

UC San Diego

UC San Diego Electronic Theses and Dissertations

Title

Simplified modeling methods for mechanically fastened connections in flight structures

Permalink

<https://escholarship.org/uc/item/0nb5n5k8>

Author

Brewer, Brett Andrew

Publication Date

2012

Peer reviewed|Thesis/dissertation

UNIVERSITY OF CALIFORNIA, SAN DIEGO

Simplified Modeling Methods for Mechanically Fastened Connections in Flight
Structures

A Thesis submitted in partial satisfaction of the requirements for the degree
Master of Science

in

Structural Engineering

by

Brett Andrew Brewer

Committee in charge:

Professor John B. Kosmatka, Chair
Professor Hyonny Kim
Professor Chia Ming Uang

2012

The Thesis of Brett Andrew Brewer is approved and it is acceptable in quality and form for publication on microfilm and electronically:

Chair

University of California, San Diego

2012

DEDICATION

This Thesis is dedicated to my future wife, Melanie Daly.

Without her continuous support, this would not have been possible.

TABLE OF CONTENTS

SIGNATURE PAGE	iii
DEDICATION	iv
TABLE OF CONTENTS	v
LIST OF NOMENCLATURE.....	vii
LIST OF FIGURES.....	viii
LIST OF TABLES.....	xii
ACKNOWLEDGEMENTS	xiii
ABSTRACT OF THE THESIS	xiv
CHAPTER 1: INTRODUCTION	1
1.1 OVERVIEW OF THE PROBLEM	1
1.1.1 <i>The Solution Approach</i>	1
1.1.2 <i>Validation Methods</i>	2
1.1.3 <i>Limitations to the Model and Suggestions for Future Development</i>	3
1.2 OUTLINE OF THE THESIS	4
CHAPTER 2: REVIEW OF LITERATURE AND PRIOR RESEARCH	7
2.1 INDUSTRY STANDARDS FOR FASTENER GEOMETRY	7
2.1.1 <i>Industry Standards for Bolt and Rivet Geometry</i>	7
2.1.2 <i>Edge Distance Requirements</i>	9
2.2 INDUSTRY GUIDELINES FOR FASTENER SPACING	11
2.2.1 <i>One Dimensional Spacing</i>	11
2.2.2 <i>Multi-Dimensional Fastener Spacing</i>	12
2.3 SIMPLIFIED MODELING APPROACHES	13
2.3.1 <i>Simplified In-Line Shear Loading Model</i>	13
2.3.2 <i>Eccentric Shear Loading Model</i>	14
2.3.3 <i>NACA Single Row Model</i>	15
2.4 FINITE ELEMENT METHOD MODELING APPROACHES.....	16
2.4.1 <i>Rigid Body Elements</i>	17
2.4.2 <i>Multi-Point Constraint Elements</i>	17
2.4.3 <i>Spring and Beam Elements</i>	18
2.4.4 <i>Explicit Fastener Elements</i>	20
2.4.5 <i>Three Dimensional Solid Elements</i>	21
CHAPTER 3: JOINT THEORIES AND FINITE ELEMENT DEVELOPMENT	23
3.1 BOLTED JOINT DEVELOPMENT	23
3.1.1 <i>Advantages and Disadvantages of Bolted Joints</i>	23
3.1.2 <i>Analysis of Mechanical Connections</i>	25
3.2 BONDED JOINT DEVELOPMENT	27
3.2.1 <i>Advantages and Disadvantages of Bonded Joints</i>	28
3.2.2 <i>Analysis of Bonded Connections</i>	29
3.3 FINITE ELEMENT MODELING APPROACH	32

3.3.1 Effective Material Property Approach.....	33
3.3.2 Solid Element Modeling Approach	36
3.3.3 Beam Element Modeling Approach	37
3.3.4 Spring Element Modeling Approach.....	38
3.3.5 Multiple Fastener Modeling Approach via Connector Pad.....	38
CHAPTER 4: FINITE ELEMENT ANALYSIS AND RESULTS.....	41
4.1 FINITE ELEMENT METHODOLOGY AND SETUP	41
4.1.1 The Specimens	42
4.1.2 Modeling the Specimens in FEMAP	44
4.1.3 Modeling the Specimens in FRANC2D/L.....	50
4.1.4 A Representative FRANC 2D/L Model	53
4.2 FINITE ELEMENT RESULTS	54
CHAPTER 5: EXPERIMENTAL TESTING AND RESULTS	60
5.1 THE PHYSICAL SPECIMENS.....	60
5.2 THE TEST SETUP	61
5.2.1 Preparing the Specimens	62
5.2.2 Preparing the Test Machine.....	69
5.3 EXPERIMENTAL RESULTS.....	75
5.3.1 End Piece Displacement Correction.....	75
5.3.2 Results of the Experiments	78
5.4 EXPERIMENTAL CORRELATION WITH COMPUTATIONAL MODELING	83
5.4.1 Correlation between FEMAP Modeling and Test Results	83
5.4.2 Correlation between FRANC 2D/L Modeling and Test Results	92
5.5 LINEAR VERSUS NONLINEAR EXPERIMENTAL BEHAVIOR.....	93
CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS.....	96
6.1 CONCLUSIONS DRAWN.....	96
6.2 RECOMMENDATIONS FOR FUTURE RESEARCH	99
APPENDIX I: END PIECE DESIGN.....	103
REFERENCES	111

