L5
Grid location	0.5"	7.5"	5"	9"	2"	3"	5"

Fig. 2.10. Carbon fiber honeycomb sandwich solid model of the D3 delamination case.

Chapter 3. Experimental Test & Methodology

This section discusses the experimental test and methodologies conducted in this study. The material properties and the mechanical properties of the selected materials are found through volume and strength testing. The mechanical properties of the carbon fiber and foam were found through tensile and compression tests and compared to the manufacturer's specifications. The density and volume fractions of the carbon fiber were obtained through a burn test. A forced harmonic vertical vibration test was conducted on the control specimens for each specimen type to find the node line locations of the vibration bending modes. A forced harmonic sinusoidal sweep horizontal vibration test was the main test conducted to measure the dynamic response of the test specimens.

3.1 Material and Mechanical Properties

3.1.1 LTM45-EL carbon fiber

The mechanical properties of the LTM45-EL carbon fiber were found through static tensile testing. It was necessary to test for the mechanical properties because of the lowered quality of the donated material. The tensile test of the carbon fiber followed the ASTM D3039 *Standard Test Method for Tensile Properties of Polymer Matrix Composite Materials*.^[20] The carbon fiber test coupons were each manufactured with four plies of LTM45-EL woven bi-directional pre-preg carbon fiber. For symmetric carbon fiber, the test standard required thin strips of carbon fiber with a test area of 1"x10". Emery cloth sections needed to be secured onto the carbon fiber test coupons to ensure a proper grip between the test specimen and the tensile pull. 2"x1.5" emery sections were wrapped around onto both 1" wide ends of each test coupon to increase friction and prevent slipping. With the 1.5" length from emery cloth sections, the test coupons needed to be cut to 1"x13" dimensions to meet the test area requirement. It was essential that the test coupons were cut in a fiber direction. The ASTM D3039 also included sampling requirements of 5 eligible test coupons. Test coupon eligibility was determined by the location of failure, which needed to be at 1" away from the gripped ends

The tensile tests were conducted with the use of the Instron 8801 Servohydraulic Fatigue Testing System, which was available in the Cal Poly Aerospace Structures/Composites laboratory.^[21] The test coupons were

placed between the crosshead wedge grips and lined up with the emery cloth sections. It was essential that only the emery cloth sections of the test coupons were contacting the grips to prevent failure at the grips. Fig. 3.1 shows the test coupons and a tensile test being conducted. The test was conducted with a constant crosshead motion of 0.05 in/min, calculated from the test standard. The tensile test would continue until a failure occurred. The criterion for failure detection was the occurrence of a 40% drop in applied load on the test coupon.

Fig. 3.1. Tensile test of carbon fiber test coupons. Shown is the aftermath of a tensile test.

The Bluehill 2 testing software associated with the Instron machine was used to collect data during the tensile test. The main measurements used to find the mechanical properties were the load and extension of the test coupons. The load, extension, and coupon dimensions were used to calculate the stress and strain for each test coupon. The stress was calculated with the measured load and the cross sectional test area, while the strain was calculated with the measured extension and the test area length. Stress-strain curves were created for five eligible test coupons and are shown in Fig. 3.2. An average maximum tensile load was measured at

Fig. 3.2. Test results of ASTM D3039 tensile test for LTM-45EL carbon fiber.

6,738lbs with a standard deviation of 2.23%. The elastic modulus was calculated as the slope of the stress-strain in the elastic region. The average elastic modulus was calculated as 8.7801 Msi with a standard deviation of 0.90%. Because the carbon fiber was woven bidirectional, the elastic modulus calculated from the tensile test was the elastic modulus, E_1 , of the fiber directions. The elastic modulus E_2 can be assumed to be equal to E_1 because of the bidirectional weave of the carbon fiber. The experimental elastic modulus of 8.7801×10^6 psi was shown to be 56.03% lower than the manufacturer's specified elastic modulus of 19.971 Msi.^[16] The experimental elastic modulus was the value used in material properties for theoretical analysis of the test specimen.

The same tensile test was used in order to calculate the Poisson's ratio of the LTM45-EL carbon fiber. The ASTM E132 *Standard Test Method for Poisson's Ratio at Room Temperature* was followed in the testing procedure.^[22] The test coupon dimensions, tolerances, and test methodology for the ASTM E132 test standard are similar to those specified in the ASTM D3039 test standard. To measure the strain in the transverse direction, a strain gage is applied on the surface on the test coupon in a transverse direction perpendicular to the tensile force line of motion. The Vishay Micro-Measurements EA-12-125AD-120 strain gages were used. The 120 Ω strain gages each had specified gage factors within 0.05% precision. One strain gage was mounted onto each test coupon using the manufacturer's specified procedure. Fig. 3.3 shows a test coupon with an attached strain gage and its connection to a Measurements Group P-3500 strain indicator box. A quarter-bridge Wheatstone bridge circuit was connected between the strain gage leads and the strain indicator box. The strain indicator box was also connected to a National Instruments BNC-2111 Data Acquisition Device, which was directly connected to the Bluehill 2 software.^[23] The tensile test was then conducted in the same fashion as the ASTM D3039 test method; however, the test was finished after a maximum load of 3,000 lb. This was done in order to avoid failure. During the test, the Bluehill 2 software would measure both the longitudinal and transverse extensions. Poisson's ratios of the test specimen were calculated using the Equation 3-1 below. Poisson's ratio is defined as the ratio between the transverse strain-load slope and the longitudinal strain-load slope in the equation.

$$\nu = \frac{\frac{\partial \varepsilon_t}{\partial P}}{\frac{\partial \varepsilon_l}{\partial P}}$$

Equation 3-1. Poisson's ratio

A Poisson's ratio of 0.048 was calculated from the measured data with a standard deviation of 11.92%. This Poisson's ratio was similar to those found in previous research efforts.^{[12][13][14]} The experimental Poisson's ratio was the value used in material properties for theoretical analysis of the test specimen.

Fig. 3.3 Vishay strain gage mounted on a test coupon (left) and the resulting Poisson's ratio test (right).

A fiber volume fraction test was conducted to measure the density of the ACG LTM45-EL carbon fiber and compare the data to the manufacturer's specifications. Test coupons were produced with four plies of the woven bidirectional carbon fiber. The test coupons were cut to have cross sectional areas of 2.5"x2.5". Fig. 3.4 shows the burn oven and a burned test coupon. The test coupons were weighed and placed into a Thermolyne 1300 furnace at approximately 1000°F. After approximately 1 hour, the resin burned out of the carbon fibers and burnt test coupon was weighed. Each test coupon was burnt in the oven individually. Equation 3-2 was used to calculate the fiber

Fig. 3.4. Burn test of LTM-45EL carbon fiber using a Thermolyne 1300 furnace.

volume fraction. The fiber and matrix densities ρ used in the equations were obtained from the manufacturer's specifications. A fiber volume fraction of 57.57% was calculated from the test coupons, with a standard deviation of 3.10%. This value is close to the 56% provided in the manufacturer's specifications and falls within the 55%-70% volume fraction of pre-impregnated carbon fibers. The density of the cure carbon fiber was calculated to be $1.4982 \times 10^{-4} \text{ lb}_f\text{s}^2/\text{in}^4$.

$$V_f = \frac{\frac{W_f}{\rho_f}}{\frac{W_f}{\rho_f} + \frac{1 - W_f}{\rho_m}}$$

Equation 3-2. Fiber volume fraction

3.1.2 FR-6710 polyurethane foam

The mechanical properties of the Last-A-Foam FR-6710 polyurethane foam were previously found through compression testing in the Cal Poly Aerospace Structures/Composites laboratory by Rider.^[15] The test followed the ASTM D1621 *Standard Test Method for Compressive Properties of Rigid Cellular Plastics*. The test method required a minimum of 1" thick material. Because only 0.5" thick sheets of

Fig. 3.5. Compressive testing for Last-A-Foam FR-6710 polyurethane foam conducted by Rider.

foam were available for testing, 1" thick test coupons were created by stacking and taping two sheet sections of the foam. Each test section was tested under compression on the Instron 8801 at a compressive rate of 0.1 in/min. Fig. 3.5 shows the foam test conducted by Rider. Rider found that the experimental mechanical properties from the test were nearly 50% less than the manufacturer's specifications. This was mainly to the difference in thickness of the test coupons. As a result, valid testing of the mechanical properties of the foam was not readily available during the time of research. The manufacturer's mechanical properties showed a compressive elastic modulus, E_c , of 17,000 psi. The experimental elastic modulus was found through an

unconventional method of matching the errors of the experimental results and the numerical results. The vibration response of the foam was found by testing a foam sheet, and the bending mode natural frequencies were compared experimentally and numerically. The method is further discussed in Chapter 5.

3.1.3 HexWeb HRH-10 Nomex honeycomb

Compressive testing of the HexWeb HRH-10 Nomex Honeycomb was considered. However, after thorough research into the test methodology and requirements for test coupons, experimental testing for the mechanical properties of the honeycomb was disregarded for all mechanical properties. This is due to the dimensions of the readily available honeycomb in the Cal Poly Aerospace Structures/Composites laboratory. Given the results of the compression testing of the FR-6710 polyurethane foam, any mechanical properties tested using the 0.5" thick honeycomb sheets would not be valid to use as theoretical mechanical properties. The HexWeb HRH-10 Nomex Honeycomb was given a compressive modulus, E_c , of 20,000 psi from the manufacturer.^[18]

3.2 Determination of the Bending Mode Node Line Locations

The locations of the bending mode node lines in each test specimen type were determined through a preliminary vibration test. A simple method incorporating the use of sugar on a vibrating test specimen was selected. As sugar was placed on the vibrating test specimen, the sugar granules would align along the node lines at each vibration mode.

The vibration machine selected for this preliminary test was the MB electronics magnetic vertical shake table, which was readily available in the Cal Poly Aerospace Structures/Composites Laboratory. Fig. 3.6 shows a set-up of the MB shake table. An analog Hewlett Packard function generator was used to generate an electronic sinusoidal frequency signal. The electronic signal from the function generator was sent to an amplifier and the amplified output signal was sent to the shake table. A frequency meter was used to read the resultant frequency. The equipment was very outdated and did not meet current technology standards. However, the outdated machinery and resulting errors did not affect the node line determination. This is because the node lines can be found solely when the sugar modules are aligned. Any frequency measurements from the archaic frequency meter and function generator could be disregarded. The only

factor of error to consider in the node line determination was the precision of the constant frequency input and resultant motion. Previous research had shown that the experimental set-up gave minimal error in node line determination.^{[13][15]} During the tests, it was also found that the node lines created by the sugar were straight and did not deviate from the lines, indicating pure bending.

Fig. 3.6. MB magnetic shake table (right) and function generator, amplifier, and frequency meter (left).

The control case of each test specimen type was used to find the node line locations. To proceed with the node line determination, the test specimen was clamped onto an aluminum fixture on the shake table and tightened. The plate and foam sandwich test specimens were tightened to screw torques of 6 ft-lb, while the honeycomb sandwich test specimen were tightened to a screw torque of 3 ft-lb. A thin layer of sugar was spread across the entire upward facing cross section of the test specimen. The displacement amplitude input of the function generator was set at 0g and the frequency was ramped up slowly to the estimated vibration mode frequency. Estimates of the bending mode frequencies were calculated using data from previous research efforts. When the input frequency came near the estimated bending mode frequency, the input amplitude was ramped up very slowly. Fig. 3.7 shows the reaction of the sugar granules on the vibrating test specimen. The input amplitude was held

Fig. 3.7. Vertical vibration test to determine node line locations.

constant at an amplitude that provided minimal bounce from the sugar granules. The frequency was then slowly changed until the sugar granules began to line up along the node lines. Additional sugar was added onto the surface on the test specimen as needed. Node line determination was conducted for the 1st vibration mode, 2nd vibration mode, and 3rd vibration mode of the control case of all specimen type. Fig. 3.8 depicts the aligned sugar granules at the vibration mode. The sugar granules are seen to line along the horizontal grid locations. The grid system notation is used to correlate each node line location with a grid location. Table 3-1 denotes the location of the node lines with respect to the grid system. For the 2nd and 3rd vibration modes, the 1st node line was ignored. With the node line locations found, the delamination case test specimens were manufactured.

Fig. 3.8. Node line locations for 1st natural frequency (left), 2nd natural frequency (middle), 3rd natural frequency (right).

Table 3-1. Node line locations with respect to grid system

Vibration mode	Node line location with respect to grid system					
	Carbon fiber plate		Foam sandwich panel		Honeycomb sandwich panel	
1 st natural frequency	0.5"		0.5"		0.5"	
2 nd natural frequency	7.5"		8"		7.5"	
3 rd natural frequency	5"	9"	5"	9"	4.5"	8.5"

3.3 Harmonic Horizontal Free Vibration Test

3.3.1 Vibration test equipment

The Unholtz-Dickie S202-ST electrodynamic shaker system was used for the free harmonic horizontal vibration test.^[25] The Unholtz-Dickie shake table, denoted in Fig. 3.9, was available for use in the Cal Poly Aerospace Structures/Composites laboratory. The shake table is able to perform vibrations in both the horizontal and vertical configurations, with forced vibrations able to reach a maximum shock of 200g. A Unholtz-Dickie Vibration Control Data Acquisition System was the software used to interpret measurements from the shaker system and the test specimen. To mount the test specimen, a fixture was manufactured to secure the test specimen to the magnesium table base during vibration tests. The fixture included two 1" tall aluminum 6061 blocks and used hex screws to clamp the test specimen together, creating a fixed end for the test specimen. The fixture was manufactured to be used interchangeably for test specimens of various thicknesses.

Fig. 3.9. Unholtz-Dickie shake table used for forced harmonic horizontal vibration test.

Two types of accelerometers were selected to measure data during the vibration test. Fig. 3.10 shows the two accelerometers. The PCB Piezotronics 353B04 single access accelerometer was used as a measure the vibration response of the test specimen. The PCB sensor uses a shear quartz sensor encased in titanium housing. The PCB accelerometer measures change in motion with an axial sensitivity of 9.740 mV/g. The measurement surface area of the PCB sensor is a .60 in² hex and the weight is approximately 0.028lb. The PCB

sensor has been proven to be a reliable sensor in previous research effects in the Cal Poly Aerospace Engineering department. Up until the start of this research, the PCB sensor was the most advanced readily-available sensor for vibration measurements in the Cal Poly Aerospace Structures/Composites Laboratory. [27]

Fig. 3.10. The PCB 353B04 accelerometer (left) and the VIP 1011A accelerometer (right).

The second accelerometer selected was the VIP Sensors 1011A piezoelectric accelerometer. The VIP accelerometer is a small and lightweight high-impedance accelerometer and is excellent at measuring shock. The VIP accelerometer uses a ring shear PZT-5 sensor encased in a stainless steel housing. Because the VIP 1011A accelerometer is a high impedance accelerometer, a signal conditioner is required to obtain measurements from the sensor. The VIP Sensors 5004-10 remote charge converter was selected to amplify the signal from the VIP 1011A accelerometer. The VIP charge converter is a low-noise, wideband, front-end signal conditional that transforms high impedance charge signals to low impedance voltage signals. When combined with the charge converter, the amplified signal of the VIP 1011A sensor has an axial sensitivity of 7.704 mV/g. The VIP 1011A sensor has a measurement cross section area of approximately 0.10 in² and is almost weightless at 0.0026lb. The VIP 1011A sensor has not yet been used to measure vibration response at Cal Poly. [28]

The PCB accelerometer was used as the control accelerometer. The two types of accelerometers were used as the response accelerometer. The VIP sensor needed to be validated and compared to the PCB sensor due to the unknown performance and difference in accelerometer characteristics. Table 3-2 shows the differences in accelerometer characteristics between the two. The VIP sensor is approximately 10x lighter in

weight than the PCB sensor. The VIP sensor also has a measurement surface area of approximately 6x less than the measurement surface area of the PCB sensor. However, the VIP sensor has a sensitivity of approximately 2.0 mV/g less than the PCB sensor, making it a more accurate sensor. Vibration response measurements were taken with the two sensor types for the control case test specimens for all specimen types.

Table 3-2. Comparison of VIP Sensors 1011A and PCB Piezotronics 353B04

Accelerometer	Weight [lb]	Measurement Surface [in²]	Axial Sensitivity [mV/g]
VIP Sensors 1011A	0.0026	0.1004	7.704
PCB Piezotronics 353B04	0.028	0.6012	9.704

3.3.2 Vibration test overview

The proper test procedures were taken to proceed with Unholtz-Dickie shake table vibration test. An outline of proper test procedures was used and followed the manufacturer's procedures, as well as the ASTM D5112-98 *Standard Test Method of Vibration (Horizontal Motion) Test of Products*.^{[25][28]} To prepare for the test, the test specimen was clamped between the aluminum test fixture and the hex bolts were tightened with a torque wrench to provide a secure fixed end for the test specimen. The carbon fiber plates and foam sandwich panels were tightened to 6 ft-lb, while the honeycomb sandwich panels were tightened to 3 ft-lb. The honeycomb sandwich panels were subjected to a lower torque to prevent delaminations between the honeycomb core and face sheets and to prevent damage to the honeycomb core cell structure. The test fixture was then secured to the magnesium table case of the shake table under a 16 ft-lb torque in the horizontal shake configuration. The accelerometers were placed into the respective control and response ports and attached to the aluminum fixture base and test specimen. A thin layer of petro wax was placed on the surface of the sensors and pressed onto the surface of the fixture base or the test specimen. Fig. 3.11 shows the control accelerometer placed on the test fixture and the response accelerometer placed on a specific grid location. The control accelerometer was horizontally and vertically centered on the surface of the aluminum test fixture. The response accelerometer was placed at the center of each grid location, which was marked on

the surface of the test specimen. The thin layer of petro wax was used to hold the accelerometers on the corresponding surface during the vibration test. The petro wax on the accelerometer was discarded and replaced after every three test runs to ensure a proper bond between the accelerometer and vibrating surface. After all parts were secured, the vibration test was started.