LIST OF NOMENCLATURE

d	Diameter of fastener
e	Edge distance of fastener
p	Pitch (longitudinal spacing) of fasteners
F, P	Point load on joint
N_x	Widthwise load on jointed structure
$S_{x,y}$	Spacing of fasteners in x, y (longitudinal, transverse) direction
P^{br}, P^s, P^t	Fastener loads in bearing, shear, and net tension
E_o, E_i	Modulus of the outer, inner connected angle
$t_1(t_o), t_2(t_i)$	Thickness of outer, inner connected angle
G_a	Shear modulus of adhesive
N_o, N_i	Widthwise load in outer, inner connected angle
τ_a	Shear stress in adhesive
Δ	Fastener displacement
h	Joint eccentricity
E_f, G_f	Fastener Young's modulus and shear modulus
A, I	Fastener area and cross sectional moment of inertia
K	K factor for solid element modeling
ν	Poisson's ratio of the angle material
E_{eff}, G_{eff}	Effective moduli for solid element development
$2c$	Overlap length of bonded joint

LIST OF FIGURES

Figure 1-1: Nested Angle Parasolid Model with 8 Fasteners	2
Figure 2-1: Industry Standard Bolt Head Styles (Reithmaier)(1999)	8
Figure 2-2: Standard Head Styles for Solid Shank Rivets (Reithmaier)(1999)	9
Figure 2-3: Edge Distance Illustration (Reithmaier)(1999)	9
Figure 2-4: Minimum Edge Distance Specification (Reithmaier)(1999)	9
Figure 2-5: Stiffener Riveted to a Plate (Cutler)(1999).....	10
Figure 2-6: Stiffener Bolted to Plate (Cutler)(1999)	11
Figure 2-7: Minimum One-Dimensional Fastener Spacing (Cutler)(1999)	12
Figure 2-8: Multi-Direction Fastener Spacing (Younger, Rice and Ward) (1935)	13
Figure 2-9: Eccentric Shear Loading Model (Rosenfeld)(1947).....	15
Figure 2-10: Symmetric Butt Joint Analyzed By (Rosenfeld)(1947)	15
Figure 2-11: Single Row Riveted Joint a) Side b) Plan (Naarayan, Kumar and Chandra) (2009).....	16
Figure 2-12: Rivet Load Distribution for FEM with Rigid Body Elements (Naarayan, Kumar and Chandra) (2009).....	17
Figure 2-13: Rivet Load Distribution for FEM with MPC Elements (Naarayan, Kumar and Chandra) (2009).....	18
Figure 2-14: Rivet Load Distribution for FEM with Spring and Beam Elements (Naarayan, Kumar and Chandra)(2009)	19
Figure 2-15: Single Lap Joint Modeled With Explicit Fastener Elements and Adhesive Elements (Cope and Lacy)(2000).....	20
Figure 2-16: Axial and Shear Stress Contour Capabilities of Solid Element Modeling (Fung and Smart)(1994)	22
Figure 2-17: Three-Dimensional FEM of Countersunk and Snap Rivets in Single Lap Joints (Fung and Smart)(1994).....	22
Figure 3-1: Various Mechanically Fastened Failure Mechanisms (Kim and Kedward)(2004)	25
Figure 3-2: Illustration of a Single Row Versus Multi-Row Fastener Configuration..	26
Figure 3-3: Representative Mechanically Fastened Element (Niu)(2009).....	27
Figure 3-4: Representative Single Lap Bonded Joint.....	30
Figure 3-5: Load Transfer Via Shear in Bonded Joints.....	30
Figure 3-6: Various Modeling Techniques Varying from Entire Connector, to Spring Elements (Kosmatka, Brewer, Pun, and Hunt (2012)).....	33
Figure 3-7: Solid Element Representation of a Fastener (http://en.wikipedia.org/wiki/Bolted_joint).....	36
Figure 3-8: Beam Element Representation of a Fastener (http://en.wikipedia.org/wiki/Bolted_joint).....	37

Figure 3-9: Multiple Fasteners Converted into an Equivalent Pad (Kosmatka, Brewer, Pun, and Hunt) (2012)	39
Figure 3-10: Effect of Fastener Density on Connector Pad Definition (Kosmatka, Brewer, Pun, and Hunt) (2012)	39
Figure 3-11: Unrealistic Continuous Load Transfer	40
Figure 4-1: Large and Small Angle Cross Sections (Kosmatka, Brewer, Pun, and Hunt) (2012)	41
Figure 4-2: Two and Eight Fastener Test Specimens in Nested Configuration (Kosmatka, Brewer, Pun, and Hunt) (2012).....	42
Figure 4-3: Hi-Lok Fastener Section View (Hi-Shear Corporation).....	43
Figure 4-4: Blind Rivet with Hollow Cross Section in the Shear Plane (Marson and Gesipa)(2012)	46
Figure 4-5: Two Fastener Finite Element Mesh with Spring Elements	47
Figure 4-6: Four Fastener Finite Element Mesh with Spring Elements	48
Figure 4-7: Six Fastener Finite Element Mesh with Spring Elements	48
Figure 4-8: Eight Fastener Finite Element Mesh with Spring Elements.....	48
Figure 4-9: Eight Fastener Connection Region with Two Connection Zones	49
Figure 4-10: Eight Fastener Connection Region with Solid Element Fastener Approximations	49
Figure 4-11: Eight Fastener Connection Region with Timoshenko Beam Elements... 49	
Figure 4-12: Flattened FRANC 2D/L Nested Angle Model	51
Figure 4-13: FEMAP Equivalent G Verification Mesh	52
Figure 4-14: FEMAP Equivalent G Model Boundary Conditions and Loads	52
Figure 4-15: FEMAP Equivalent G Results	52
Figure 4-16: FRANC 2D/L Equivalent G Results	53
Figure 4-17: FRANC 2D/L Model of Eight Connector Specimen.....	54
Figure 4-18: Hi Lok and Stainless Steel Displacement Estimates	58
Figure 4-19: POP Rivet Displacement Estimates.....	58
Figure 5-1: Strain Gage Locations for NHL6-01 (Fasteners in Red).....	63
Figure 5-2: General Strain Gage Labeling Convention on Outside Surfaces	63
Figure 5-3: General Strain Gage Labeling Convention on Outside Surfaces	64
Figure 5-4: Load versus Displacement Cycles for NHL6-01.....	64
Figure 5-5: Axial Strain Data Between Fasteners	65
Figure 5-6: Far Field Axial Strain for Large Angle	65
Figure 5-7: Far Field Axial Strain for Small Angle	66
Figure 5-8: Opposing Strain Differences for NHL6-01	66
Figure 5-9: Radius of Curvature for Test Specimen	68
Figure 5-10: 110 Kip Material Testing System Machine.....	69
Figure 5-11: Native Data Acquisition Software	70