Fig. 3.11. Vibration test set-up with the PCB accelerometer (left) and VIP accelerometer (right) on 04-02 grid location.

The horizontal vibration test was conducted for all test specimen configurations. The vibration amplifier gain was held at 50% of maximum amplifier gain throughout the test. The vibration test for the carbon fiber plates was run with a logarithmic frequency sweep of 5 Hz – 460 Hz for 1 minute and 34 seconds at a sweep rate of 4.16 octaves/min. The vibration test for the foam sandwich was run with a logarithmic frequency sweep of 5.00Hz – 1600Hz for 2 minutes and the vibration test for the honeycomb sandwich was run with a logarithmic frequency sweep of 5.00Hz – 1850Hz for 2 minutes and 3 seconds. The sweep rate of all the vibration tests were kept constant at 4.16 octaves/min. The input acceleration of each vibration test was set constant at 0.5g peak acceleration. The vibration response of each grid location was measured for each test specimen.

Chapter 4. Experimental Test Results

This section discusses the effect of the sensor mass, sensor location, and delamination location on composite structures under dynamic loading. The composite structures consist of carbon fiber plates, carbon fiber-foam sandwich panels, and carbon fiber-honeycomb sandwich panels. The 1st, 2nd, and 3rd vibration bending modes are the main vibration modes studied. The effect of the sensor mass is first discussed. Measurements from the VIP Sensors 1011A accelerometer are compared against the measurements from the PCB Piezotronics 353B04 accelerometer. The effect of sensor location is studied through the comparison of vibration response across horizontal and vertical locations. Response measurements at the off-center left and right horizontal locations are compared against the measurements at the middle horizontal locations. Lastly, the effect of delamination and delamination location is studied. The vibration responses of delamination cases are compared against the control case. Bending mode natural frequency and response amplitude are the main criteria that determined the effect of due to the delamination.

4.1 Vibration Response Overview

The vibration responses of the composite structures were measured during the horizontal vibration test. The response was taken along the vertical middle horizontal locations for the control case composite structures. The vibration responses were studied to give a basic overview of the vibration characteristics. Fig. 4.1 shows a vibration response plot produced from the vibration response measurements of the control case foam sandwich panel. The measurements for this plot were gathered using the VIP Sensors 1011A accelerometer. The peaks from the vibration response plot were noted as areas of resonance excitation, where the bending mode natural frequencies occurred. The 1st peak corresponds to the 1st bending mode natural frequency, the 2nd peak corresponds to the 2nd bending mode natural frequency, and the 3rd peak corresponds to the 3rd bending mode natural frequency. The study only concerned the vibration bending modes – torsion modes were not analyzed. It was seen that natural frequencies and response amplitudes varied with a change in vertical grid location. The change in natural frequency and amplitude corresponded to the deformation shapes of the vibration bending modes. A theoretical plot of the deformed bending mode

shapes is shown in Fig. 4.2. The y-axis of the plot is measured in relative maximum displacement of the oscillations. At the intersections of zero displacement, the bending mode node lines are formed.

Fig. 4.1. Vibration response plot of control carbon fiber plate for various middle vertical grid points.

Fig. 4.2. 1st, 2nd, and 3rd bending modes of a typical fixed-free vibration specimen (top to bottom).

It was seen that amplitude fluctuations was measured in the vibration response of the carbon fiber plate as the resonance excitation ramped down. As the peak response amplitudes increased, fluctuation of the measurements increased. A visual phenomenon occurred during the experimental test at the 1st mode natural

frequency for the carbon fiber plates. As the frequency increased past the 1st bending mode natural frequency, the free end of the plate immediately stopped vibrating and there was a visual zero displacement at the free end. Shortly after, the free end restarted the oscillation. This effect was thought to be the result of the damping characteristics of the carbon fiber plates and the reason for the inconsistent measurements. The amplitude fluctuations were measured regardless of the type of response accelerometer.

As damping characteristics were added to the composite structure, the amplitude fluctuations after the resonance peaks disappeared. Fig. 4.3 shows a vibration response plot of the control case honeycomb sandwich panel. There were no fluctuations of amplitude measurement after the natural frequency peaks in the honeycomb sandwich panels. This was largely due to the added damping characteristics from the honeycomb, as well as the increased thickness and increased moment of inertia. There was some noise after the 3rd natural frequency; however this was due to small torsion effects in the 3rd vibration bending mode. The foam sandwich panels also produced similar results with minimal noise in the vibration measurements.

Fig. 4.3. Vibration response plot of control honeycomb sandwich panel for various middle vertical grid points.

The difference damping characteristics in the composite structures contributed to various natural frequency measurements. Shown in Table 4-1 are the bending mode natural frequencies for each composite structure. The carbon fiber plates produced the lowest natural frequency ranges. This was due to the undamped properties from the geometry and material properties of the carbon fiber. The honeycomb sandwich panel produced the largest natural frequencies for the 1st and 2nd modes. When comparing the

sandwich panels, the foam sandwich panel weighted 67.0% more than the honeycomb sandwich panel and measured a 1st natural frequency 25.1% less than the honeycomb sandwich. The foam sandwich panel also measured a 2nd natural frequency 2.2% lower, but a 3rd natural frequency 2.5% higher than the honeycomb sandwich panel. The advantages and disadvantages between the two sandwich panels are trade-offs. Though the foam sandwich panel weighs more than the honeycomb sandwich panel, the foam creates a significantly larger bonding surface area between skin and core.

Table 4-1. Natural frequencies for the composite structures

	Natural Frequency [Hz]		
Vibration Mode	Carbon fiber plate	Foam sandwich panel	Honeycomb Sandwich panel
1 st bending mode	18.059	178.301	200.908
2 nd bending mode	115.220	725.04	742.553
3 rd bending mode	324.049	1,487.243	1,449.005

Table 4-2 shows the maximum amplitudes of the bending mode natural frequencies for the composite structures. The honeycomb sandwich panels produced the highest response amplitude at the 1st natural frequency. The flexibility of the honeycomb core gave the honeycomb sandwich panel higher response amplitudes. It was seen that as the order of the bending mode increased, the maximum amplitude of the sandwich panel structures greatly decreased. The carbon fiber plates consistently produced high maximum response amplitudes throughout the bending modes.

Table 4-2. Maximum response amplitudes for the composite structures

	Maximum Response Amplitude [g]		
Vibration Response	Carbon fiber plate	Foam sandwich panel	Honeycomb sandwich panel
1 st bending mode	32.982	58.928	68.4792
2 nd bending mode	56.474	23.735	27.722
3 rd bending mode	48.079	9.185	12.811

A time domain analysis was conducted using the vibration response measurements to determine the damping characteristics of the time response. The vibration characteristics of the composite structures decayed with time due to the various damping characteristics present in the material. The damping ratios were first calculated using the half-power bandwidth method, shown in Fig. 4.4. The half-power bandwidth method obtains frequencies of the half-power points. The half power points are defined where the

Fig. 4.4. Half-power bandwidth method used to determine damping ratios.

response amplitude measures within 70.7% ($1/\sqrt{2}$) of the peak response amplitude. The corresponding frequencies occur at two points, before and after the resonance peak. The peak frequency, f_n and half-power frequencies, f_1 and f_2 , can be utilized to determine the damping ratio, ζ_n shown in Equation 4-1.

$$\zeta_n = \frac{f_2 - f_1}{2f_n}$$

Equation 4-1. Damping ratio through half-power bandwidth method

Once the damping ratio was determined, the time domain response was calculated. The frequency range was converted to a time range, t . The resonance frequencies were also converted to units of rad/s. The time-domain response amplitude u_n was then calculated using Equation 4-2.

$$u_n(t) = e^{-\zeta_n \omega_n t} (\cos \omega_n t + \frac{\zeta_n}{\sqrt{1 - \zeta_n^2}} \sin \omega_n t)$$

Equation 4-2. Time-domain response amplitude

Each composite structure was seen to have very similar trends in vibration response. The effects of the sensor mass, sensor location, and delamination location also had similar trends throughout each composite structure. For each effect, only one composite structure type was plotted to highlight the effect in the discussion. The remaining composite structure types were discussed; however the plots were omitted from the report and can be found in the appendix section.

4.2 Determining Change in Vibration Response

The deviation of natural frequency and response amplitude measurements were the basis for analyzing the effects on vibration response. A difference in measurements was evaluated against the control case composite structures and two criteria were created: natural frequency deviation and amplitude difference. The difference of the frequency measurements could not be solely determined by frequency magnitude due to the sample rate from the logarithmic sinusoidal sweep frequency input. A change of 1 Hz in 1st natural frequency had a larger effect than in the 2nd natural frequency. In addition, a change of 1 Hz for one composite structure had a different effect than in another composite structure. To account for this, frequency sampling was used to determine precision of frequency measurement. Sample point deviation was calculated as the difference in Hz divided by the sample rate at the given frequency. Table 4-3 shows the sample rates and natural frequency for each composite structure. The sample rates referred to those taken from the VIP Sensors 1011A accelerometer. As an example, a deviation of 2 measurement points in the foam sandwich panel resulted in a change in 1.6846 Hz at the 1st natural frequency, 6.8932 Hz at the 2nd natural frequency, or 14.249 Hz at the 3rd natural frequency.

Five control case test specimens were tested for each composite structure. The 1st and 2nd natural frequencies were measured within ± 1 sample point when comparing accelerometer locations. Most of the 3rd natural frequencies were measured within ± 1 sample points; however, all were measured within ± 2 sample

points. These natural frequency deviations led to a change of ± 2 sample points as the basis of determining deviation in natural frequency.

Table 4-3. Average sample rates for the composite structures

Vibration Mode	Average Sample Rate [Hz/point]		
	Carbon fiber plate	Foam sandwich panel	Honeycomb Sandwich panel
1 st mode	0.0420	0.8423	0.9460
2 nd mode	0.2725	3.4466	3.5168
3 rd mode	0.7651	7.1245	6.8627

The difference in response amplitude measurements was found by comparing amplitude measurements between control case specimens for the composite structures. Again, five control test specimens were used for each composite structure and the response amplitudes were measured using the VIP Sensors 1011A accelerometer. The amplitudes were compared using measurements for various test specimens while holding the accelerometer location constant. Between each test specimen, most response amplitude measurements fell within $\pm 10\%$ of the average for all composite structures; however, there were some outliers. Most of the outliers occurred in the 3rd bending modes. The outlier response amplitudes were assumed to be due to small defects in the composite structure. Inclusive of the outliers, all response amplitudes fell within $\pm 15\%$ standard deviation. The outliers were disregarded for precision guidelines; however were kept in mind. A baseline difference of $\pm 10\%$ amplitude was chosen as the basis for evaluating change in response amplitude.

4.3 Effects of Sensor Mass on Vibration Response

The effect of the sensor mass on the vibration response of the composite structures was found. The VIP Sensors 1011A accelerometer and the PCB Piezotronics 353B04 accelerometer were selected as a response accelerometer to present a change in sensor mass. The VIP accelerometer had not been used to measure vibration response in the Cal Poly Aerospace Engineering department. All prior vibration response studies at the Cal Poly Aerospace Structures/Composites laboratory used the PCB accelerometer as the response

measurement device. The VIP accelerometer was approximately 10x lighter in weight than the PCB sensor. This lighter weight was desirable due to the smaller inertial effects on the vibration test specimen from the measurement sensor. At higher vertical locations from the fixed end of the composite structure, the effect from the sensor mass inherently became greater and introduced decreased accuracy in the sensor. The measurement surface area of the VIP sensor was also 6x smaller than the surface area of the PCB sensor. This smaller surface area opened the opportunity for higher discretization of the grid system for future research. The foam sandwich panel was the composites structure selected as the vibration test specimen to determine the effects of sensor mass. The carbon fiber plate and honeycomb sandwich panel were not selected due to the possibly of damage caused by the PCB sensor mass during the vibration test.

4.3.1 1st Vibration Bending Mode

The vibration responses for the composite structures were measured using the PCB sensor and the VIP sensor. The natural frequencies and corresponding response amplitudes were recorded for each of the 10 vertical locations on the middle horizontal location. Fig. 4.5 shows a comparison between the VIP sensor and PCB sensor for the 1st bending mode natural frequencies and amplitudes for the foam sandwich panels. For every vertical location, the VIP sensor measured a higher 1st mode natural frequency and amplitude than the PCB accelerometer. The VIP sensor seemed to be a more accurate and precise vibration response measurement device. At the 1" grid location, the VIP sensor measured a natural frequency 8 sample points (6.4 Hz) higher than the PCB sensor. Through the vertical grid locations, the VIP sensor measured maximum natural frequency deviation of 5 sample points (4.3 Hz), while the PCB sensor measured a maximum deviation of 37 sample points (31.0 Hz). The higher accuracy and precision of the natural frequency measurements was due to the smaller mass of the VIP sensor. The VIP sensor also measured higher response amplitudes than the PCB sensor. Between the 1" grid location and 3" grid location, the amplitude responses of the VIP sensor were close to that of the PCB sensor. However, the VIP sensor measured higher response amplitudes at all other vertical locations. The VIP sensor measured a response amplitude of 56.2% (24.0g) higher than PCG sensor at the 10" grid location. The accuracy of amplitude measurements was due to the smaller mass of the VIP sensor. At further locations of free vibration, the sensor mass has a greater effect. It was seen that the effect of sensor location was greatly coupled with the effect of sensor mass.

Fig. 4.5. 1st mode natural frequencies and response amplitudes of foam sandwich panels for the VIP and PCB accelerometers.

A time domain analysis was conducted to find the 1st bending mode time response of the test specimen for the VIP sensor and the PCB sensor. The time response measured from the VIP sensor was found to reach a steady state time 16.0% later than the PCB sensor. With the VIP sensor, the damping ratio of the composite structure was measured to be 7.5% smaller than with the PCB sensor. This was due to the larger mass and area of the PCB sensor. The time response of the VIP accelerometer also induced a higher time domain

Fig. 4.6. 1st mode time response of foam sandwich panels for the VIP and PCB accelerometers.

response amplitude. The difference in 1st mode natural frequencies and amplitude had a significant effect on the time domain response. The accelerometer mass was coupled with the measured damping ratios, where a larger mass would increase damping effects.

4.3.2 2nd Vibration Bending Mode

The VIP sensor also outperformed the PCB sensor in the 2nd vibration bending mode. Fig. 4.7 depicts a comparison between the VIP sensor and PCB sensor for the 2nd vibration mode natural frequencies and response amplitudes. Results from the 2nd mode were very similar to those found in the 1st mode. However, at the 7" and 8" grid locations, the PCB sensor measured 2nd mode natural frequencies slightly higher than the VIP sensor and response amplitudes slightly lower than the VIP sensor. This was due to the location of the 2nd mode node line. The 2nd mode node line produced an area of zero displacement across the horizontal grids at that location. For the 2nd natural frequency, there was minimal effect from the accelerometer mass on the measurements at the 7" and 8" grid locations. The production of zero displacement at the 2nd mode node line resulted in minimal acceleration and vibration, making VIP sensor and PCB sensor comparable measurement devices at that location. It was also seen that after the 8" grid location, the measure response amplitude begins to increase. This was contradictory to the theoretical displacement shape at the 2nd vibration mode.

Fig. 4.7. 2nd mode natural frequencies and response amplitudes of foam sandwich panels for the VIP and PCB accelerometers.

This discrepancy occurred because the accelerometers measured the response magnitude and disregarded direction.

4.3.3 3rd Vibration Bending Mode

The 3rd vibration mode also produced similar results. Fig. 4.8 shows a comparison between the VIP sensor and PCB sensor for the 3rd bending mode natural frequencies and response amplitudes. Again, the VIP sensor measured significantly higher natural frequencies. At the node line locations, the PCB sensor measured natural frequencies comparable to the VIP sensor. The sensors followed similar correlations at the node line locations for the 3rd bending mode.

Fig. 4.8. 3rd mode natural frequencies and response amplitudes of foam sandwich panels for the VIP and PCB accelerometers.

4.3.4 Conclusion of the effects due to sensor mass

The sensor mass was found to have a very significant effect on the precision and accuracy of natural frequency and response measurements for the vibration bending modes of the foam sandwich panels. The VIP sensor outperformed the PCB sensor in natural frequency accuracy, natural frequency precision, and amplitude accuracy. The marks of the VIP sensor were due to the smaller mass and smaller measurement surface area. The smaller mass and area of the VIP sensor created less of added inertial effect on the dynamic response of the composite structures. The lower axial sensitivity of the VIP sensor did not seem to affect

measurements. It was also seen that the effect of sensor location had a strong correlation with the effect of sensor mass.

4.4 Effects of Sensor Location on Vibration Response

The effect of sensor location on the vibration response of the composite structure was determined by varying the sensor location along the horizontal and vertical dimensions. Only the control case composite structures were used to evaluate the effect of the sensor location. It was previously shown that the light weight of the VIP Sensors 1011A accelerometer helped output precise frequency and amplitude measurements through the vertical locations at the horizontal centers. The light weight of the sensor still had a small effect through the vertical locations of the composite structure, and it was shown in the previous section that the effect of sensor location was coupled with the effect of sensor mass. The off-center left and right horizontal locations on the test specimens were thought to further introduce the effect of the sensor mass and create a slight change in moment of inertia of the composite structure. The vibration response at the left and right horizontal locations was compared against the middle horizontal location to determine the effect due to sensor location.