Figure 5-12: Secondary Data Acquisition System	71
Figure 5-13: End Piece Grips (with Fasteners)	72
Figure 5-14: Representative Specimen in the Test Machine	73
Figure 5-15: Typical Load versus Time Profile	74
Figure 5-16: NHL4-01 with Displacement Transducers	76
Figure 5-17: Load versus Displacement with Displacement Transducers	77
Figure 5-18: 2 Fastener Load versus Displacement Data.....	79
Figure 5-19: 2 Fastener Load versus Displacement Slopes.....	79
Figure 5-20: 4 Fastener Load versus Displacement Data.....	80
Figure 5-21: 4 Fastener Load versus Displacement Slopes.....	80
Figure 5-22: 6 Fastener Load versus Displacement Data.....	81
Figure 5-23: 6 Fastener Load versus Displacement Slopes.....	81
Figure 5-24: 8 Fastener Load versus Displacement Data.....	82
Figure 5-25: 8 Fastener Load versus Displacement Slopes.....	82
Figure 5-26: Load vs Displacement Comparison for Various Modeling Techniques for 2 Fastener Specimens (NHL and NSS)	88
Figure 5-27: Load vs Displacement Comparison for Various Modeling Techniques for 2 Fastener Specimens (NPR).....	88
Figure 5-28: Load vs Displacement Comparison for Various Modeling Techniques for 4 Fastener Specimens (NHL and NSS)	89
Figure 5-29: Load vs Displacement Comparison for Various Modeling Techniques for 4 Fastener Specimens (NPR).....	89
Figure 5-30: Load vs Displacement Comparison for Various Modeling Techniques for 6 Fastener Specimens (NHL and NSS)	90
Figure 5-31: Load vs Displacement Comparison for Various Modeling Techniques for 6 Fastener Specimens (NPR).....	90
Figure 5-32: Load vs Displacement Comparison for Various Modeling Techniques for 8 Fastener Specimens (NHL and NSS)	91
Figure 5-33: Load vs Displacement Comparison for Various Modeling Techniques for 8 Fastener Specimens (NPR).....	91
Figure 5-34: Various Section Displacements of FRANC 2D/L Eight Fastener Model Under 1000 lbf Load	92
Figure 5-35: Linear and Nonlinear Response of NHL 2-03.....	94
Figure 5-36: Linear and Nonlinear Response of NHL8-03.....	94
Figure AI-1: Large Angle Cross Sectional Geometry	103
Figure AI-2: Small Angle Cross Sectional Geometry	103
Figure AI-3: Nested Section Centroid Location.....	104
Figure AI-4: End Piece Cross Section.....	106
Figure AI-5: End Piece Bolting Configuration (Dimensions in Inches)	107

Figure AI-6: End Piece Design Cross Section	110
Figure AI-7: End Piece Design Isometric	110

LIST OF TABLES

Table 2-1: Rivet Load Distribution in Five Fastener Lap Joint (Naarayan, Kumar and Chandra) (2009).....	19
Table 3-1: K Factors for Solid Element Fastener Deflections	34
Table 3-2: Empirically Derived Fastener Constants (Cope and Lacy)(2000), (Swift)(1984)	35
Table 4-1: Effective Fastener Properties for FEM	55
Table 4-2: Hi-Lok and 18-8 Stainless Steel Model Displacement Results in FEMAP	56
Table 4-3: Aluminum POP Rivet Model Displacement Results in FEMAP.....	57
Table 5-1: Displacement of Deformable Sections for NHL4-01	77
Table 5-2: Axial Displacement Correlation Results.....	85
Table AI-1: End Piece Strength Checks	109

ACKNOWLEDGEMENTS

First and foremost, I would like to thank Dr. John Kosmatka, as his help and guidance has been proven invaluable throughout my time researching and writing at this university. Without his expertise and experience, this Thesis could never have been completed.

I would also like to thank Dr. Hyonny Kim and Dr. Chia Ming Uang for participating as members of my committee. They have also been extremely helpful in developing the analytical models and test apparatuses throughout the research.

Finally, I would like to thank Eduardo Velazquez and Benjamin Martins for all of their help, both past and present. Much of what will be presented in this Thesis was developed from a framework laid out by Eduardo Velazquez, and appears primarily in Chapter 2. Benjamin Martins helped complete much of the testing during this project, primarily with specimen preparation.

ABSTRACT OF THE THESIS

Simplified Modeling Methods for Mechanically Fastened Connections in Flight Structures

by

Brett Andrew Brewer

Master of Science in Structural Engineering

University of California, San Diego, 2012

Professor John B. Kosmatka, Chair

Simplified modeling and analyses methods for aerospace connectors have been around for the better part of 30 years, but continue to evolve with technology. For the preliminary design process, the modeling of individual fasteners is time consuming and unforgiving, as the fastener sizes and locations often change during the process, necessitating a new model each iteration. Further simplifying the design process

introduces a so called connector pad, which acts as a representative for entire groups of fasteners within a finite element model. Expanding upon previous developments from the last quarter century, this new method creates small groups of solid elements that are able to accurately represent any number of fasteners that exist within that region. Using FEMAP and NX NASTRAN to validate extensive linear displacement test data, this connector pad can be seen as another successful way to model mechanical connectors in flight structures. The success of the model lies strictly in linear displacements, however. Stresses, strengths, and margins of safety are not considered in this Thesis, though it is possible to extract such data. A group of specimens, consisting of varying state of the art connector types such as Hi-Lok fasteners, and ranging in connection density from two to eight fasteners, was tested axially within the linear elastic range. This data was compared to several different computer based fastener modeling techniques, such as using springs, beams, solids, or the new connector pad to represent the fastener. Not only does this study work to further validate the established techniques, but it also gives rise to the possibilities and advantages of a new, more simplified approach to modeling mechanical fasteners in aircraft.

CHAPTER 1: INTRODUCTION

Finite element modeling has grown into a design and analysis powerhouse in the engineering world in the last half century, allowing engineers to answer complex problems that would be impossible to solve without the speed and computing power of modern technology. This method is heavily used in the field of aerospace structural engineering, where large scale testing is much too expensive. The joints within these aerospace structures pose a large problem for the finite element process; both in modeling time and computational expense, and beg for simplified modeling solutions.

1.1 Overview of the Problem

A primary concern for modeling mechanically fastened connections for aerospace flight structures is the speed with which they can be modeled. Historically, each fastener has been modeled individually by one of several different element types and formulations ranging from simple spring elements to beams and solids, with the latter representing the most complete, yet most time consuming modeling method.

1.1.1 The Solution Approach

The goal of this Thesis is to demonstrate that rows of fasteners can be modeled much more quickly for the preliminary design process using a layer of solid elements called a connector pad that cover the area that all of the fasteners are contained within. This means that less effort must be spent in locating where fasteners exist within the connection area, since the exact location often changes throughout the design process. The fact that the connectors are in the connection area changes the effective properties

of the solid elements that span the connection area. This method is based on an adaptive scaling approach that modifies the effective properties of the fasteners based on the density and type of fasteners within the total connection region.

1.1.2 Validation Methods

In order to test the hypothesis that the connector pad is a valid way of modeling mechanically fastened connections, several tests were performed with various numbers and types of modern connectors ranging from Hi-Lok rivets to stainless steel bolts. The test specimens are composed of two 16 inch long L channels made of aluminum 6061 T6 that are fastened together using anywhere from two to eight of the aforementioned connectors (See Figure 1-1).

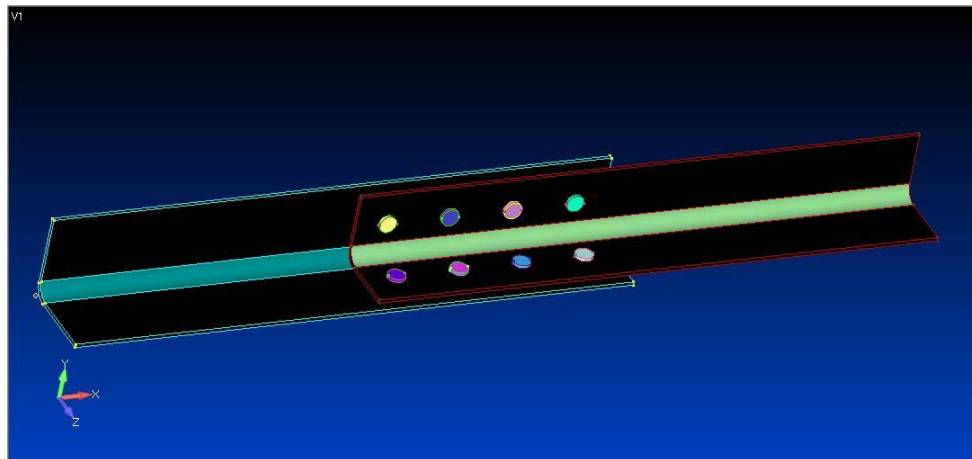


Figure 1-1: Nested Angle Parasolid Model with 8 Fasteners

The tests that were performed were quasi-static axial loading tests in which the nested angle specimens were inserted into a 110 kip test machine, and tested within the elastic range of the material, in this case to 1000 pounds of force. The test results from the physical specimens were then compared to many different finite element

models of the same specimen, each with a different method of modeling the fastener in the attempt to show that the simple connector pad performed as well as some of the more complicated modeling techniques.

1.1.3 Limitations to the Model and Suggestions for Future Development

The development of the connector pad is in its adolescence, and has not been expanded to accommodate for several factors. As mentioned above, the tests were all done strictly within the linear regime of the material. Therefore, beginning at the onset of local yielding around the fastener holes, the approach begins to break down. While it would not be expected that an actual structure would be loaded into the plastic regime, designing with some plasticity in mind is sometimes desirable. Additionally, the model does not account for such parameters as bolt preloading or other frictional effects. This is primarily a concern in nonlinear analyses, which is not the focus of the current research effort. Finally, the only joint type that was analyzed and testing during this research was single lap joints. No explicit relationships were developed for double or higher lap joint configurations, though the principles of scaling the fastener properties to match the new connector pad area would remain the same. In a similar fashion, if various attached pieces had differing hole sizes or tolerances, then the bearing loads would differ between layers, presenting another breaking point for the model. All of these areas present opportunity to further develop the connector pad in future research.