4.4.1 Carbon fiber plates

The vibration response of the carbon fiber plates was measured at the left, middle, and right horizontal location through the vertical length. Fig. 4.9 shows the vibration response at the middle horizontal grids, Fig. 4.10 shows the vibration response at the left horizontal grids, and Fig. 4.11 shows the vibration response at the right horizontal grids. Small peaks in response amplitude before the 2nd and 3rd bending natural frequencies were measured when the sensor was placed at an off-center horizontal location. The small peaks did not show up when vibration response was taken from the center horizontal locations. The small peaks occurred as the response begin to ramp up to the 2nd and 3rd bending natural frequencies and were thought be the torsion vibration modes. Though the mass of the VIP sensor was almost weightless, the off-center placement may introduce further torsion effects in the torsion vibration modes. Though there were concerns of off-center torsion from the sensor location, there was almost no effect on the measured bending

Fig. 4.9. Vibration response plot of control carbon fiber plates for middle horizontal locations.

Fig. 4.10. Vibration response plot of control carbon fiber plates for left horizontal locations.

mode natural frequencies and amplitudes of the composite structure. The findings at each bending mode are discussed in the next subsections.

Fig. 4.11. Vibration response plot of control carbon fiber plates for right horizontal locations.

4.4.1.1 1st Vibration Bending Mode

The effect of sensor location was studied for the 1st mode vibration response. Vibration response measurements were taken from various horizontal and vertical locations. Fig. 4.12 shows a plot of the 1st bending mode natural frequencies and response amplitudes of the carbon fiber plates for vertical and

Fig. 4.12. 1st bending mode natural frequencies and response amplitudes of the carbon fiber plates for vertical and horizontal grid locations.

horizontal locations. There was very little deviation in the measured 1st mode natural frequency of the carbon fiber plates between the horizontal accelerometer locations – the natural frequencies were measured within ± 1 sample point. The off-center horizontal location did not significantly affect the 1st mode natural frequencies. The effect of the vertical sensor locations was similar to the trend shown in the previous section, as there was a drop in natural frequency measurements. The decrease in 1st mode natural frequencies varied a maximum of 5 sample points through the vertical locations.

The measured amplitudes through the vertical locations corresponded to a half-oscillation of the bending mode shape. A comparison of the amplitude measurements between the horizontal locations was conducted and it was found that, per vertical location, most amplitudes measured within 10% of each other. There were some outlier data, as shown in the 3" and 6" vertical grid location; however there were extreme cases that occurred in the carbon fiber plates.

A relationship between the natural frequency and amplitude trends was found. At the maximum amplitude measurements, minimum natural frequencies were measured. This was due to the deformed shape of the bending vibration mode and inherently affected the geometry of the composite structure. With the geometry disrupted, the natural frequency changed. At natural frequency was measured at a maximum at the node line, due to the undisrupted shape. The trend between natural frequency and amplitude through vertical locations was more clearly seen in the 2nd and 3rd bending modes.

Fig. 4.13. 1st mode time response of carbon fiber plates for horizontal grid locations at 10" vertical grid locations.

A time domain analysis was conducted to determine the effects of horizontal and vertical sensor locations on the 1st bending mode. Fig. 4.13 shows the 1st mode time response of the carbon fiber plates for the horizontal locations at the 10" vertical grid locations. The 1st mode time response closely related to the precision of 1st mode natural frequencies between horizontal locations. There was minimal deviation and phase shift of the time responses between horizontal locations.

The 1st mode time response through vertical locations closely related to the natural frequency measurements through vertical locations. Fig. 4.14 shows the 1st mode time response of the carbon fiber plates for vertical locations. As the sensor location increased in vertical grid location, both response amplitude and phase shift increased in the time response of the control carbon fiber plate. Between the 1" to 5" grid locations, there was no phase shift, but there was an increase in response amplitude. Between the 5" and 10" grid locations, the response amplitude stayed constant; however the phase shift increased as vertical location increased. At the 10" vertical grid location, there was a maximum 8.8% phase shift in time response.

Fig. 4.14. 1st mode time response of carbon fiber plates for vertical grid locations.

4.4.1.2 2nd Vibration Bending Mode

The effect of sensor location was studied for the 2nd vibration bending mode. The trends were very similar to the effects studied in the 1st bending mode. Fig. 4.15 shows the 2nd bending mode natural frequencies and response amplitudes of the control carbon fiber plate for vertical and horizontal locations. It

can be seen that the 2nd mode natural frequencies were measured within ± 1 sample points for the carbon fiber plates. It was seen more clearly in the 2nd bending mode that the natural frequency measurements through vertical locations followed the trend of the mode shape. At the node locations, the natural frequencies were at the maximum, and at the maximum displacements, the natural frequencies were at the minimum.

Fig. 4.15. 2nd bending mode natural frequencies and response amplitudes of the carbon fiber plates for vertical and horizontal grid locations.

The 2nd mode amplitudes of the carbon fiber plates were within $\pm 10\%$ through the horizontal locations. The change in amplitude along the vertical grid locations directly followed the mode shape. At the 9" and 10" vertical grid locations, the response amplitudes deviated from the mode shape in the opposite direction. This was because single access accelerometers strictly measure in one direction.

A time domain analysis was also conducted for the 2nd bending mode to find the effect of sensor location on time response. Fig. 4.16 shows the 2nd mode time response of the carbon fiber plates for horizontal locations at the 10" vertical grid location. The time response between horizontal locations reflected the measurement precisions between horizontal grid locations.

The effect of the vertical locations on the time response of the carbon fiber plates is shown in Fig. 4.17. Similar to the 1st mode time response, phase shifts were seen throughout the vertical locations. The phase

shifts in time response corresponded to the change in natural frequency measurement through the vertical locations due to the bending mode shape.

Fig. 4.16. 2nd mode time response of carbon fiber plates for horizontal locations at 10" vertical grid locations.

Fig. 4.17. 2nd mode time response of carbon fiber plates for vertical grid locations.

4.4.1.3 3rd Vibration Bending Mode

The effect of sensor location was examined for the 3rd vibration bending mode. Fig. 4.18 shows the 3rd mode natural frequencies and response amplitudes of the carbon fiber plates through various sensor

locations. The effect of horizontal and vertical sensor location on the 3rd natural frequency measurements was very similar to those shown in the 1st and 2nd mode. The natural frequency deviations in the 3rd mode seemed high in magnitude; however, this was due to the higher frequency sample rate at the 3rd mode. All 3rd mode natural frequencies were measured within ± 1 sample point between the horizontal locations for the carbon fiber plates. The precision of natural frequency measurements between horizontal locations dropped through higher order bending modes. The findings are more significant in the foam and honeycomb sandwich panels. The change in natural frequency along vertical locations again followed the trend of the mode shape.

Fig. 4.18. 3rd bending mode natural frequencies and response amplitudes of the carbon fiber plates for vertical and horizontal locations.

All of the amplitudes measured in the 3rd mode were within the baseline $\pm 10\%$ for the carbon fiber plates. This was contrary to the large deviations found in the foam and honeycomb sandwich panels. The deviations in the 3rd bending mode of the sandwich panels are discussed in the next subsections.

A time domain analysis was conducted to find the effect of horizontal and vertical sensor location on the 3rd bending mode time response. Shown in Fig. 4.19 is the 3rd mode time response of the carbon fiber plates for horizontal locations at the 2" vertical grid. The trend of the 3rd mode time response was very similar to the trend of the 1st and 2nd mode time responses. There was slightly more deviation in the 3rd mode time response between horizontal locations. This was due to the increase in deviations presented in the higher order bending modes.

Fig. 4.19. 3rd mode time response of carbon fiber plate for horizontal locations at 10" vertical grid locations.

The effect of the vertical locations on the time response of the carbon fiber plates is shown in Fig. 4.20. In the 3rd bending mode, the phase shifts became a greater effect on the time response. The phase shifts in time response corresponded to the change in natural frequency measurement through the vertical locations due to the bending mode shape. The responses through the vertical locations were show to be very compressed in

Fig. 4.20. 3rd mode time response of carbon fiber plate for vertical grid locations.

the plot due to the larger effects of the phase shifts. This was one of the reasons less emphasis was placed on the study of the 3rd bending mode time response in this research.

4.4.2 Foam sandwich panels

The vibration response of the foam sandwich panels was measured at the left, middle, and right horizontal location through the vertical length. Fig. 4.21 shows the vibration response at the middle horizontal grids, Fig. 4.22 shows the vibration response at the left horizontal grids, and Fig. 4.23 shows the vibration response at the right horizontal grids. Small peaks in response amplitude before the 2nd and 3rd bending natural frequencies were measured when the sensor was placed at an off-center horizontal location. The small peaks did not show up when the vibration response was taken from the center horizontal locations. The small peaks occurred as the response begin to ramp up to the 2nd and 3rd bending natural frequencies and were thought be the torsion vibration modes. Though the mass of the VIP sensor was almost weightless when compared to the foam sandwich panel, the off-center placement may introduce further torsion effects in the torsion vibration modes. Though there were concerns of off-center torsion from the sensor location, there was almost no effect on the measured bending mode natural frequencies and amplitudes of the composite structure. The findings at each bending mode are discussed in the next subsections.

Fig. 4.21. Vibration response plot of control foam sandwich panels for middle horizontal locations.

Fig. 4.22. Vibration response plot of control foam sandwich panels for left horizontal locations.

Fig. 4.23. Vibration response plot of control foam sandwich panels for right horizontal locations.

4.4.2.1 1st Vibration Bending Mode

The effect of sensor location was studied for the 1st mode vibration response of the foam sandwich panels. Vibration response measurements were taken at various horizontal and vertical locations. Fig. 4.24 shows a plot of the 1st bending mode natural frequencies and response amplitudes for vertical and horizontal locations. There was very little deviation in the measured 1st mode natural frequency of the foam sandwich

panels between the horizontal accelerometer locations – the natural frequencies were measured within ± 1 sample point. The off-center horizontal location did not significantly affect the 1st mode natural frequencies. The effect of the vertical sensor locations showed to correlate with deformed shape of the bending mode. At the maximum deflection, a minimum natural frequency was measured. Near the node line, maximum natural frequencies were measured.

Fig. 4.24. 1st bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal grid locations.

The measured amplitudes through the vertical locations corresponded to a half-oscillation of the bending mode shape. Between the horizontal locations, the amplitudes were measured well within $\pm 10\%$ for the 1st bending mode of the foam sandwich panels. When compared to the carbon fiber plates, the off-center horizontal sensor placement had a much smaller effect on the measured amplitude.

A time domain analysis was conducted to determine the effects of horizontal and vertical sensor locations on the 1st bending mode of the foam sandwich panels. Fig. 4.25 shows the 1st mode time response of the foam sandwich panels for the horizontal locations at the 10" vertical grids. The minimal deviation and phase shift of the time responses between horizontal locations was highlighted by the precision of natural frequency measurements. The left, right, and middle horizontal locations created nearly identical time responses for the 1st mode of the foam sandwich panels.

Fig. 4.25. 1st mode time response of foam sandwich panels for horizontal grid locations at 10" vertical grid locations.

The 1st mode time response through vertical locations closely related to the natural frequency measurements through vertical locations. Fig. 4.26 shows the 1st mode time response of the foam sandwich panels through vertical locations. As the vertical sensor location increased, both response amplitude and phase shift increased in the time response of the foam sandwich panel. At sensor locations close to the fixed end, the foam sandwich panel dampened more quickly.

Fig. 4.26. 1st mode time response of foam sandwich panels for vertical grid locations.

4.4.2.2 2nd Vibration Bending Mode

The effect of sensor location was studied for the 2nd vibration bending mode. The trends were very similar to the effects studied in the 1st bending mode. Fig. 4.27 shows the 2nd bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal locations. It can be seen that all natural frequencies were measured within ± 1 sample points for the foam sandwich panels. The natural frequency deviations seem high due to the higher sample rate and range of the y-axis of the plot, but all the 2nd natural frequency measurements were within ± 1 sample points between horizontal locations. It was seen more clearly in the 2nd bending mode that the natural frequency measurements through vertical locations followed the trend of the mode shape. At the node locations, the natural frequencies were at the maximum, and at the maximum displacements, the natural frequencies were at the minimum.

Fig. 4.27. 2nd bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal grid locations.

The 2nd mode amplitudes of the foam sandwich panels were within $\pm 10\%$ through the horizontal locations. The change in amplitude along the vertical grid locations directly followed the mode shape. At the 9" and 10" vertical grid locations, the response amplitudes deviated from the mode shape in the opposite direction. This was because the accelerometer strictly measured magnitude, regardless of direction.

A time domain analysis was also conducted for the 2nd bending mode to find the effect of sensor location on time response. Fig. 4.28 shows the 2nd mode time response of the foam sandwich panels for horizontal

Fig. 4.28. 2nd mode time response of foam sandwich panels for horizontal locations at 10" vertical grid locations.

locations at the 10" vertical grid location. The time response between horizontal locations reflected the measurement precisions between horizontal locations.

The effect of the vertical locations on the time response of the foam sandwich panels is shown in Fig. 4.29. Similar to the 1st mode time response, phase shifts were seen throughout the vertical locations. The phase shifts in time response corresponded to the change in natural frequency measurement through the vertical locations due to the bending mode shape. The time response of the 2nd bending mode dampened out

Fig. 4.29. 2nd mode time response of foam sandwich panels for vertical grid locations.

more quickly than the time response of the 1st bending mode for the foam sandwich panels. Because of this, the phase shifts in time response had a larger effect, making it harder to distinguish between the time responses at difference vertical locations.

4.4.2.3 3rd Vibration Bending Mode

The effect of sensor location was examined for the 3rd vibration bending mode. Figure 4.30 shows the 3rd mode natural frequencies and response amplitudes of the foam sandwich panels through various sensor locations. Most of the 3rd mode natural frequencies were measured within ± 1 sample points between the horizontal locations; however there were some discrepancies. At the 2" and 3" grid locations, the 3rd mode natural frequencies were measured within ± 2 sample points between horizontal locations for the foam sandwich panels. The precision of natural frequency measurements between horizontal locations dropped in the 3rd bending mode for the foam sandwich panels. A similar finding was seen in the honeycomb sandwich panels. The higher sample rate and y-axis range accentuated the deviation in natural frequency. Again, the change in natural frequency along vertical locations followed the trend of the mode shape, with maximum natural frequencies measured at the node lines.

Figure 4.30. 3rd bending mode natural frequencies and response amplitudes of the foam sandwich panels for vertical and horizontal locations

The effect of sensor location was much more significant on response amplitude at the 3rd bending mode. The maximum response amplitude deviations occurred at the 2", 3", and 10" grid locations, with deviations at

a maximum of $\pm 17\%$. The 3rd mode response amplitudes of the foam sandwich panels presented the highest deviations among the composite structure population. Most of the outlier response amplitudes occurred during measurements in the 3rd bending mode. The higher deviation in response amplitude was a reason less emphasis was placed on the analysis of the 3rd vibration bending mode.

A time domain analysis was conducted to find the effect of horizontal and vertical sensor location on the 3rd bending mode time response. Shown in Fig. 4.31 is the 3rd mode time response of the foam sandwich panels for horizontal locations at the 2" vertical grid. The trend of the 3rd mode time response between horizontal locations was very similar to the trend of the 1st and 2nd mode time responses. There was slightly more change in time response amplitude in the 3rd mode between horizontal locations. This was due to the increase in deviations presented in the higher order bending modes.

Fig. 4.31. 3rd mode time response of foam sandwich panels for horizontal locations at 10" vertical grid locations.

Shown in Fig. 4.32 is the 3rd mode time response of the foam sandwich panels for vertical locations. It was seen that in the time response damped more quickly in the 3rd vibration mode than in the 1st and 2nd. With more damping at the 3rd mode, the phase shifts of the time response had a larger effect and as a result, created less distinguishable time responses between vertical locations. This case was similar to the 2nd bending mode. Because of this, less emphasis was placed on the time response of the 2nd and 3rd bending mode in the foam sandwich panels. The same argument was made for the honeycomb sandwich panels and is discussed in the next sections.

Fig. 4.32. 3rd mode time response of foam sandwich panels for vertical grid locations.

4.4.3 Honeycomb sandwich panels

The vibration response of the honeycomb sandwich panels was measured at the left, middle, and right horizontal location through the vertical length. Fig. 4.33 shows the vibration response at the middle horizontal grids, Fig. 4.34 shows the vibration response at the left horizontal grids, and Fig. 4.35 shows

Fig. 4.33. Vibration response plot of control honeycomb sandwich panels for middle horizontal locations.

Fig. 4.34. Vibration response plot of control honeycomb sandwich panels for left horizontal locations.

Fig. 4.35. Vibration response plot of control honeycomb sandwich panels for right horizontal locations.

the vibration response at the right horizontal grids. Small peaks in response amplitude before the 2nd and 3rd bending natural frequencies were measured when the sensor was placed at an off-center horizontal location. The small peaks did not show up when the vibration response was taken from the center horizontal locations. The small peaks occurred as the response begin to ramp up to the 2nd and 3rd bending natural frequencies and were thought be the torsion vibration modes. Though the mass of the VIP sensor was almost weightless when compared to the foam sandwich panels, the off-center placement may introduce further torsion effects in the

torsion vibration modes. In the 3rd vibration mode, the torsion and bending modes seem to intersect; however, when the measurements were closely examined, the 3rd bending mode was clearly distinguishable. There was almost no effect on the measured bending mode natural frequencies and amplitudes from the torsion mode. The findings at each bending mode are discussed in the next subsections.