1.2 Outline of the Thesis

This Thesis will guide the reader through all of the current state of the art modeling techniques, present the development of the connector pad, and compare its finite element results to the testing done on the nested angel specimens. The remainder of the Thesis is broken up into five chapters and an appendix and is organized in a fashion that moves from industry standards and current state of the art to new developments and testing.

Chapter two covers the industry standards in aerospace structure geometry and common fastener types. This is an important baseline from which to start the discussion as it demonstrates that all of the testing and analysis that was done is consistent with the currently established guidelines that govern the aerospace industry. Such standards include fastener spacing, edge distance, diameter, material, et cetera. Additionally, some special cases are covered that are relevant to the particular specimens that were tested, such as the edge distance of a fastener to a rolled edge.

Chapter three covers many developments of joint theories as well as the current state of the art in finite element modeling for such joints. First, mechanically fastened joints are developed. Advantages and disadvantages of this type of joint are explained to aid in understanding the types of scenarios in which they may be used. Then, the ideas of load sharing, failure modes, and basic design are outlined such that later developments in the finite element method can draw from them. Next, the Volkersen (1938) model, an extension of the shear lag model, is discussed. This has long been an effective model for the analysis of bonded joints. Since the connector

pad will behave similar to an adhesive bond, this model is important to the understanding of the load transfer mechanism in the connector pad. Finally, the different state of the art methods of modeling mechanical fasteners is discussed. Several authors have developed methods for modeling such connectors in the past using beam elements, springs elements, or groups of solid elements in a full 3D model. The work of these authors is not used only as history, but rather expanded upon to develop the connector pad model.

Chapter four will outline the methods with which the nested angle specimens were modeled. As mentioned in the previous section, many models were made in order to compare the various modeling methods against the test data. Details of these models will contribute to the repeatability of this research for future progressions. Additionally, the results of the finite element analyses will be discussed in this chapter. These results will be compared to the test results in the following chapter.

Chapter five, as mentioned above, will focus on presenting the test results from the nested angle testing. The correlations will be shown between the many finite element analysis models that were developed, including those using the connector pad to join the two angles.

In chapter six, final conclusions will be drawn from the test data and its correlation to the finite element results. The implications of the findings will be outlined, including the potential for the use of the connector pad idea in industry applications for the preliminary design process. Additionally, the framework for future research will be laid out, such that the idea of the connector pad can continue to

be expanded upon and made more of a complete design tool in the years to come. These recommendations will include some of the ideas mentioned in the previous section in greater detail in order to guide more explicitly the future of the project.

Lastly, an appendix will be included that will include miscellaneous designs, setups, et cetera. One example is the design of the grips that held the nested angles in the test machine. These grips were specially designed in order to minimize the effects of bending in the angles during the axial testing.

CHAPTER 2: REVIEW OF LITERATURE AND PRIOR RESEARCH

The design of fasteners used on aircraft follows strict set of well established guidelines that have been generally accepted and unchanged for several decades. These guidelines are recommendations ranging from fastener diameters, spacing, and edge distances, to jointed member geometry and material composition. These specifications dictate not only the ways that aircraft structures are built, but naturally, how they are analyzed. This is inclusive of many large commercial aircraft, which also includes an adhesive between the layers of composite in addition to mechanical fasteners, aiding in the shear transfer to improve fatigue life. This dual shear transfer was not researched in this project. Several authors have developed simplified models for fastened joints in recent years, ranging from strength of materials approaches to closed form analytical solutions. Further discussion will follow on these methods in section 2.3.

2.1 Industry Standards for Fastener Geometry

Before delving into the modeling of mechanically fastened joints, one should first briefly review the industry standards that govern the construction of metallic aircraft.

2.1.1 Industry Standards for Bolt and Rivet Geometry

Conventional bolts in many aerospace structures, such as the fuselage and wing structures, range between $\frac{3}{16}$ and $\frac{9}{16}$ inches in nominal diameter. Various types of

heads are available including but not limited to: fillister, brazier, MS type, pan, 12-point tension, flush head, hexagonal, and 12 point shear as shown in Figure 2-1.

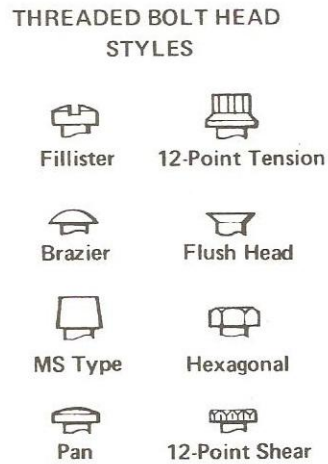


Figure 2-1: Industry Standard Bolt Head Styles (Reithmaier)(1999)

Like bolts, rivets come in a variety of sizes, ranging from $\frac{1}{16}$ and $\frac{3}{8}$ inches most commonly. Rivets come in two main types, solid-shank and blind rivets. Solid-shank rivets have several head styles, including: universal, round, brazier, countersunk, and flat, as seen in Figure 2-2. Blind rivets have three main categories of their own: hollow or pull through, self-plugging friction locking, and self-plugging mechanically locked. The first type is used in nonstructural applications, while the second and third are used in place of standard solid shank rivets. The primary difference between the latter two is in the vibration resistance of the fastener. The locking mechanism provides a greater resistance to vibration than those rivets that use friction along to lock.

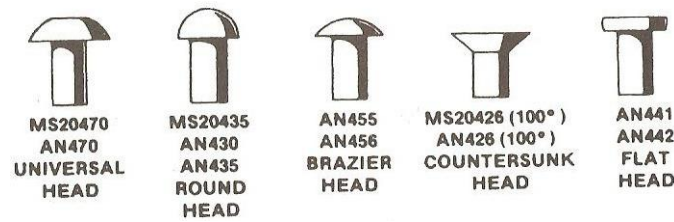


Figure 2-2: Standard Head Styles for Solid Shank Rivets (Reithmaier)(1999)

2.1.2 Edge Distance Requirements

Edge distance is defined as the distance from the outermost fastener to the nearest free edge of the attached component as illustrated in Figure 2-3. Bolts and rivets seem to have similar requirements for this value. Both Niu (2009) and Reithmaier (1999) give a minimum value for edge distance of $2d+0.03$ (Figure 2-4). Older guidelines set out by Younger, Rice and Ward (1935) state that the edge distance can go as low as $1.5d$, though this is much less common in modern practice. Some authors have specified requirements on fastener edge distance for specialized joint configurations, rather than outlining general guidelines (Cutler (1999)).

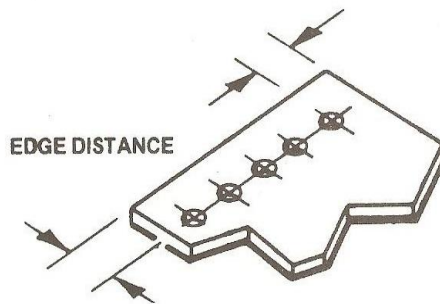


Figure 2-3: Edge Distance Illustration (Reithmaier)(1999)

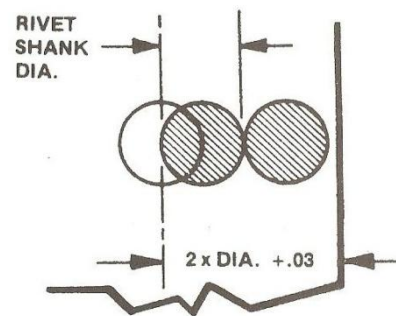


Figure 2-4: Minimum Edge Distance Specification (Reithmaier)(1999)

Cutler (1999) outlines special requirements for edge distance when plates are joined to stiffening elements, with rivets and bolts having different requirements. Figure 2-5 shows the situation with a rivet in place. In this case, Cutler (1999) suggests a value of $A \geq 0.3$ inches and an edge distance of at least 0.25 inches for a $\frac{1}{8}$ inch diameter solid shank rivet (blind rivets may reduce A to 0.25 inches). This edge distance is slightly smaller than that recommended by Niu (1999) and Reithmaier (1999). Both the values of A and edge distance increase by 0.05 inches for a $\frac{5}{32}$ or $\frac{3}{16}$ rivet. Cutler (1999) estimates that if the rivet hole position is off by ± 0.02 inches, that the strength will be reduced by 6.5% for a 0.3 inch edge (A), and by 10% for a 0.2 inch edge.

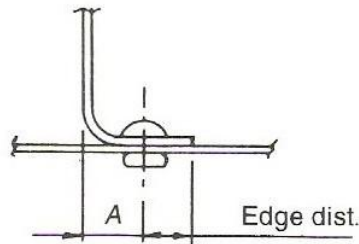


Figure 2-5: Stiffener Riveted to a Plate (Cutler)(1999)

Cutler (1999) has similar recommendations for bolted variations of this configuration (shown in Figure 2-6). The distance A varies in this case with the size of the wrench needed to tighten the bolt. The requirement on the edge distance is simply that it is greater than A . Cutler (1999) recommends that when the edge distance and A

are equal, that the preloading force in the bolt is twice that of the applied load P shown in the figure.

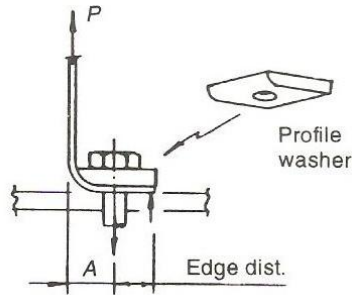


Figure 2-6: Stiffener Bolted to Plate (Cutler)(1999)

2.2 Industry Guidelines for Fastener Spacing

In order to minimize the effects to interaction of stress concentration zones surrounding the fasteners, several guidelines exist to ensure safe and effective spacing of rivets and bolts.

2.2.1 One Dimensional Spacing

When only one line of fasteners is being considered, only the edge distance and the space between each fastener (the pitch) are relevant. Both Cutler(1999) and Niu(1999) give the same recommendation that the spacing should be no less than $4d$ (see Figure 2-7). Younger, Rice and Ward (1935) suggest that for edge riveting, the spacing between two rows of rivets can be as small as $2d$. Other spacing requirements referring to multiple rows of rivets can be found in the next subsection.

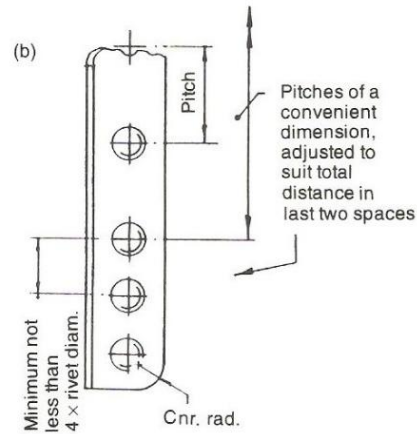


Figure 2-7: Minimum One-Dimensional Fastener Spacing (Cutler)(1999)

2.2.2 Multi-Dimensional Fastener Spacing

Younger, Rice and Ward (1935) give two major spacing provisions for designing riveted connections with multiple rows of fasteners. Figure 2-8 shows the situation that is applicable. Younger, Rice and Ward suggest that the distances a and b should both be at least 1.2 times larger than d in order to prevent crack propagation between holes. In later sections, this topic will be discussed further. Additionally, they suggest that the pitch p is dependent on the application (i.e. small p for gas tanks to prevent leaks). However, we accept this pitch, p to have a lower bound of $4d$, as mentioned in section 2.2.1.