4.4.3.1 1st Vibration Bending Mode

The effect of sensor location was studied for the 1st mode vibration response of the honeycomb sandwich panel. Vibration response measurements were taken at various horizontal and vertical locations. Fig. 4.36 shows a plot of the 1st bending mode natural frequencies and response amplitudes for vertical and horizontal locations. There was very little deviation in the measured 1st mode natural frequency of the honeycomb sandwich panels between the horizontal accelerometer locations – the natural frequencies were measured within ± 1 sample point. The 1st natural frequency measurements between horizontal locations for the honeycomb sandwich panel were the most precise of all the composite structures. The off-center horizontal location did not affect the 1st mode natural frequencies. The effect of the vertical sensor locations showed to correlate with deformed shape of the bending mode. At the maximum deflection, a minimum natural frequency was measured. Near the node line, maximum natural frequencies were measured.

Fig. 4.36. 1st bending mode natural frequencies and response amplitudes of the honeycomb sandwich panels for vertical and horizontal grid locations.

The measured amplitudes through the vertical locations corresponded to a half-oscillation of the bending mode shape. Between the horizontal locations, the amplitudes were measured well within $\pm 10\%$ for the 1st bending mode of the honeycomb sandwich panels. When compared to the carbon fiber plates, the off-center horizontal sensor placement in the honeycomb sandwich panel introduced a much smaller effect on the measured amplitude. The honeycomb sandwich panels also measured the smallest amplitude differences between horizontal locations when compared to the foam sandwich panels and carbon fiber plates.

A time domain analysis was conducted to determine the effects of horizontal and vertical sensor locations on the 1st bending mode of the foam sandwich panels. Fig. 4.37 shows the 1st mode time response of the honeycomb sandwich panels for the horizontal locations at the 10" vertical grids. The minimal deviation and phase shift of the time responses between horizontal locations were reflected by the precision of natural frequency measurements. The left, right, and middle horizontal locations created nearly identical time responses for the 1st mode of the honeycomb sandwich panels.

Fig. 4.37. 1st mode time response of honeycomb sandwich panels for horizontal locations at 10" vertical grid locations.

The 1st mode time response through vertical locations closely related to the natural frequency measurements through vertical locations. Fig. 4.38 shows the 1st mode time response of the honeycomb sandwich panels through vertical locations. As the vertical sensor location increased, both response amplitude and phase shift increased in the time response of the honeycomb sandwich panel. At sensor locations close to the fixed end, the honeycomb sandwich panels dampened more quickly.

Fig. 4.38. 1st mode time response of honeycomb sandwich panels for vertical grid locations.

4.4.3.2 2nd Vibration Bending Mode

The effect of sensor location was studied for the 2nd vibration bending mode. The trends were very similar to the effects studied in the 1st bending mode. Fig. 4.39 shows the 2nd bending mode natural frequencies and response amplitudes of the honeycomb sandwich panels for vertical and horizontal locations. It can be seen that all natural frequencies were measured well within ± 1 sample points for the honeycomb

Fig. 4.39. 2nd bending mode natural frequencies and response amplitudes of the honeycomb sandwich panels for vertical and horizontal grid locations.

sandwich panels. As in the 1st mode, the 2nd natural frequency measurements between horizontal locations for the honeycomb sandwich panel were the most precise when compared to the other composites structures. It was seen more clearly in the 2nd bending mode that the natural frequency measurements through vertical locations followed the trend of the mode shape. At the node locations, the natural frequencies were at the maximum, and at the maximum displacements, the natural frequencies were at the minimum.

The 2nd mode amplitudes of the honeycomb sandwich panels were well within $\pm 10\%$ through the horizontal locations. The change in amplitude along the vertical grid locations directly followed the mode shape. At the 8", 9", and 10" vertical grid locations, the response amplitudes deviated from the mode shape in the opposite direction. This was because the accelerometer strictly measured magnitude, regardless of direction. The 2nd mode amplitudes of the honeycomb sandwich panels measured with the highest precision when compared to the other composite structures.

A time domain analysis was also conducted for the 2nd bending mode to find the effect of sensor location on time response. Fig. 4.40 shows the 2nd mode time response of the honeycomb sandwich panels for horizontal locations at the 10" vertical grid location. The time response between horizontal locations reflected the measurement precisions between horizontal locations.

Fig. 4.40. 2nd mode time response of honeycomb sandwich panels for horizontal locations at 10" vertical grid locations.

The effect of the vertical locations on the time response of the honeycomb sandwich panels is shown in Fig. 4.41. Similar to the 1st mode time response, phase shifts were seen throughout the vertical locations. The phase shifts in time response corresponded to the change in natural frequency measurement through the vertical locations due to the bending mode shape. The time response of the 2nd bending mode dampened out more quickly than the time response of the 1st bending mode for the honeycomb sandwich panels. Because of this, the phase shifts in time response had a larger effect, making it harder to distinguish between the time responses at difference vertical locations.

Fig. 4.41. 2nd mode time response of honeycomb sandwich panels for vertical grid locations.

4.4.3.3 3rd Vibration Bending Mode

The effect of sensor location was examined for the 3rd vibration bending mode. Fig. 4.42 shows the 3rd mode natural frequencies and response amplitudes of the honeycomb sandwich panels through various sensor locations. The 3rd mode natural frequencies of the honeycomb sandwich panels were not measured within ± 1 sample points. The largest natural frequency measurements occurred in the 2", 3", 4", and 8" grid locations. This large deviation in natural frequency was contrary to the small deviation found in the 1st and 2nd bending mode of the honeycomb sandwich panels. Similar to the foam sandwich panels, the natural frequency deviation increased in the 3rd bending mode for the honeycomb sandwich panels.

Fig. 4.42. 3rd bending mode natural frequencies and response amplitudes of the honeycomb sandwich panel for vertical and horizontal locations.

The effect of sensor location was much more significant on response amplitude at the 3rd bending mode. The 3rd mode response amplitudes of the honeycomb sandwich panels presented some of the highest deviations among the composite structures. Most of the outlier response amplitudes occurred during measurements in the 3rd bending mode. The higher deviation in response amplitude was a reason less emphasis was placed on the analysis of the 3rd vibration bending mode.

A time domain analysis was conducted to find the effect of horizontal and vertical sensor location on the 3rd bending mode time response. Shown in Fig. 4.43 is the 3rd mode time response of the honeycomb sandwich panels for horizontal locations at the 2" vertical grid. The trend of the 3rd mode time response between horizontal locations was very similar to the trend of the 1st and 2nd mode time responses. There was slightly more change in time response amplitude in the 3rd mode between horizontal locations. This was due to the increase in deviations presented in the 3rd bending mode.

Shown in Fig. 4.44 is the 3rd mode time response of the honeycomb sandwich panels for vertical locations. It was seen that in the time response damped more quickly in the 3rd vibration mode than in the 1st and 2nd. With more damping at the 3rd mode, the phase shifts of the time response had a larger effect and as a result, created less distinguishable time responses between vertical locations. This case was similar to the 2nd

bending mode. Because of this, less emphasis was placed on the time response of the 2nd and 3rd bending mode in the honeycomb sandwich panels.

Fig. 4.43. 3rd mode time response of honeycomb sandwich panel for horizontal locations at 10" vertical grid locations,

Fig. 4.44. 3rd mode time response of honeycomb sandwich panel for vertical grid locations.

4.4.4 Conclusion of effects due to sensor location

The effects of sensor location on the dynamic response of the composite structures were found through experimental testing. At off-center horizontal sensor locations, the vibration torsion modes were measured. The measurement of the torsion modes did not affect the measurement of the bending modes. The off-center horizontal sensor locations produced minimal deviations in bending mode natural frequency and amplitude measurements for most of the vibration bending modes of the composite structures. At vertical sensor locations, the half-oscillation of the bending mode shape can be seen in the amplitude measurements. The natural frequency measurements through the vertical sensor locations followed the trend of the bending mode shape, with the maximum natural frequencies measured at or near the node lines and the minimum natural frequencies measured at or near the maximum displacement.

For the carbon fiber plates, the natural frequencies were measured within ± 1 sample points between horizontal locations for the 1st, 2nd, and 3rd bending modes. The response amplitudes between horizontal locations of the carbon fiber plate were measured within $\pm 10\%$ for the 1st, 2nd, and 3rd bending mode.

For the foam sandwich panels, the natural frequencies were measured within ± 1 sample points between horizontal locations for the 1st and 2nd bending modes. Most of the 3rd mode natural frequencies were measured within ± 1 sample points between horizontal locations; however there were a couple discrepancies that included measurements within ± 2 sample points. The response amplitudes between horizontal locations of the foam sandwich panels were measured within $\pm 10\%$ for the 1st and 2nd bending modes. For the 3rd bending mode, all amplitude measurements were within $\pm 17\%$ between horizontal locations.

The honeycomb sandwich panels included natural frequencies that were measured within ± 1 sample points between horizontal locations for the 1st and 2nd bending modes. The natural frequency precision in the honeycomb sandwich panels were the highest of all composite structures for those two bending modes. About half of the 3rd mode natural frequencies were measured within ± 1 sample points between horizontal locations. The response amplitudes between horizontal locations of the honeycomb sandwich panels were measured with $\pm 10\%$ for the 1st and 2nd bending modes. The honeycomb sandwich panels measured the lowest amplitude differences among the composite structures for those two bending modes. The 3rd mode amplitudes of the honeycomb sandwich panels were measured within $\pm 17\%$ between horizontal locations.

The deviations in natural frequency and amplitude measurements were not affected by the off-center horizontal sensor locations of the composite structures. This finding was also coupled with the light sensor mass and measurement area. For the 1st and 2nd bending mode, the natural frequency measurements were within ± 1 sample point and the amplitude measurements were within $\pm 10\%$ between horizontal locations for all composite structures. For the 3rd bending mode, larger deviations in natural frequency and amplitude were measured for the sandwich panel structures.

4.5 Effects of Delamination Location on Vibration Response

The effect of delamination location on dynamic response was analyzed for the 1st, 2nd, and 3rd bending modes for the composite structures. The delamination cases included delaminations located at the 1st node line (D1), 2nd node line (D2), 3rd node lines (D3), 2" grid location (L2), 3" grid location (L3), and 5" grid location (L5). Each delamination case was compared to the control case per composite structure type. Deviation from natural frequency measurement and difference in response amplitude were the effects studied for all bending modes. The criteria used for determining delamination effect were natural frequency deviations larger than ± 1 sample points and amplitude differences larger than $\pm 10\%$. A time domain analysis was also conducted to examine the phase shift and response amplitude change of the time response. The 1st mode time response was the only significant time domain analysis - the 2nd and 3rd mode time responses were omitted from the report and can be found in the appendix.

4.5.1 Carbon fiber plates

The effects of delamination on the vibration response of the carbon fiber plates were found by determining the deviation in natural frequency and amplitude from the control case. The 1st, 2nd, and 3rd vibration bending modes were studied. A time domain analysis was conducted to find the change in time response due to delamination for the 1st bending mode of the carbon fiber plates. The 2nd and 3rd mode time responses were omitted from the discussion and can be found in the appendix. Off-center horizontal sensor location was found have an insignificant effect on vibration response of the control carbon fiber plates. The same results for the off-center horizontal sensor locations were found for the delamination cases of the carbon fiber plates. The results are omitted from this report; however can be found in the appendix.

4.5.1.1 1st Vibration Bending Mode

The 1st mode vibration responses of the carbon fiber plates were studied to determine the effects of delamination locations. The 1st mode natural frequencies of the delamination cases were compared to those from the control cases. Fig. 4.45 shows the 1st mode natural frequency deviations and response amplitude differences of carbon fiber plates for the delamination cases. Delamination cases D1 and L2 measured natural frequencies 5-10 sample points lower than the control, well outside of the ± 1 sample point range. This drop in natural frequency was due to the location of the delamination relative to the fixed end and the 1st mode node line. The L2 case included a delamination at the 2" grid location, which is approximately 3.5" from the node line. The changes in natural frequencies were also a result of the change of the 1st mode shape geometry.

Delamination cases L3, L5, and D2 measured natural frequencies both inside and outside of the ± 1 range; however the natural frequency deviations did not provide significant differences from the control. The delamination locations of the L3, L5, and D2 cases were at the farther location from the fixed end and 1st mode node line. The L3 case included a delamination located 1" away from the L2 case and created a very slight decrease in sample point measurement. The effect on 1st mode natural frequency due to delamination became insignificant when located 4.5" away from the node line and fixed end.

The D3 delamination case registered higher natural frequency measurements, approximately 7-12 points higher than the control. The D3 case included a delamination at the 5" grid location and a delamination at the 9" grid location. When compared to the L3 case, the 2nd delamination greatly affected the 1st mode natural frequency measurements. The existence of two delaminations in the D3 case changed the mode shape of the carbon fiber plate to increase the overall 1st natural frequency.

Fig. 4.45. 1st bending mode natural frequency deviations and response amplitude differences of carbon fiber plates for delamination cases.

The 1st mode response amplitudes of the delamination cases were compared to the control case. For all delamination cases, there were significant differences in response amplitude between the 1" to the 5" grid locations. At the 1" grid location, the average amplitude difference was the greatest, with more than half of the delamination cases measuring over $\pm 100\%$. The D1 delamination case measured a decrease in response amplitude, while all other delamination cases measured an increase in response amplitude.

The increases in response amplitude at the 1" vertical location were due to the effect of the delamination at a distance away from the fixed end. The delamination decreased the displacement at the delamination location. As a result, reaction occurred at the 1" grid location as an increase in response amplitude to balance the decrease at the delamination location. The magnitude of the response was increased, but the direction was unknown.

At the 2" grid location, most delamination cases measured a significant decrease in response amplitude when compared to the control. This was due to the increased distance from the reaction at the node line. However, the D3 and L5 cases measured maximum response amplitude increases at the 2" grid location. For the D3 and L5 cases, the 2" grid location was at the center between the node line and nearest delamination. With the two zero displacements at the node line and 5" grid location, a maximum response occurred at the center. Although the 2" grid location was between the node line and delamination from the L3 case, a decrease in response amplitude was measured due to the close vicinity of the delamination.

At the 3" grid location, all delamination cases measured a significant decrease in response amplitude when compared to the control. As the sensor was placed at further vertical grid locations, the response amplitudes of the delamination cases reached a steady state within $\pm 10\%$ of the control case. Starting at the 4" grid location, the effect of the delamination became insignificant for all delamination cases. It was noted that the changes in amplitude were the greatest near the 1st mode node line and fixed end. The changes dampened out as the sensor was placed further from the node line and fixed end.

A time domain analysis was conducted to determine the effects of delamination location on the 1st bending mode. Fig. 4.46 shows the 1st mode time response of the carbon fiber plates for the delamination cases at the 10" vertical grid locations. The plot is separated into two sections: the first section compares the node line delamination cases against the control case; the second section compares the grid location delamination cases against the control case. It was seen that the phase shifts in time response were related to the deviations in 1st mode natural frequency for the delamination cases.

Fig. 4.46. 1st mode time response of delamination cases for the carbon fiber plate at the 10" grid location.

4.5.1.2 2nd Vibration Bending Mode

The 2nd mode vibration responses of the carbon fiber plates were analyzed to determine the effects of delamination locations. The 2nd mode natural frequencies of the delamination cases were compared to the control case and the frequency measurement deviations were evaluated. Fig. 4.47 depicts the 2nd bending

mode natural frequency deviations and response amplitude differences of the carbon fiber plates for the delamination cases. The largest frequency deviations were found in the D1, L2, L3, and D2 delamination cases. The L2 case measured 2nd mode natural frequencies 27-30 sample points lower than the control case. The D1 case measured natural frequency drops 15-18 sample points from the control. The L3 and D2 delamination cases also produced drops in 2nd natural frequency, ranging between 8-13 sample points. The 1" movement of the delamination in the L3 case significantly decreased the drop in natural frequency. The location of the delamination on the 2nd node line in the D2 case introduced a significant 8-10 sample point drop in 2nd natural frequency; however did not produce maximum natural frequency deviations through the delamination cases.

The L5 and D3 delamination cases did not produce very significant 2nd mode natural frequency deviations through the vertical grid locations. All vertical grid locations in the L5 case measured decreased natural frequencies, with most of the measurements were within 3 sample points of the lower limit. The natural frequency deviation was not as significant as the decrease measured in other delamination cases. At the 8" grid location of the L5 case, a significant 7 sample point natural frequency was measured. The D3 delamination case measured no deviation in 2nd mode natural frequency for almost every vertical grid location. The only natural frequency deviation was a 2 sample point increase measured at the 2nd node line.

Fig. 4.47. 2nd bending mode natural frequency deviations and response amplitude differences of carbon fiber plates for delamination cases.

The 2nd mode response amplitudes of the delamination cases were compared against the control. The response amplitudes measured at the 8" grid location presented the most significant differences and findings. The D1, L5, D2, and D3 delaminations measured large response amplitude differences. The D1, D2 and D3 cases measured significant amplitude increases of 313%, 134%, and 67%, respectively. The node line delamination cases introduced very large increases in 2nd mode response amplitude at the location of the 2nd node line (8" grid location). The large increase in response amplitude from the D1 case may be due to the effect of the delamination at the 1st node line during the 2nd bending mode. With the initial delamination placed, the effect of the 2nd node line at the 8" grid location is greatly enhanced. A large increase in response amplitude also occurred when the delamination was placed at the 2nd node line for the D2 case. The delamination placed at the 9" grid location in the D3 case presented a delamination close to the 2nd node line and also introduced an increase in response amplitude at the 8" grid location.

The 3"-7" grid locations presented changes in response amplitude trend. The L2 and L3 cases were the most notable changes in this grid range. Maximum response amplitude decreases were measured at the 5" grid location for the L2 and L3 case in the range of the 3"-7" grid locations. The trend of large amplitude changes at the node lines was the most notable trend. At the 2nd node line, the amplitudes of 2nd bending mode produced the largest changes.

4.5.1.3 3rd Vibration Bending Mode

The 3rd mode vibration responses of the carbon fiber plates were analyzed to determine the effects of delamination locations. The 3rd mode natural frequencies of the delamination cases were compared to the control case and the frequency measurement deviations were evaluated. Shown in Fig. 4.48 are the 3rd bending mode natural frequency deviations and response amplitude differences of the carbon fiber plates for the delamination cases. All delamination cases measured very significant decreases in 3rd mode natural frequencies for all grid locations. The drops in natural frequency were due delamination effects on the mode shape. The D1 and L2 delamination cases measured the highest drops in natural frequency of 27-33 sample points. The locations of the delaminations at the 1st node line (0.5" grid location) and 2" grid location were

Fig. 4.48. 3rd bending mode natural frequency deviations and response amplitude differences of carbon fiber plates for delamination cases.

relatively close to the fixed end. However, the L3 delamination case, which has a delamination at the 3" grid location, produced less severe drops in natural frequency measurement. The L3, L5, D2, and D3 cases measured natural frequency decreases of 10-14 sample points.

The 3rd mode response amplitudes for the delamination cases were compared to the control case to find the effects of the delamination locations. There was a general decrease in response amplitude for the delamination cases, noted in the D1, L2, L5, and D3 delamination cases. The L3 and D2 cases did not measure significant response amplitude differences through modes of the grid locations. The most significant trend is the behavior of the response amplitudes at the 2nd node line (5" grid location) and 3rd mode node line (9" grid location). A sudden change in response amplitude occurs for all delamination cases at the 2nd node line and 3rd node line. The changes do not follow a correlated pattern; however the D2 case measured increases in response amplitude at both node locations. The most noteworthy trend was the amplitude change at the node lines. The node line delamination cases provided some of the largest amplitude changes at the node lines.

4.5.2 Foam sandwich panels

The effects of delamination on the vibration response of the foam sandwich panels were found by determining the deviation in natural frequency and amplitude from the control case. The 1st, 2nd, and 3rd

vibration bending modes were studied. A time domain analysis was conducted to find the change in time response due to delamination for the 1st bending mode of the foam sandwich panels. The 2nd and 3rd mode time responses were omitted from the discussion and can be found in the appendix. Off-center horizontal sensor location was found have an insignificant effect on vibration response of the control foam sandwich panels. The same results for the off-center horizontal sensor locations were found for the delamination cases of the foam sandwich panels. The results are omitted from this report; however can be found in the appendix.

4.5.2.1 1st Vibration Bending Mode

The 1st mode vibration responses of the foam sandwich panels were studied to determine the effect of delamination location. The natural frequency deviations and response amplitude differences were used to determine the effects. Fig. 4.49 shows the 1st mode natural frequency deviations and response amplitude differences of the foam sandwiches for the delamination cases.

Fig. 4.49. 1st bending mode natural frequency deviations and response amplitude differences of foam sandwich panels for delamination cases.

The D1 delamination case produced a decrease in natural frequency measurement. The D1 case measured 1st mode natural frequencies approximately 10-12 points lower than the control. The L2, L3, and L5 delamination cases measured 1st mode natural frequencies higher than the +1 sample point upper limit of the control. The L2 and L5 cases measured natural frequencies 4-6 points higher and the L3 case measured natural frequencies approximately 8 points higher. Delamination cases D2 and D3 did not measure significant

1st mode natural frequency deviations from the control. The measurements of the D2 and D3 cases were close to the lower limit of the frequency measurement precision.

There were no overall significant differences in response amplitudes of the foam sandwiches for the delamination cases – most amplitude differences of the delamination cases were measured within $\pm 10\%$ of the control. This was mainly due to the damping characteristics of the foam core in the foam sandwich panel. The significant changes in amplitude difference occurred at the 1" and 5" grid locations, where small changes in amplitude difference were found. The effects of delamination could not be determined through the response amplitude differences, however, the 1st mode natural frequency deviations showed the effects of delamination location on vibration response.

A time domain analysis was conducted to determine the effects of the delamination location on time response. Fig. 4.50 shows the 1st bending mode time response of the foam sandwiches for the delamination cases. The plot is separated into two sections: the first section compares the node line delamination cases against the control case; the second section compares the grid location delamination cases against the control case. There was a significant phase shift in the time responses of the D1 case. The time response of the D1 case accounted for a phase shift delay of more than 180° . Phase shifts in the time response of the D2 and D3 cases were not significant until the response came closer to a steady state. The L2, L3, and L5 cases each presented a significant time response phase shift when compared to the control case. The L2 and L5 case time

Fig. 4.50. 1st bending mode time response of foam sandwich panels for delamination cases.

responses measured phase shift advances of approximately 180° and reached a steady state sooner than the control. The L3 case time response measured phase shift advances greater than 180° and reached a steady state sooner than the control case and L2 and L5 cases. These phase shift advances were due to the increased 1st mode natural frequencies at the L2, L3, and L5 delamination cases. The same phase shift characteristics applied to the carbon fiber plate and honeycomb sandwich delamination cases. An increase in 1st mode natural frequency led to a phase shift advance and the time response reached a steady state at a faster time. A decrease in 1st mode natural frequency led to a phase shift delay and resulted in a later steady state time.

4.5.2.2 2nd Vibration Bending Mode

The 2nd mode vibration responses of the foam sandwich panels were studied to determine the effect of delamination location. The natural frequency deviations and response amplitude differences were used to determine the effects. Fig. 4.51 shows the 2nd bending mode natural frequency deviations and response amplitude differences of the foam sandwich panels for the delamination cases. The natural frequencies measured through vertical grid locations for the delamination cases did not have significant measurement deviations. Most of the 2nd natural frequencies measured for the delamination cases fell within the ± 1 sample point precision of the control case. The damping characteristics of the foam core were a reason for the relatively consistent 2nd mode natural frequency measurements. The L5 delamination case showed the largest

Fig. 4.51. 2nd bending mode natural frequency deviations and response amplitude differences of foam sandwich panels for delamination cases.

average deviation with a 2 sample point increase. There was a change in natural frequency measurement for all delamination cases between the 7" and 9" grid locations. This was caused by the location of the 2nd node line at the 7.5" grid location.

There were also very little significant differences in response amplitude at the 2nd mode for the delamination cases. Because of the damping characteristics of the foam core, the delaminations presented very little effect in the 2nd bending mode. A notable finding in amplitude difference occurred between the 7" and 9" grid locations. The changes in amplitude were due to the 2nd node line at the 7.5" grid location. The 2nd mode natural frequency deviations and response amplitude differences of delamination cases of the foam sandwich panels offered little evidence of the existence of a delamination.

4.5.2.3 3rd Vibration Bending Mode

The 3rd mode vibration responses of the foam sandwich panels were surveyed to examine the effects of delamination location. Fig. 4.52 shows the 3rd bending mode natural frequency deviations and response amplitude differences of the foam sandwich panels for the delamination cases. The only significant 3rd natural frequency deviation was presented in the L3 case, where a delamination existed between the 1st node line (0.5" grid location) and 2nd node line (5" grid location). The L3 delamination case measured natural frequencies 6-7 sample points higher than the control. All other delamination cases did not produce

Fig. 4.52. 3rd bending mode natural frequency deviations and response amplitude differences of foam sandwich panels for delamination cases.

significant deviations in natural frequency measurement due to the damping characteristics presented by the foam core.

The 3rd mode response amplitudes for the delamination cases were compared to the control case to examine the effects of the delamination locations. There was no significant difference in response amplitude for all delamination cases, as most response amplitudes fell within the $\pm 10\%$ range or close to the limits. The effects of the delamination were considered insignificant for the 3rd bending mode of the foam sandwiches. It was apparent that the 2nd node line (5" grid location) and 3rd node line (9" grid location) dictated changes in amplitude at grid locations near the node lines. The trend was noted and was found to be similar to the trends found in the previous bending modes of the foam sandwich panels and carbon fiber plates.

4.5.3 Honeycomb sandwich panels

The effects of delamination on the vibration response of the honeycomb sandwich panels were found by determining the deviation in natural frequency and amplitude from the control case. The 1st, 2nd, and 3rd vibration bending modes were studied. Off-center horizontal sensor location was found have an insignificant effect on vibration response of the control honeycomb sandwich panels. The same results for the off-center

horizontal sensor locations were found for the delamination cases of the honeycomb sandwich panels. The results are omitted from this report; however can be found in the appendix.

4.5.3.1 1st Vibration Bending Mode

The 1st mode vibration responses of the honeycomb sandwich panels were studied to determine the effects of delamination locations. The natural frequency deviations and response amplitude differences from the control case were used to evaluate the effects. Fig. 4.53 shows the 1st bending mode natural frequency deviations and response amplitude differences of honeycomb sandwiches for delamination cases. The 1st mode natural frequency deviations of the delamination case were relatively insignificant. Though most of the delamination cases measured 1st mode natural frequencies outside of the ± 1 sample point precision, none of the delamination cases measured natural frequencies outside of a ± 4 sample point range. The insignificance in 1st mode natural frequency deviation is due to the cell structure of the honeycomb core. There was

Fig. 4.53. 1st bending mode natural frequency deviations and response amplitude differences of honeycomb sandwich panel for delamination cases.

exponentially less contact area between the honeycomb core and carbon fiber face sheets when compared to the foam sandwich and carbon fiber plates.

The effect of delamination location was more visible in the 1st mode response amplitude differences than in the natural frequency deviations. A decrease in response amplitude from the control was measured for

almost delamination cases. The L2, L3, L5, and D3 cases measured response amplitude decreases of 20-55% through the vertical grid locations.

The D1 and D2 delamination cases mostly measured response amplitudes at the lower limit of the control response amplitude. The D2 case had a delamination at a further distance away from the 1st mode node line, decreasing the effect of the delamination. Small changes in amplitude difference were seen near the node line. However, the trend was less significant than those previously seen in the other composites structures.

4.5.3.2 2nd Vibration Bending Mode

The 2nd mode vibration responses of the honeycomb sandwich panels were studied to determine the effects of delamination locations. The natural frequency deviations and response amplitude differences from the control case were used to evaluate the effects. Shown in Fig. 4.54 are the 2nd bending mode natural frequency deviations and response amplitude differences of the honeycomb sandwich panels for the delamination cases. Most of the delamination cases did not produce noteworthy natural frequency deviations

Fig. 4.54. 2nd bending mode natural frequency deviations and response amplitude differences of honeycomb sandwich panels for delamination cases.

– most of the 2nd natural frequency measurements fell within the ± 1 sample point range of the control or within ± 2 sample points. The D1 and L3 delamination cases produced significant 2nd mode natural frequency deviations. The L3 delamination case measured natural frequencies 3-5 sample points higher than the

control. The most significant deviations were measured in the D1 delamination case, with natural frequencies 14-16 sample points lower than the control.

The effect of the delamination location was more visible in the 2nd mode response amplitude differences of the honeycomb sandwiches. There were some decreases in response amplitude for the delamination cases. The D1, L2, and D3 cases generally measured decreases in response amplitude, while the L3, L5, and D2 cases mostly produced response amplitudes within the $\pm 10\%$ range. The trend of the honeycomb sandwich response amplitudes embarked on a significant finding. It can be seen that between the 6-8 grid locations, the largest response amplitude change between grid locations occurred for all delamination cases. The 2nd mode line at the 7.5" grid location induced the sudden change in response amplitude.

4.5.3.3 3rd Vibration Bending Mode

The 3rd mode vibration responses of the honeycomb sandwich panels were examined to find the effects of delamination location. Fig. 4.55 shows the 3rd bending mode natural frequency deviations and response amplitude differences of the honeycomb sandwich panels for the delamination cases. There were notable

Fig. 4.55. 3rd bending mode natural frequency deviations and response amplitude differences of honeycomb sandwich panels for delamination cases.

deviations in 3rd natural frequency measurements for almost all delamination cases. The D1 delamination case produced a 6-8 sample point drop in natural frequency. It can be noted that for the 2nd and 3rd bending modes, the D1 case produced the largest decrease in natural frequency. The L2, L3, L5, and D2 delamination

cases all registered increases in natural frequency. An interesting finding was presented with the D3 delamination case. The natural frequencies measured in the D3 case did not deviate outside of the ± 1 sample point range. The delaminations of the D3 case were located directly on the 2nd node line (4.5" grid location) and 3rd node line (8.5" grid location). The delaminations at the node lines did not change the measured 3rd natural frequencies in the D3 case.

The 3rd mode vibration responses of the honeycomb sandwich panels were examined to determine the effects of delamination locations. For most delamination cases and grid locations, a decrease in response amplitude was measured. The most notable effects occur at the 2nd and 3rd node lines. Similar to the 3rd mode of the carbon fiber plates, the highest change in response amplitude occurred at the 2nd and 3rd node lines. The maximum response amplitude decreases were measured at the node lines per each delamination case. At the 4" grid location, the D1 case measured the highest decrease in natural frequency. The increase effects at the node lines were due to the delamination at the 1st node line (0.5" grid location).

4.5.4 Conclusion of effects due to delamination location

The effects of delamination location on the dynamic response of the composite structures were found through experimental testing. The natural frequency deviation and amplitude differences from the control were used to evaluate the effects due to delamination.

The carbon fiber plates had the largest effect due to delamination of all the composite structures. This was mainly due to the plate structure. The largest natural frequency deviations due to delamination were found in the carbon fiber plates. It was generally seen that as the delamination was placed farther from the fixed end, the vibration response changes became less significant. The changes in amplitude difference were produced very noteworthy findings. Near the node lines, large changes in amplitude were found for all delamination cases of the carbon fiber plates.

The foam sandwich panels were not as largely affected by the delaminations when compared to the carbon fiber plates. This was due to the damping properties presented by the foam core. The natural frequency deviations were the largest for the 1st bending mode and greatly decreased for the higher order bending modes. The changes in amplitude near the node lines for all bending modes were less significant than the carbon fibers plates; however the changes were still noted.

The honeycomb sandwich panels was affected by the delaminations; however the effects were not as large as those presented in the carbon fiber plates. The delamination effect on natural frequency was very small for the 1st bending mode, but increased for the higher order bending modes. The changes in amplitude near the node lines for all bending modes were the most noteworthy trend. Near the node lines, large changes in amplitude were found for all delamination changes.

Chapter 5. Numerical Analysis Results

This section discusses the vibration characteristics of the test specimens determined through numerical analysis. The finite element method was selected, with Simulia ABAQUS being selected as the finite element program. The vibration bending mode natural frequencies and mode shapes of the composite structures were calculated using the ABAQUS program. The finite element methodology applied to the analysis is first discussed in this section. An experimental and numerical validation of the control test specimens is also discussed, followed by the numerical results of the delamination cases.

5.1 Numerical Analysis Approach & Methodology

A finite element analysis was conducted using Simulia ABAQUS/CAE 6.11.^[29] The student version of the program was used in analysis due to limitations of full access to the finite element program. A main restriction of the student version was a limited maximum model size of 1,000 elements. The program was made available by Cal Poly.

The composite structures were modeled as 3"x10.5" rectangular sections in ABAQUS. The LTM-45EL carbon fiber plates and facesheets were modeled as a composite planar shell section with a specified thickness. The FR-6710 polyurethane foam core was modeled as an extruded homogeneous solid section. The HRH-10 Nomex honeycomb core was modeled as a extruded homogenous shell section. The model of the honeycomb core presented created the greatest restrictions on model accuracy due to the model size limit. To account for the model size limits, each honeycomb cell was sized at 0.3", rather than 0.125". The material

Table 5-1 Material properties used in ABAQUS model

	Material Properties			
Material	ρ [lbf·s ² /in ⁴]	E_1 [psi]	E_2 [psi]	ν_{12}
LTM45-EL carbon fiber	1.498×10^{-4}	8,780,000	8,780,000	0.048
Delaminated carbon fiber	1.498×10^{-4}	7,024,000	7,024,000	0.048
FR-6710 polyurethane foam	1.560×10^{-5}	15,000	--	0.3
HRH-10-Nomex honeycomb	4.490×10^{-6}	16,000	--	0.3

properties inputted into the ABAQUS models are shown in Table 5-1. It was essential that all material properties inputted followed a consistent unit format due to the dimensionless property inputs required by ABAQUS. A problem occurred when attempting to input the elastic moduli of the foam and honeycomb material. The elastic moduli could not be obtained through static testing of the foam and honeycomb. For the honeycomb core ABAQUS model, the cell dimensions were increased, thus intuitively decreasing the effective elastic modulus of the honeycomb. For the ABAQUS model, a foam elastic modulus was selected as 90% of the manufacturer's specifications, and a honeycomb elastic modulus was selected as 80% of the manufacturer's specifications. The selection of these elastic moduli was found by matching the experimental results and numerical models. The method and a justification for the selected elastic moduli are discussed in the next section.

The delaminations in the composite structures were modeled with as 3"x1" partitioned sections in the carbon fiber with decreased strength. This method provided an easier way of modeling damage in the carbon fiber without adding the 0.003" thickness presented by the delamination peel ply. The delaminated carbon fiber section was given a 20% decrease in elastic modulus. The decrease represented a disconnect due to the peel ply delamination and a resulting decrease in structural integrity. A value of 20% was found to be the correct amount, as the numerical and experimental errors for the carbon fiber plates matched that of the control case. A value of 10% or 30% created very large percent differences and did not present a viable solution.