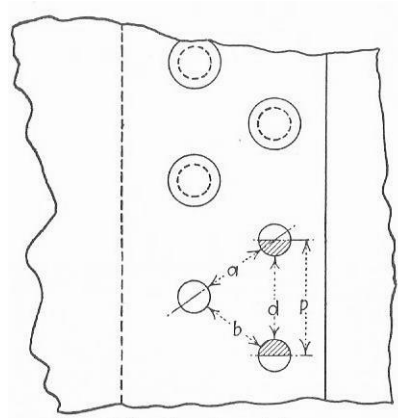


Figure 2-8: Multi-Direction Fastener Spacing (Younger, Rice and Ward) (1935)

Knowledge of the aforementioned industry standards will serve as the backbone for the modeling of fastened connections by helping the modeler to create reasonable geometries.

2.3 Simplified Modeling Approaches

Several researchers have developed methods for modeling riveted connections over the past 75 years, with developments increasingly focused on finite element methods within the last quarter century. There are three primary models that are seen in the literature: a simplified in line shear loading model, an eccentric loading model, and a NACA single row model.

2.3.1 Simplified In-Line Shear Loading Model

Some conventional, simplified analyses of in line shear loaded members have made the key assumption that all fasteners take equal load. However, at the

conceptual level, this is quite a problematic assumption due to the fact that the edge connectors generally take the greatest load (Naarayan, Kumar and Chandra) (2009). Consequently, such an assumption can lead to significant errors and unexpected failures in the member. This phenomenon is apparent in aircraft failures ranging from the Comet air crash of 1970 to the Aloha air crash in 1988. Though these catastrophes were ultimately fatigue related, the damage was found to have initiated at the outermost row of fasteners.

2.3.2 Eccentric Shear Loading Model

This model was inspired by the situation where the loading has some eccentricity from the centroid of the fastener group. Figure 2-9 provides an illustration of such a situation. This model is based on the St. Venant principle of resolving a distant force into a near field force, F and an equivalent, moment producing couple of magnitude F_e . Once the force has been moved, the shear load in each fastener due to the force, F is simply equal to F divided by the number of fasteners, since the load is passing through the centroid of the fastener group. Furthermore, the shear force in each fastener due to the equivalent couple is found using moment equilibrium. It should be noted that this model operates under two main assumptions: 1) that the connecting plates are rigid, and 2) that the fasteners behave elastically.

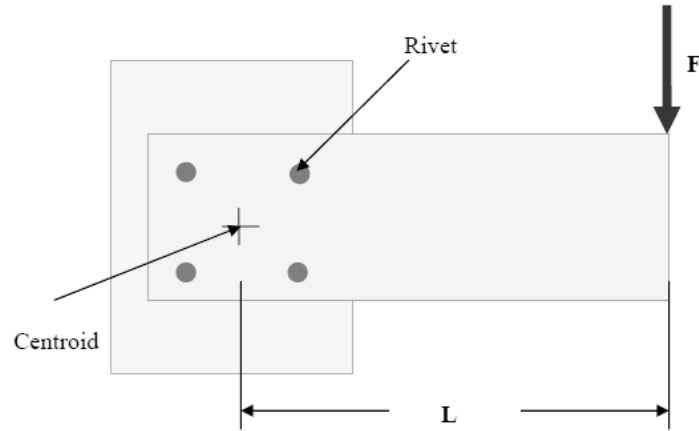


Figure 2-9: Eccentric Shear Loading Model (Rosenfeld)(1947)

2.3.3 NACA Single Row Model

The model proposed by Rosenfeld (1947) allows for the closed form solution of symmetrical butt joints with bolts in line with the axial load. The model formulates a recurrence formula that, in conjunction with the appropriate boundary conditions, is used to obtain sets of simultaneous linear equations that result in the bolt load distribution. An appropriate model problem for this approach is shown in Figure 2-10.

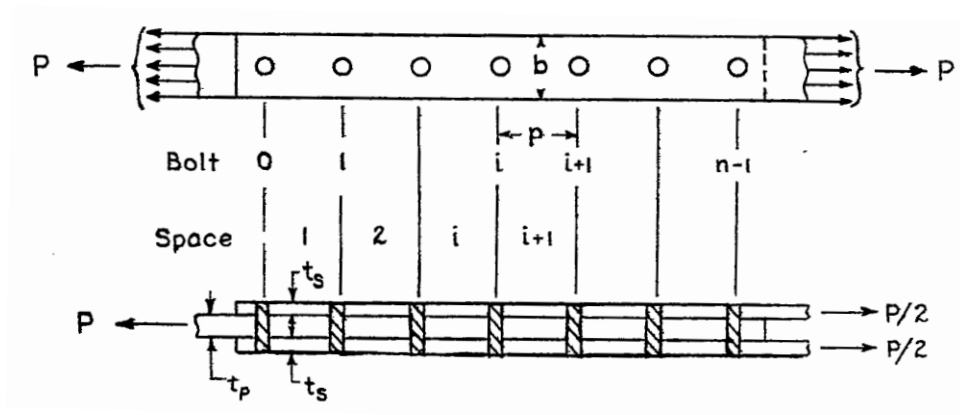


Figure 2-10: Symmetric Butt Joint Analyzed By (Rosenfeld)(1947)