The surface interfaces were modeled with the use of tie constraints. For the carbon fiber plates, two shells, each representing 2 plies of carbon fiber, were constructed and interacted with the use of a tie constraint. The facesheet-core interfaces of the sandwich panels were also modeled using tie constraints. The use of tie constraints allowed for the placement of the delaminations. The delaminations were integrated into the model by creating 3"x1" partitioned sections in the carbon fiber. For the carbon fiber plates, each of the two shell facesheets was given the partitioned section. For the sandwich panels, only one of the shell facesheets was given the delaminated section, corresponding to the side of the delamination on the sandwich panel. The tie constraints were used to create the bonding surfaces of the composite lay-up. The partitioned delamination sections were not given tie constraints between the bond surfaces to create the discontinuity between plies.

The specimen models were meshed to specific element sizes. The element types used were SR4. The number of elements was specified to create 2 elements per every 1" in the carbon fiber shells. For the foam core, the same 2:1 ratio was used for the rectangular section and 2 elements were created about the core thickness. For the honeycomb core shell, an approximate global seed size of 0.5 was selected and applied to the model. With the specified element size on the carbon fiber shells, the grid center points can be represented as the center nodes in the 2 element squares and the outputs can be found for those locations. The carbon fiber plate were modeled using approximately 300 elements. The foam sandwich panels were modeled using approximately 550 elements, and the honeycomb sandwich panels were modeled using approximately 700 elements. The boundary condition implemented in the model was an encastre condition on the fixed end surface of each composite structure.

5.2 Validation of the Control Case Finite Element Models

5.2.1 Carbon fiber plate

The numerical results from the finite element models were compared against the experimental results to validate the control case composite structures. The calculations from the finite element model of the carbon fiber plates were first validated. The 1st, 2nd, and 3rd vibration bending modes were found in the ABAQUS model, shown in Fig. 5.1. Shown in Table 5-2 is a comparison of the bending mode natural frequencies for the carbon fiber plates. For the experimental results, the maximum natural frequencies through the vertical grid points are shown in the table. The maximum natural frequencies were selected to minimize the disturbance of resonance and acceleration. The percent differences between the experimental test results and the ABAQUS model were calculated relative to the ABAQUS model. The sample point deviations were calculated by estimating the sample rate of the frequencies given by the ABAQUS model. The sample point deviation was the difference between experimental and numerical results, divided by the sample rate at the numerical frequency. The percent difference and sample point deviation between the experimental results and ABAQUS model showed very small magnitudes. The ABAQUS model calculated slightly higher natural frequencies due properties of the tie constraint. The tie constraint created a perfect bond between the two carbon fiber shells, resulting in higher natural frequencies. The precision between the experimental and ABAQUS natural frequencies becomes greater through higher order bending modes.

Fig. 5.1. ABAQUS model of 1st, 2nd, and 3rd vibration bending modes of carbon fiber plates (left to right).

Table 5-2 Comparison of natural frequencies for carbon fiber plate test specimens and ABAQUS model

Vibration Mode	Natural Frequency [Hz]		Error: Experimental vs. ABAQUS	
	Experimental	ABAQUS	Percent Difference [%]	Sample pt. deviation
1 st mode	18.3374	18.459	-0.6588	-1.409
2 nd mode	116.9721	116.4100	0.4829	1.033
3 rd mode	329.2195	330.1200	-0.2728	-0.588

5.2.2 Foam sandwich panel

With the experimental results and numerical model of the carbon fiber plates validated, an investigation of the elastic modulus of the Last-A-Foam FR-6710 rigid polyurethane foam was initiated. Due to the restricted thickness dimensions of the available foam, the elastic modulus of the could not be obtained through static testing. The vibration response of the foam core was found and compared to an ABAQUS model. The foam was modeled as a extruded homogenous solid. The same density and Poisson's ratio was

inputted into the model and several elastic moduli were selected to attempt to match the bending mode natural frequencies. A foam elastic modulus of $E_1=15,000$ psi presented the least amount of deviation between experimental results and the numerical model. Table 5-3 below shows the comparison of the natural frequencies. Though this was not a standardized method of finding the elastic modulus of the foam, the method provided a means of calculating the foam elastic modulus. The selected foam elastic modulus was 90% of the manufacturer's specification.

Table 5-3 Comparison of natural frequencies for FR-6710 foam test specimen and ABAQUS model

	Last-A-Foam FR-6710 elastic modulus $E_1 = 15,000$ psi			
	Natural Frequency [Hz]		Error: Experimental vs. ABAQUS	
Vibration Mode	Experimental	ABAQUS	Percent Difference [%]	Sample pt. deviation
1 st mode	22.7429	22.823	-0.3510	-0.734
2 nd mode	139.4081	138.0500	-0.9840	1.993
3 rd mode	398.6497	381.9400	4.1130	8.195

The ABAQUS model of the control case foam sandwich panel was also compared against the experimental results. Fig. 5.2 shows the 1st, 2nd, and 3rd bending modes of the foam sandwich panels constructed in ABAQUS. The bending mode natural frequencies were calculated for the numerical model and compared to the experimental results. Table 5-4 shows the results of the comparison. Again, the experimental natural frequencies were taken as the maximum value to reduce deviation due to mode shape. The percent differences were calculated relative to the numerical results. The sample point deviation was calculated by estimating the sample rate of the mode frequencies given by the ABAQUS model. The sample point deviation was the difference between experimental and numerical results, divided by the sample rate at the numerical frequency. The ABAQUS model presented higher natural frequencies when compared to the experimental results; however the magnitudes of difference were much higher than in the case of the carbon fiber plates. The tie constraints became a much bigger role in terms of error for the foam sandwich panels. The difference in bond strength in the facesheet-core interface was greater because of the molecular structure of the foam.

Fig. 5.2. ABAQUS model of 1st, 2nd, and 3rd vibration bending modes of foam sandwich panels (left to right).

Table 5-4 Comparison of natural frequencies for foam sandwich panels and ABAQUS model

Vibration Mode	Natural Frequency [Hz]		Error: Experimental vs. ABAQUS	
	Experimental	ABAQUS	Percent Difference [%]	Sample pt. deviation
1 st mode	179.8452	202.08	-11.0030	-23.616
2 nd mode	732.3851	768.83	-4.7403	-10.363
3 rd mode	1506.6550	1558.4	-3.3204	-7.263

The foam interface was a higher density material and presented a surface that did not adhesively integrate as well as the carbon fiber plies. Because the tie constraints model a nearly perfect bond, this lead to increased natural frequencies in the numerical model. Although the ABAQUS model produced relatively high differences, the error was consistant through each test specimen case and was simply seen as a tare value.

5.2.3 Honeycomb sandwich panel

The ABAQUS model of the control case honeycomb sandwich panel was compared against the experimental results. Fig. 5.3 shows the 1st, 2nd, and 3rd bending modes of the honeycomb sandwich

Fig. 5.3. ABAQUS model of 1st, 2nd, and 3rd vibration bending modes of honeycomb sandwich panels (left to right).

panels constructed in ABAQUS. The bending mode natural frequencies were calculated for the numerical model and compared to the experimental results. Table 5-5 shows the results of the comparison. A COSMOS/M model was not created due to time consumption required by the model. Again, the experimental natural frequencies were taken as the maximum value to reduce deviation due to mode shape. The percent differences were calculated relative to the numerical results. The sample point deviation was calculated by estimating the sample rate of the mode frequencies given by the ABAQUS model. The sample point deviation was the difference between experimental and numerical results, divided by the sample rate at the numerical frequency. The honeycomb cell size in the numerical model was dimensioned at 0.3", rather than 0.125", due to the model size limits of the student edition of ABAQUS. To account for this increase in cell dimension, the

Table 5-5 Comparison of natural frequencies for honey sandwich panels and ABAQUS model

Vibration Mode	Natural Frequency [Hz]		Error: Experimental vs. ABAQUS	
	Experimental	ABAQUS	Percent Difference [%]	Sample pt. deviation
1 st mode	202.9364	210.56	-3.6206	-7.570
2 nd mode	750.7898	759.65	-1.166	-2.471
3 rd mode	1493.5485	1508.4	-0.9846	-2.103

honeycomb elastic modulus was selected to be 80% of the manufacturer's specifications. With these changes, the ABAQUS model showed to be in acceptable agreement with the experimental results. The ABAQUS model presented slightly higher natural frequencies when compared to the experimental results. This was thought to be due to the tie constraints. As in the previous control case numerical models, the natural frequency precision increased through higher order bending modes.

5.3 Finite Element Model of Delamination Cases

Delamination sections were partitioned into the carbon fiber shells in the ABAQUS models to create the delaminations. The delaminations in the numerical models were placed at locations corresponding to the experimental composite structures. The 1st, 2nd, and 3rd bending mode natural frequencies were calculated in the ABAQUS models for all delamination cases of the composite structures. The mode shapes and corresponding displacements were also calculated and compared against the control case.

5.3.1 Carbon fiber plates

The ABAQUS models were constructed for the delamination cases of the carbon fiber plates. The appearances of the deformed shapes were visually very similar to the control model. However when the natural frequencies were observed, deviations from the control model were found. Table 5-6 shows the bending mode natural frequencies obtained from the ABAQUS models of carbon fiber plates. The 1st, 2nd, and 3rd bending mode natural frequencies were. The sample point deviations were also found for each bending mode frequency. The sample rates were estimated using the numerical control natural frequencies. Because of this, the sample point deviations were not rounded to whole integers and the estimated deviations are given. For the 1st and 2nd bending modes, the natural frequencies did not deviate more than -6.0 sample points from the control. In the 1st bending mode, the largest decreases in natural frequency were seen in the D1 and L2 delamination cases. It was observed that after the 2" grid location, as the delamination distance from the fixed end increased, there was less effect from the delamination section. A minimal effect was observed in the D2 case, where the nearest delamination was placed furthest from the fixed end. For the 2nd bending mode, the largest decreases in natural frequency were calculated when the delaminations occurred in the L5 and D3 delamination cases. The 5" grid location was a common variable between the two

Table 5-6 Natural frequencies of delamination cases for carbon fiber plate ABAQUS models

	1st bending mode		2nd bending mode		3rd bending mode	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control
C	18.459	--	116.41	--	330.12	--
D1	17.993	-5.400	117.71	2.389	348.23	11.834
L2	17.987	-5.461	115.26	-2.114	323.65	-4.228
L3	18.125	-3.870	115.27	-2.095	322.89	-4.725
L5	18.328	-1.518	113.69	-5.000	324.67	-3.561
D2	18.447	-0.139	115.16	-2.297	321.2	-5.829
D3	18.326	-1.541	113.33	-5.661	320.24	-6.456

delamination cases. The decrease in natural frequency was due to the delamination located somewhat centrally between the 1st and 2nd node line in the 2nd bending mode. The D1 delamination case was noted as producing the only increase in natural frequency. For the 3rd bending mode, the D1 delamination case produced another increase in natural frequency and was the largest frequency deviation from the control. The other delamination cases for the 3rd bending mode produced drops in natural frequency, with a maximum in the D3 delamination case.

The deformed shapes of the control and delamination cases were compared for the carbon fiber plate ABAQUS models. Fig. 5.4 shows the mode shape amplitude differences between the control and delamination cases for the carbon fiber plate ABAQUS models. The 1st, 2nd, and 3rd bending modes are shown in the figure from top to bottom. The amplitude differences were calculated as percent differences relative to the control case. For the 1st bending mode, the D1 delamination case increased almost 25% in amplitude. There were some small differences in amplitude for the other delamination cases; however not as large as the D1 case. Through the length of the carbon fiber plate, the amplitude differences dampened to a steady state for the 1st bending mode. For the 2nd bending mode, the D1 delamination again measured the highest positive amplitude difference – an increase of almost 20%. The amplitude differences of the delamination cases created sudden

amplitude changes at the 8" grid location, which was the location of the 2nd bending mode node line. The D1 and D2 delamination cases, which included delaminations at the node lines, produced the highest changes in response amplitude at the 8" grid location. The 3rd bending mode also produced the same amplitude changes around the node line locations: the 5" grid and 9" grid locations for the 3rd mode. There were also more deviations through the plate length for all delamination cases in the 3rd mode. For all bending modes, the amplitude differences were relatively small, within 20% of the control. The trend at each bending mode node location presented the main significance in the plots. The delamination cases yielded sudden changes in amplitude difference around the bending mode node lines, with the greatest changes occurring in the node line delamination cases. These trends followed the theory that the node line delaminations would force the highest changes in mode shape.

Fig. 5.4. Amplitude differences of delamination cases for the 1st, 2nd, and 3rd bending modes of the carbon fiber plate ABAQUS models (top to bottom).

5.3.2 Foam sandwich panels

The delamination cases of the foam sandwich panel ABAQUS models were compared against the control case model. Propagating delaminations were seen in several delamination case models for higher order bending modes. Fig. 5.5 shows an example of a propagating delamination in the mode shape of the foam sandwich.

Fig. 5.5. Delamination present in the 3rd bending mode shape of the D3 delamination case of the foam sandwich panel ABAQUS model.

Table 5-7 Natural frequencies of delamination cases for foam sandwich panel ABAQUS models

Case	1 st bending mode		2 nd bending mode		3 rd bending mode	
	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control
C	202.08	--	768.83	--	1558.4	--
D1	197.16	-5.226	736.3	-9.250	1504.2	-7.785
L2	197.69	-4.663	741.72	-7.709	1550.1	-1.192
L3	198.39	-3.919	753.59	-4.333	1542.6	-2.269
L5	199.72	-2.507	765.08	-1.066	1494.0	-9.250
D2	201.64	-0.467	758.32	-2.989	1516.0	-6.090
D3	199.47	-2.772	757.54	-3.210	1456.9	-14.578

The natural frequencies and deformed shape amplitudes were noted in the comparison. Table 5-7 shows the calculated natural frequencies for the foam sandwich panel ABAQUS models. The sample point deviations of the delamination cases were calculated. The sample rates for the frequencies given from the ABAQUS model

were estimated using the control case frequency. The sample point deviations were then calculated using the frequency difference and the estimated sample rate. For the 1st vibration bending mode, the D1 delamination case gave the highest deviation and the largest drop in natural frequency. As the delamination was moved through the vertical grid locations and further from the fixed end, the effect of the delamination was reduced. For the 2nd bending mode, the largest decrease in natural frequency was seen in the D1 delamination case, where the delamination was placed at the 1st node line of the 2nd bending mode. The lowest natural frequency deviation was seen for the L5 case. For the 3rd bending mode, the D3 delamination case provided the highest sample point deviation in natural frequency, with a decrease of almost 15 sample points. The delaminations at the node lines provided higher sample point deviations for all bending modes.

The deformed shapes of the control and delamination cases were compared for the foam sandwich panel ABAQUS models. Fig. 5.6 shows the amplitude differences of the delamination cases. The 1st, 2nd, and 3rd bending modes are shown from top to bottom in the figure. The amplitude differences were calculated as percent differences relative to the control case. For the 1st bending mode, the D1 delamination case produced the highest amplitude difference, a resulting increase of almost 25% at the 1" grid location. The L2 delamination case also produced a significant increase in amplitude, peaking at the 3" grid location. Smaller amplitude differences were seen in the other delamination cases; however were not as significant. Most of the differences occurred before the 5" grid location. Through the vertical grid locations, all delamination case amplitudes dampened to match the control case. For the 2nd bending mode, similar trends were seen for the 1"-3" grid locations. The D1 case again produced the greatest increase in that region. A noteworthy development occurs between the 7" and 9" grid location – sudden changes in the amplitude differences for each delamination case. This sudden change was denoted by the location of the 2nd node line in that region. The D2 case produced the greatest spike in amplitude difference at the 8" grid location. The D3 delamination case yielded a change similar in magnitude to the D2 case. This was due to the D3 case's relatively close vicinity of the delamination. In the 3rd bending mode, the trend is seen again, with highest deviations and changes in amplitude difference for all bending modes. The largest changes at seen in the D1, D2, and D3 delamination cases. For each bending mode, the node line delamination cases generated the highest changes in amplitude at the node lines in the ABAQUS models. These trends followed the theory that the node line

Fig. 5.6. Amplitude differences of delamination cases for the 1st, 2nd, and 3rd bending modes of the foam sandwich panel ABAQUS models (top to bottom).

delaminations would force the highest changes in mode shape. The location delamination cases also presented amplitude differences, but were not as severe as the node line delamination cases.

5.3.3 Honeycomb sandwich panels

The delamination cases of the honeycomb sandwich panel ABAQUS models were compared against the control case model. The natural frequencies and deformed shape amplitudes were noted in the comparison. Table 5-8 shows the calculated natural frequencies for the honeycomb sandwich panel ABAQUS models. The sample point deviations of the delamination cases were calculated. The sample rates for the frequencies given from the ABAQUS model were estimated using the control case frequency. The sample point deviations were then calculated using the frequency difference and the estimated sample rate. For the 1st bending mode, the L2 and D3 delamination cases gave the natural frequency highest deviation, while the D2 case gave the smallest deviation. The small deviation in the D2 case was intuitive due to the far location of the delamination. The effect of the D1 case was not as high as the previous composite structures due to the contact points between the honeycomb cells and the carbon fiber facesheet. The D3 cases produced the highest natural frequency deviations for all bending modes. The high deviation was most likely due to the effect of two delaminations in the honeycomb sandwich panel. The increase in cell structure created a

Table 5-8 Natural frequencies of delamination cases for honeycomb sandwich panel ABAQUS models

	1st bending mode		2nd bending mode		3rd bending mode	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control
C	210.56	--	759.65	--	1508.4	--
D1	205.91	-4.617	730.28	-8.192	1443.7	-9.160
L2	202.35	-8.152	713.35	-12.915	1471.9	-5.167
L3	205.70	-4.826	739.67	-5.573	1496.9	-1.628
L5	207.93	-2.611	755.39	-1.188	1446.2	-8.806
D2	209.47	-1.082	748.37	-3.146	1499.2	-1.302
D3	202.48	-8.023	694.56	-18.156	1340.6	-23.756

decrease in contact points, thus producing higher effects of multiple delaminations. For the 2nd bending mode, the largest deviations were seen in the L2 and D3 delamination cases and the smallest deviation was seen in the L5 case. For the 3rd bending mode, the D1 and D3 cases presented the largest natural frequency deviations, while the D2 case presented the smallest deviations. The trend of natural frequency deviations in the honeycomb sandwich panels was much different than other composite structures. This was due to the cell structure of the honeycomb in the ABAQUS model.

The deformed shapes of the control and delamination cases were compared for the honeycomb sandwich panel ABAQUS models. Fig. 5.7 shows the amplitude differences of the delamination cases. The 1st, 2nd, and 3rd bending modes are shown from top to bottom in the figure. The amplitude differences were calculated as percent differences relative to the control case. For the 1st bending mode, the D1 delamination case produced the highest amplitude difference – an increase of over 50% at the 1” grid location. The L2 case also produced a high increase in amplitude difference, peaking at an increase of over 20% at the 3” grid location. Through the vertical grid locations, all delamination amplitudes dampened to match the control case. It was seen that there was an effect in mode shape of the delamination cases due to the 1st node line. However, as the delamination location increased through the panel, the effect became significantly smaller.

Fig. 5.7. Amplitude differences of delamination cases for the 1st, 2nd, and 3rd bending modes of the honeycomb sandwich panel ABAQUS models (top to bottom).

For the 2nd bending mode, the D3 case gave the most change in amplitude difference through the vertical locations. This change correlated with the high 2nd mode natural frequency deviation on the D3 case. A noteworthy development occurs in all delamination cases between the 6" and 9" grid locations – sudden changes in the amplitude differences. This sudden change was denoted by the location of the 2nd node line in that region. In the 3rd bending mode, erratic amplitude differences are seen for all delamination cases. Again, the D3 case gave the most change, correlating with the high 3rd mode natural frequency deviation. The same trend at the node lines occurs in the 3rd bending mode. The trends at the lines for all bending modes are not as prevalent in the honeycomb sandwich panels as the trends seen in the other composite structures. This was due to the modeling of the honeycomb cell structure in the numerical models.

Chapter 6. Comparison of Experimental Test Results and Numerical Analysis Results

This section discusses the comparison of the experimental test results and numerical analysis results. The 1st, 2nd, and 3rd vibration bending modes were studied for the composite structures. Bending mode natural frequency and resultant mode shape and amplitude were the two main variables analyzed in the comparison. The deviation in natural frequency sampling and amplitude differences were used to determine significant effects due to delaminations and delamination locations.

6.1 Carbon Fiber Plate Comparison

The experimental test results and numerical mode of the carbon fiber plate were compared. The delamination cases were compared against the control case to find natural frequency deviations and response amplitude differences. The 1st, 2nd, and 3rd vibration bending modes were studied.

6.1.1 1st Vibration Bending Mode

In the 1st bending mode, the natural frequencies were measured in the carbon fiber plates and calculated in the corresponding ABAQUS model. The delamination cases were compared against the control case. Table 6-1 shows the comparison between the experimental and numerical results. The D1 and L2 delamination cases gave the largest decreases in natural frequency for the experimental results and numerical model. The delamination trend through the vertical locations were similar the experimental and numerical results. As the delamination was placed further from the fixed end, the effect of the delamination became less significant. However there was a difference in the D3 delamination case. The plate test specimens measured a large increase in natural frequency for the D3 case, while the ABAQUS model calculated a slight decrease in natural frequency. The sample point deviation between the experimental results and numerical model were also calculated. The deviation was calculated relative to the numerical results, with an estimated sample rate at the numerical 1st mode frequency. The deviation magnitudes relative to the control case were higher for the experimental results than the numerical results; however the trend was very similar. For both the

Table 6-1 Comparison of 1st bending mode natural frequencies of experimental results and ABAQUS model for carbon fiber plates

Experimental Results			ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	18.3374	--	18.459	--	-0.6588	1.409
D1	18.0745	-8	17.993	-5.400	0.4530	1.940
L2	17.9877	-10	17.987	-5.527	0.0317	0.136
L3	18.2933	-4	18.125	-3.870	0.9286	4.007
L5	18.4258	3	18.328	-1.518	0.5336	2.329
D2	18.3815	2	18.447	-0.139	-0.3551	-1.560
D3	18.7389	11	18.326	-1.541	2.2531	9.801

experimental and numerical models, the maximum decrease in natural frequency occurred in the L2 delamination case.

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.1. Both models showed

Fig. 6.1. Comparison of 1st bending mode amplitude differences of experimental results and ABAQUS model for carbon fiber plates.

similar trends through vertical locations. The highest amplitude changes occurred at the 1" grid location and dampened out through the vertical locations. The test specimens measured much greater differences in amplitude. Although the magnitudes of amplitude difference varied greatly, the trends through the vertical locations were similar. A significant finding was the change in amplitude difference near the node location for all delamination cases. Through the 1st node line did not fall in the presented vertical locations, the projections of the amplitude differences show the large amplitudes near the node line.

6.1.2 2nd Vibration Bending Mode

In the 2nd bending mode, the natural frequencies were measured in the carbon fiber plates and calculated in the corresponding ABAQUS models. The delamination cases were compared against the control. Table 6-2 shows the comparison between the experimental and numerical results. The control cases between the plate models were validated and fell within 1.03% and 0.483 sample points relative to the numerical model. However, the delamination cases did not produce comparable natural frequency results between the experimental and numerical models. In the plates test specimens, the D1 and L2 cases produced the highest

Table 6-2 Comparison of 2nd bending mode natural frequencies of experimental results and ABAQUS model for carbon fiber plates

	Experimental Results		ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	116.9721	--	116.41	--	1.0330	0.483
D1	112.5540	-18	117.71	2.389	-4.3804	9.477
L2	109.3500	-29	115.26	-2.114	-5.1275	-10.862
L3	113.643	-13	115.27	-2.095	-1.4119	-2.991
L5	114.3081	-7	113.69	-5.000	0.5437	1.136
D2	114.1908	-11	115.16	-2.297	-0.8416	-1.781
D3	116.9721	2	113.33	-5.661	3.2137	6.694

decreases in natural frequency, while the L5 and L3 cases produced the highest decreases in the numerical model. When comparing the delamination cases between models, the D1 and L2 cases calculated the greatest difference between experimental and numerical results. This was due to the partitioned section representing the delamination in the numerical model. Though some of the delamination cases calculated high correlations between the experimental and numerical natural frequency values, some outliers occurred due to the modeling of the delamination in the numerical model. The magnitudes of deviation from control was much higher for the experimental model than the numerical model.

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.2. The change in the 2nd mode shape of the delamination cases followed a similar trend for both the experimental results and numerical model. The largest changes in amplitude difference occurred between the 7" and 9" grid location, where the 2nd node line was located. The D1, D2, and D3 cases produced the largest amplitude differences in that vertical location range. However, the experimental model calculated large increases in amplitude for the D1, D2 and D3 case, while the numerical model calculated an amplitude increase for the D2 and D3 case and an amplitude decrease for the D1 case. Nevertheless, both the experimental and numerical model followed the large disturbance in mode shape at the node line. In the other vertical locations, the mode shapes did

Fig. 6.2. Comparison of 2nd bending mode amplitude differences of experimental results and ABAQUS model for carbon fiber plates.

change; however no clear correlation was made between the numerical and experimental model. The important trend to note was the large amplitude disturbance at the 2nd node line for the delamination cases.

6.1.3 3rd Vibration Bending Mode

In the 3rd bending mode, the natural frequencies were measured in the carbon fiber plates and calculated in the corresponding ABAQUS model. The delamination cases were compared against the control. Table 6-3 discusses the comparison of the experimental and numerical 3rd mode natural frequencies for the delamination cases. A small deviation between 3rd mode natural frequencies was present between the experimental and numerical control models. Most of the delamination cases also showed low deviation between the experimental and numerical models. However, the D1 and L2 delamination cases provided high percent differences and sample point deviations relative to the numerical model. There were also some disconnects when comparing the experimental and numerical model. For the experimental results, the D1 and L2 cases measured the highest sample point deviations for the 3rd mode natural frequency. However for the numerical model, the D1 case measured the highest increase and the D3 case measured the highest

Table 6-3 Comparison of 3rd bending mode natural frequencies of experimental results and ABAQUS model for carbon fiber plates

	Experimental Results		ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	329.2195	--	330.12	--	-0.2728	-0.588
D1	307.7671	-30	348.23	11.834	-11.620	-26.441
L2	304.8187	-33	323.62	-4.228	-5.8184	-12.306
L3	319.0796	-13	322.89	-4.725	-1.1801	-2.490
L5	319.0796	-13	324.67	-3.561	-1.722	-3.653
D2	318.3126	-14	321.20	-5.829	-0.8989	-1.887
D3	320.6190	-12	320.24	-6.456	0.1183	0.248

decrease. The natural frequencies measured for all delamination cases were decreases from the control, with the exception of the D1 numerical case.

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.3. The most notable characteristics that both models had in common were the changes in mode shape around the node line locations. The node line delamination cases D1, D2, and D3 presented the largest changes in mode shape and amplitude difference for both the numerical and experimental results. At the 5" grid location, the D2 and D3 cases had the highest increase in amplitude, while the D1 and L2 cases had the largest amplitude decreases. The D2 and D3 cases also had some of the largest amplitude changes at the 9" grid location. The L3 experimental case measured an unusually large amplitude increase at the 9" grid location. Even so, the large change in amplitude and mode shape at the node lines became a consistent trend through all bending modes and specimen types.

Fig. 6.3. Comparison of 3rd bending mode amplitude differences of experimental results and ABAQUS model for carbon fiber plates.

6.2 Foam Sandwich Panel Comparison

The experimental test results and numerical mode of the foam sandwich panel were compared. The delamination cases were compared against the control case to find natural frequency deviations and response amplitude differences. The 1st, 2nd, and 3rd vibration bending modes were studied.

6.2.1 1st Vibration Bending Mode

The 1st mode natural frequencies were measured in the foam sandwich panels and calculated in the corresponding ABAQUS model. The delamination cases were compared against the control case. Table 6-4 discusses the comparison of the experimental results and numerical results. The sample point deviation between the experiment and numerical results were noted for the high magnitudes; however this was due to the use of tie constraints to model the bond surface in the ABAQUS model. The sample point deviation of low-20's can be used as a tare value. With that, the D1, D2, and D3 delamination cases presented the closest experimental trends to the numerical model. In both models, the D1 case produced the highest decrease in natural frequency. The D2 and D3 cases produced relatively lower natural frequency deviations, due to the damping properties of the foam core in the sandwich panel. The location delamination cases L2, L3, and L5 measured increases in natural frequencies for the test specimens; however were calculated to produce moderate decreases in the numerical model.

Table 6-4. Comparison of 1st bending mode natural frequencies of experimental results and ABAQUS model for foam sandwich panels

	Experimental Results		ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	179.8452	--	202.08	--	-11.0030	-23.616
D1	171.4012	-11	197.16	-5.226	-13.065	-27.359
L2	184.2220	6	197.69	-4.663	-6.8127	-14.3048
L3	186.8990	10	198.39	-3.919	-5.9721	-12.205
L5	183.3382	5	199.72	-2.507	-8.2024	-17.400
D2	178.9824	-2	201.64	-0.467	-11.2367	-24.065
D3	178.1238	-2	199.47	-2.772	-10.702	-22.673

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.4. The test specimen

Fig. 6.4. Comparison of 1st bending mode amplitude differences of experimental results and ABAQUS model for foam sandwich panels.

measurements followed a similar trend when compared to the numerical model. The D1 delamination case measured amplitude increases of approximately 20% at the 1" grid location in both models. In the experimental results, the amplitudes fell within the $\pm 10\%$ precision of the accelerometer for almost all delamination cases and vertical grid locations. The ABAQUS model showed comparable results as the amplitude differences decreased through the vertical locations. The effects of the delamination around the 1st node line were less severe in the foam sandwich panels than the carbon fiber plates. This was due to the damping characteristics employed by the foam core in the sandwich panel.

6.2.2 2nd Vibration Bending Mode

In the 2nd bending mode, the natural frequencies were measured in the foam sandwich panels and calculated in the corresponding ABAQUS model. The delamination cases were compared against the control. Table 6-5 discusses the comparison of the 2nd bending mode natural frequencies. The control case of the numerical and experimental models showed a smaller percent difference and sample point deviation between the models for the 2nd bending mode, as compared to the 1st bending mode. The 2nd mode natural frequencies of the experimental results provided very little deviation in natural frequency when compared to the control case. When compared to the numerical model, the experimental results greatly outperformed the numerical model.

Table 6-5. Comparison of 2nd bending mode natural frequencies of experimental results and ABAQUS model for foam sandwich panels

Experimental Results			ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	732.3851	--	768.83	--	-4.7403	-10.363
D1	728.8716	-1	736.3	-9.250	-1.0089	-2.112
L2	735.9156	2	741.72	-7.709	-0.7826	-1.650
L3	735.9156	2	753.59	-4.333	-2.3454	-5.026
L5	735.9156	2	765.08	-1.066	-3.8119	-8.293
D2	732.3851	1	758.32	-2.989	-3.4200	-7.375
D3	728.8716	0	757.54	-3.210	-3.7844	-8.152

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.5. The amplitude differences were relatively small for the experimental results and numerical model. The same trend occurs

Fig. 6.5. Comparison of 2nd bending mode amplitude differences of experimental results and ABAQUS model for foam sandwich panels.

for both models near the location of the 2nd node line, which falls between the 6" and 9" grid location. The node line delamination cases D1, D2 and D3 produced the largest mode shape differences between that region. The D1 case induced a strikingly similar trend between the 6" and 9" grid locations for both the experimental and numerical model. To coincide with the natural frequency data previously discussed, the experimental results measured smaller mode shape disturbances than the numerical model.

6.2.3 3rd Vibration Bending Mode

In the 3rd bending mode, the natural frequencies were measured in the foam sandwich panels and calculated in the corresponding ABAQUS models. The delamination cases were compared against the control. Table 6-6 discusses the comparison of the experimental results and numerical model. The natural frequency deviation between the experimental and numerical control cases continued to decrease for higher order bending modes. As in the 2nd bending mode, the experimental results outperformed the numerical model for most delamination cases of the foam sandwich panels. The numerical model measured large decreases in natural frequency for the node line delamination cases, with the largest occurring in the D3 case.

Table 6-6 Comparison of 3rd bending mode natural frequencies of experimental results and ABAQUS model for foam sandwich panels

Case	Experimental Results		ABAQUS Results		Comparison	
	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	1487.243	--	1558.4	--	-3.3204	-7.432
D1	1506.655	1	1504.2	-7.785	0.1632	0.353
L2	1506.655	1	1550.1	-1.192	-2.8027	-6.240
L3	1550.761	7	1542.6	-2.269	0.5290	1.172
L5	1506.655	1	1494.0	-9.250	0.8470	1.818
D2	1506.655	2	1516.0	-6.090	-0.6164	-1.342
D3	1492.234	-1	1456.9	-14.578	2.4253	5.075

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.6. The changes in amplitude difference of the experimental results mirrored a new mode shape through the vertical locations for all delamination cases. The trend of change in amplitude difference at the node line locations was against prevalent. There were more erratic changes in amplitude difference in the numerical model than the experimental results. These erratic changes may be due to the modeling of the delaminations in the numerical model. Even so, the trend in amplitude difference remained.

Fig. 6.6. Comparison of 3rd bending mode amplitude differences of experimental results and ABAQUS model for foam sandwich panels.

6.3 Honeycomb Sandwich Panel Comparison

The experimental test results and numerical mode of the honeycomb sandwich panel were compared. The delamination cases were compared against the control case to find natural frequency deviations and response amplitude differences. The 1st, 2nd, and 3rd vibration bending modes were studied. The comparison did not produce significant findings due to the limitations in the numerical model of the honeycomb sandwich panel.

6.3.1 1st Vibration Bending Mode

The 1st mode natural frequencies were measured in the honeycomb sandwich panels and calculated in the corresponding ABAQUS model. The delamination cases were compared against the control case. Table 6-7 discusses the comparison of the experimental results and numerical results. For the experimental results, the D1, L5, and D3 delamination cases measured the highest decrease in natural frequency, while the L2 and D3 cases measured the highest decreases for the numerical model. The sample point deviation from the control was much higher in the numerical model than in the experimental model. The deviation trends through delamination cases show small correlation between the models. Though the frequencies are calculated to be fairly similar, the honeycomb cell structure in the numerical model does not match that of the experimental model, resulting in differences in trend.

Table 6-7 Comparison of 1st bending mode natural frequencies of experimental results and ABAQUS model for honeycomb sandwich panels

	Experimental Results		ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	202.9364	--	210.56	--	-3.6206	-7.570
D1	200.9936	-3	205.91	-4.617	-2.3876	-4.882
L2	201.9626	-1	202.35	-8.152	-0.1915	-0.385
L3	204.8979	2	205.70	-4.826	-0.3899	-0.7965
L5	200.9936	-3	207.93	-2.611	-3.3359	-6.888
D2	207.8979	2	209.47	-1.082	-2.1827	-4.540
D3	200.0292	-3	202.48	-8.023	-1.2104	-2.434

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.7. The D1 delamination case measured the highest amplitude increase at the 1" grid location in both models. In the experimental results, the amplitudes differences stay fairly consistent through vertical locations for most delamination

Fig. 6.7. Comparison of 1st bending mode amplitude differences of experimental results and ABAQUS model for honeycomb sandwich panels.

cases. However, there was a strong negative amplitude difference. The effects of the delamination around the 1st node line were seen for both models, though were not as significant as the other composite structures for the 1st mode.

6.3.2 2nd Vibration Bending Mode

In the 2nd bending mode, the natural frequencies were measured in the honeycomb sandwich panels and calculated in the corresponding ABAQUS models. The delamination cases were compared against the control. Table 6-8 shows the comparison between the experimental and numerical results. For the experimental results, the D1 delamination case measured the largest deviation in natural frequency, while the L2 case measured the largest deviation for the numerical model. The D3 case for the numerical model was disregarded due to the large changes discussed earlier. There was no clear trend between the experimental and numerical models in the natural frequency deviations for the delamination cases. The honeycomb cell structure in the numerical model does not match that of the experimental model, resulting in differences in trend.

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.8. The same trend occurred for both models near the location of the 2nd node line, which falls between the 6" and 9" grid

Table 6-8 Comparison of 2nd bending mode natural frequencies of experimental results and ABAQUS model for honeycomb sandwich panels

Experimental Results			ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	750.7898	--	759.65	--	-1.1664	-2.471
D1	698.5316	-16	730.28	-8.192	-4.3474	-8.856
L2	747.1874	-1	713.35	-12.915	4.7434	9.438
L3	765.3738	5	739.67	-5.573	3.4750	7.170
L5	758.0468	2	755.39	-1.188	0.3517	0.741
D2	754.4095	1	748.37	-3.146	0.8070	1.685
D3	743.6023	-2	694.56	-18.156	7.0609	13.679

Fig. 6.8. Comparison of 2nd bending mode amplitude differences of experimental results and ABAQUS model for honeycomb sandwich panels.

location. Changes in amplitude difference occurred in that range for most delamination cases. Although the trend in natural frequency deviation did not match between experimental and numerical models, the trends were in better agreement for the amplitude differences.

6.3.3 3rd Vibration Bending Mode

In the 3rd bending mode, the natural frequencies were measured in the honeycomb sandwich panels and calculated in the corresponding ABAQUS models. The delamination cases were compared against the control. Table 6-9 discusses the comparison of the experimental results and numerical model. There was little correlation in natural frequency deviation between the experimental and numerical models, again due to the difference in honeycomb structure between the experimental and numerical models. The experimental results mostly measured increases in natural frequency – the reasoning for this was unknown.

Table 6-9 Comparison of 3rd bending mode natural frequencies of experimental results and ABAQUS model for honeycomb sandwich panels

	Experimental Results		ABAQUS Results		Comparison	
Case	Frequency [Hz]	Sample pt. deviation vs. control	Frequency [Hz]	Sample pt. deviation vs. control	Percent Difference [%]	Sample pt. deviation
C	1493.5485	--	1508.4	--	-0.9846	-2.103
D1	1423.414	-6	1443.7	-9.160	-1.4052	-2.872
L2	1507.9857	6	1471.9	-5.167	2.4516	5.109
L3	1486.3822	4	1496.9	-1.628	-0.7026	-1.478
L5	1507.9847	6	1446.2	-8.806	4.2722	8.747
D2	1493.5485	4	1499.2	-1.302	-0.3770	-0.800
D3	1465.0891	-1	1340.6	-23.756	9.2861	17.624

The changes in the mode shape were also determined for the delamination cases. The experimental results and numerical model were compared and the comparison is shown in Fig. 6.9. The same trend occurred for both models near the location of the 2nd and 3rd node lines. Changes in amplitude difference occurred in that range for most delamination cases. Although the trend in natural frequency deviation did not match between experimental and numerical models, the trends were in better agreement for the amplitude differences.

Fig. 6.9. Comparison of 3rd bending mode amplitude differences of experimental results and ABAQUS model for honeycomb sandwich panels.

6.4 Conclusion of Experimental and Numerical Comparison

The experimental test results and numerical models were compared for the composite structures. Bending mode natural frequency and resultant mode shape and amplitude were the two main variables analyzed in the comparison.

The carbon fiber plates showed the best agreement between the experimental and numerical models. The natural frequency deviations followed a similar trend for the two models. The experimental results measured larger magnitudes of natural frequency deviation for the delamination cases; however the trends were similar. The changes in mode shapes were found to be in excellent agreement in trend. Though the magnitudes of amplitude difference were found to be much higher for the experimental test results than the numerical model, the trends near the node lines were noted. For all carbon fiber plate delamination cases, large changes in amplitude were present near the bending mode node lines.

The foam sandwich panels showed to be in good agreement between the experimental and numerical models. The trends of natural frequency deviation for the delamination cases were very similar. For the 2nd and 3rd bending modes, it was found that the experimental test results measured smaller natural frequency deviations than the numerical models. This was due to the assumptions made in the numerical model for the facesheet-core bond interface. The comparison of the mode shape changes showed to be in good agreement.

Though the magnitudes of amplitude difference were found to be much smaller for the experimental test results than the numerical model, the trends near the node lines were noted. For all foam sandwich panel delamination cases, large changes in amplitude were present near the bending mode node lines.

The honeycomb sandwich panels showed to be in moderate agreement between the experimental and numerical models. The numerical model of the honeycomb sandwich panels were restricted to the model size given in the finite element software. As a result, the numerical models created were not ideal for comparison. The natural frequency deviations measured similar deviation magnitudes between the experimental and numerical model; however the trends were not similar. The comparison of the mode shape changes showed to be in good agreement for the 1st and 2nd bending modes. The changes in amplitude near the bending mode node lines were similar for the 2 bending modes for all delamination cases of the honeycomb sandwich panels. However, the mode shape changes of the 3rd bending mode did not match between the experimental and numerical models.

It was found that the majority of experimental and numerical comparison showed to be in good agreement. The trends of natural frequency deviation were moderate between the experimental and numerical models. There was a high correlation between the experimental and numerical models for the changes in mode shape and amplitude changes near the bending mode node lines.

Chapter 7. Conclusion

The objective of this study was to determine the effects of sensor mass, sensor location, and delamination location on the dynamic response of composite structures. The bending mode natural frequencies and response amplitudes were measured for the 1st, 2nd, and 3rd bending modes. The deviation in natural frequency and mode shape amplitude difference were the criteria for evaluating the effects.

The sensor mass was found to produce significant effects on the vibration response of the test specimens. Higher precision and accuracy in resonance measurements were highlighted with smaller sensor weights. The effect of sensor location was coupled with the effect of sensor mass. Off-center horizontal sensor locations produced small excitation spikes in response amplitude before the 2nd and 3rd resonance peaks. When the sensor mass was minimized, the effect of off-center horizontal sensor location became insignificant and did not affect the vibration response of the composite structures. The effect of vertical sensor location correlated with bending mode shape. The bending mode natural frequencies were measured at a maximum at the node line locations and measured at a minimum at the maximum deflection areas. As the sensor mass was minimized, the range between maximum and minimum measurements greatly decreased.

The effect of delamination was found to be greatly dependent on free vibration distance of the delamination and the composite structure type. As the delamination was placed at a further free vibration distance from the fixed end, the natural frequency measurements produced less deviations. As damping characteristics were added to the composite structure, as in the foam sandwich panels and honeycomb sandwich panels, the natural frequency deviations decreased. For all delamination cases, erratic changes in mode shape were measured near the bending mode node lines. This effect on the response amplitude of the composite specimen was found true for each of the composite structures. For the mode shape effect due to delamination, the experimental measurements were found to be in good agreement with the numerical models.

With the effects of sensor mass, sensor location, and delamination location concluded, the results can provide preliminary research in the use of accelerometer measurements as a method of structural health monitoring. The results showed that the detection of delamination through dynamic response may be dependent on the composite structure material. As damping characteristics are increased in the composite

structure, the damage detection ability may be decreased. The placement of the sensor can also be a large factor in detection ability. The ability to detect damage through dynamic response can be increased by placing the sensor at locations of minimum displacement. An application to aircraft design may be to place the sensors at fixed constraints of the aircraft. Minimization of the sensor mass would be ideal, possibly through the use of thin piezoelectric sheets. Placement of sensors at the fuselage skin-bulkhead bonding areas and wing skin-spar bonding areas on composite aircraft may provide an active structural health monitoring system that can help reduce lifecycle cost. However, further research will be needed to better study damage detection techniques using dynamic response. Possible future work and research is discussed in the next chapter.

Chapter 8. Future Work

Many additional studies and investigations will be needed to find more relations between dynamic response measurements and damage detection in the field of structural health monitoring. A larger numerical model for the honeycomb sandwich panels presented in this research would provide for better experimental and numerical agreement. A larger numerical model would require a finite element software with a higher model size limit. With a larger numerical model, the honeycomb sandwich panel can be more accurately represented. The reactions and effects at the honeycomb cells can be calculated at a microscopic level and compared to reaction present in a solid foam core. A different approach to modeling the delamination can be pursued as well. Detailed research can be conducted to focus on the findings of a specific composite structure type. The use of difference vibration inputs, such as random vibration or synthesized shock, can also present opportunities for more research.

The effect of delaminations must be further investigated. The erratic changes in bending mode shape due to the delaminations opened the door for micro-laminate analysis. A study of the force reactions due to the discontinuities may be pursued. Delaminations due to impact would provide for larger damage effects and possibly more significant dynamic response effects. As the overall effects of damage are better understood, smaller and more centralized damage locations can be implemented to test for accuracy. As the damages are made smaller, the significance of the dynamic response will decrease. A minimum damage size can be then correlated.

In addition to the study of the vibration bending modes, the vibration torsion modes can also be studied. This would require an increase in horizontal width of the test specimen for proper horizontal sensor placement. However, the numerical and experimental values of the control test specimens must be first examined to determine any validity in the sensor measurements.

References

- [1] Jones, R. M. *Mechanics of Composite Materials*. 2nd ed. Philadelphia, PA: Taylor & Francis, 1999. Print.
- [2] *Advanced Composites Cargo Aircraft (X-55A)*. Aviation Week: 2011 Program Excellence Award Nomination, 1999. Web.
- [3] *Composites Basics: Materials 4-Other Matrix Constituents*. American Composites Manufacturers Association, 2004. Web.
- [4] Sierakowski, R. L., Chaturvedi, S. K. *Dynamic Loading and Characterization of Fiber-Reinforced Composites*. New York: John Wiley & Sons, 1997. Print.
- [5] Derriso, M. M., Olson, S. E. *The Role of Structural Health Monitoring for Future Air Vehicles*. AIAA Structures, Structural Dynamics, and Materials Conference. May 2006. Web.
- [6] Dukkipati, R. V. *Vibration Analysis*. Connecticut: Alpha Science International, 2004. Print.
- [7] Mukhopadhyay, M. *Structural Dynamics: Vibrations & Systems*. New Delhi: Ane Boks India, 2008. Print.
- [8] Newman, E. S. *Piezoelectric Accelerometers in Structural Health Monitoring*. University of Massachusetts, Lowell, May 2010. Print.
- [9] Kessler, S. S., Spearing, S. M. *In-Situ Sensor-Based Damage Detection of Composite Materials for Structural Health Monitoring*. AIAA Structures, Structural Dynamics, and Materials Conference. April 2002. Web.
- [10] Kosmatka, J. B., Oliver, J. *Development of an In-Flight Structural Health Monitoring System for Composite Unmanned Aircraft*. AIAA Structures, Structural Dynamics, and Materials Conference. May 2006. Web.
- [11] Giurgiutiu, V., Santoni-Bottai, G. *Structural Health Monitoring of Composite Structures with Piezoelectric-Wafer Active Sensors*. AIAA Structures, Structural Dynamics, and Materials Conference. March 2011. Web.
- [12] Lopez, M. D. *Modal Analysis of Composite Laminated Plates under Dynamic Loading*. Thesis. California Polytechnic State University, San Luis Obispo, June 2007. Print.
- [13] Saguibo, L. *Modal Analysis of Composite Laminated Plates with Delaminations under Static and Dynamic Loading*. Thesis. California Polytechnic State University, San Luis Obispo, November 2008. Print.
- [14] Tremaine, K. *Modal Analysis of Composite Structures with Damping Material*. Thesis. California Polytechnic State University, San Luis Obispo, June 2012. Print.

- [15] Rider, K. A. *The Effect of a Low-Velocity Impact on the Flexural Strength and Dynamic Response of Composite Sandwiches with Damage Arrestment Devices*. Thesis. California Polytechnic State University, San Luis Obispo, August 2012. Print.
- [16] "ACG LTM 40 Series Component Prepregs." *Advanced Composites Group PDS 1042* (2007): 1-14. Print.
- [17] "LAST-A-FOAM FR-6710 RIGID POLYURETHANE FOAM Technical Datasheet." *General Plastics Manufacturing Company* (2006): 1. Print.
- [18] "HexWeb HRH-10 Aramid Fiber/Phenolic Resin Honeycomb Technical Datasheet." *Hexcel* (2006): 1-6. Print.
- [19] "MTP-8, 10 Versatile Bench Top-Precision Press." *Tetrahedron Associates Inc.* (2003): 1-1. Web.
- [20] "Standard Test Method for Tensile Properties of Polymer Matrix Composite Materials." *ASTM International* (2008): 1-13. Print.
- [21] Instron Corporation. *Instron 100kN Hydraulic Wedge Grips Operator's Manual*. N.p.: Instron Corporation, 1994. Print.
- [22] "Standard Test Method for Poisson's Ratio at Room Temperature." *ASTM International* (2010): 1-3. Print.
- [23] Vishay Measurements Group, Inc. *Strain Gage Installations with M-Bond 200 Adhesive*. USA: Vishay Measurements Group, 1979. Print.
- [24] Amini, A. J. *Static and Fatigue Failure Response of Woven Carbon Fiber Specimens with Double-Edge Notches*. Thesis. California Polytechnic State University, San Luis Obispo, December 2010. Print.
- [25] Foley, J., Cheney, L. *Notes on Operating Aero Structures/Composites Lab Vibration Table*. California Polytechnic State University, San Luis Obispo, February 2011. Print.
- [26] "PCB Piezotronics Accelerometer, ICP Model Number 353B04 Technical Datasheet." *PCB Piezotronics Vibration Division* (2007): 1. Print.
- [27] "VIP Sensors Piezoelectric Accelerometer Model 1011A Technical Datasheet." *VIP Sensors* (2011): 1-2. Print.
- [28] "Standard Test Method for Vibration (Horizontal Linear Motion) Test of Products." *ASTM International* (2009): 1-4. Print.
- [29] "Getting Started with Abaqus v6.8." *Dassult Systems* (2008): 1-645. Print.

- [30] "COSMOS/M: A Complete Finite Element Analysis System v.2.7." *Structural Research & Analysis Corp* (2001): 1-770. Print.
- [31] Vasiliev, Valery V., and Evgeny V. Morozov. *Mechanics and Analysis of Composite Materials*. New York: Elsevier Science, 2001. Print.
- [32] Chopra, Anil K. *Dynamics of Structures: Theory and Applications to Earthquake Engineering*. Upper Saddle River, NJ: Prentice Hall, 2012. Print.
- [33] Barbero, E. "A Micromechanics Approach for Damage Modeling of Polymer Matrix Composites." *Composite Structures* 67.4 (2005): 427-36. Print.
- [34] Gere, James M. *Mechanics of Materials*. Belmont (CA.): Brooks/Cole Thomson Learning, 2004. Print.
- [35] Davis, Richard A. *The Effects of a Damage Arrestment Device on the Mechanical Behavior of Sandwich Composite Beams under Four-Point Bending*. Thesis. California Polytechnic State University, San Luis Obispo, May 2011. Print.
- [36] Staszewski, W. J., Mahzan, S., Traynor, R. *Health monitoring of aerospace composite structures – Active and passive approach*. Composite Science Technology. December 2007. Journal.
- [37] Coppotelli, G., Marzocca, P., Behal A. *Structural Health Monitoring Techniques for Aerospace Applications*. AIAA. April 2005. Conference.
- [38] Park, Chun, Peters, Kara. *Optimization of Embedded Sensor Placement for Structural Health Monitoring of Composite Airframes*. AIAA. November 2012. Conference.
- [39] Composite Materials Handbook. Materials Usage, Design, and Analysis. [Washington, D.C.]: U.S. Dept. of Defense, 2002. Print.

Appendix

A.1 Vibration Response of Delaminated Composite Structures (Left and Right Sensor Location)

A.1.1 Carbon fiber plates

The L2 and L5 delamination cases were omitted from the plots because the specimens were thought to be damaged in between tests. No visible damages were found. Upon remeasurement, the natural frequencies were 15 sample points below original measurements, indicating damage in the composite structure.

i. Left

i. Right

A.1.2 Foam sandwich panels

i. Left

ii. Right

A.1.3 Honeycomb sandwich panels

i. Left

ii. Right

A.2 ABAQUS Tutorial

A.2.1 Carbon fiber plates

1.

A.2.2 Foam sandwiches

A.2.3 Honeycomb sandwiches