2.4 Finite Element Method Modeling Approaches

Several approaches are identified in current literature for using Finite Element Modeling for riveted/bolted connections. The bibliography presented by Mackerle (2003) studies the use of FEM from 1990 to 2002 for different types of connectors (pins, bolts, rivets, fittings, etc). For a more current view of modeling, Naarayan, Kumar and Chandra (2009) present a short review of FEM modeling of only riveted joints in the elastic regime. The case that was considered is shown in Figure 2-11. Five different approaches are presented here that span the major trends observed in the different types of models (in order of increasing complexity): 1) a rigid element, 2) a multi-point constraint element, 3) a spring or beam element, 4) an explicit fastener element, 5) a complete three-dimensional model.

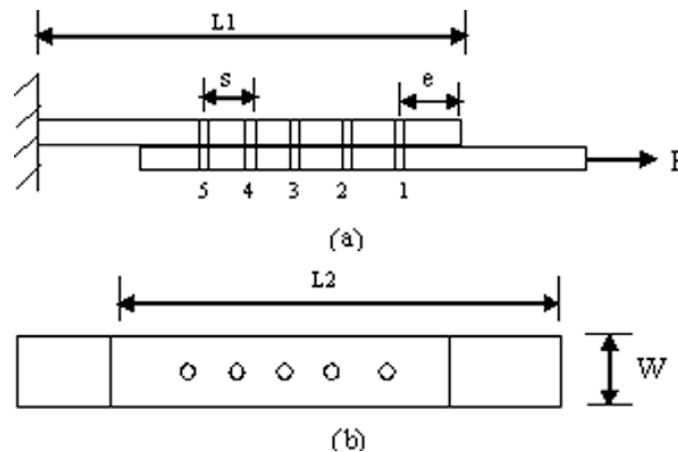


Figure 2-11: Single Row Riveted Joint a) Side b) Plan (Naarayan, Kumar and Chandra) (2009)

2.4.1 Rigid Body Elements

Representing fasteners as rigid links of infinite stiffness will result in correct load distributions among the fasteners, as shown in Figure 2-12. However, due to the disparity between the idealized stiffness (infinite) and the actual fastener stiffness, the displacement of the fasteners and the bearing behavior surrounding the fastener would be inaccurate. Studies performed by Swift (1984) have shown that more than 75% of the overall displacement of a joint comes directly from the fastener displacement, making this model impractical for displacements due to the stiffness disparity.

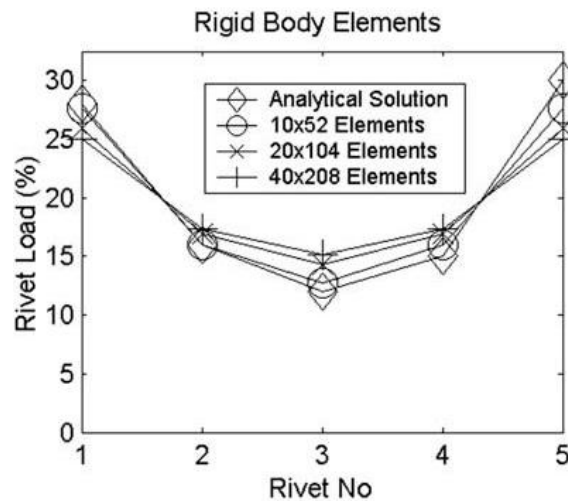


Figure 2-12: Rivet Load Distribution for FEM with Rigid Body Elements (Naarayan, Kumar and Chandra) (2009)

2.4.2 Multi-Point Constraint Elements

Naarayan, Kumar and Chandra have also done work with multi-point constraint (MPC) models (Figure 2-13). These models impose additional constraints

on the two nodes that are attached to the element. These constraints can range from imposing identical displacement and rotation at the two tied nodes to allowing some relative motion between the two. In other words, MPCs behave in a similar fashion to springs, but are much more rigorous in the enforcement of the constraints.

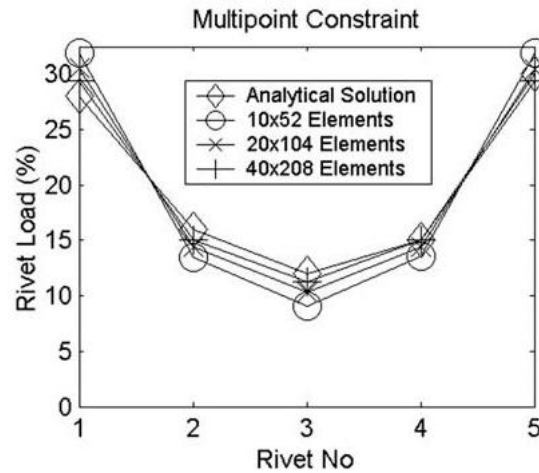


Figure 2-13: Rivet Load Distribution for FEM with MPC Elements (Naarayan, Kumar and Chandra) (2009)

2.4.3 Spring and Beam Elements

The most common method found in the literature is to model fasteners as spring or beam elements (Chen, Wawrzynek and Ingraffea (1999), Swift (1984), Seshadri and Newman Jr.(2000), Naarayan, Kumar and Chandra (2009)). These elements each have 6 degrees of freedom per node corresponding to axial, flexural, and torsional degrees of freedom. The elements are connected to two nodes corresponding to the location of the fastener in the model. These stiffness values are calculated empirically using relationships outlined by Swift (1984), which will be discussed further in chapter three.

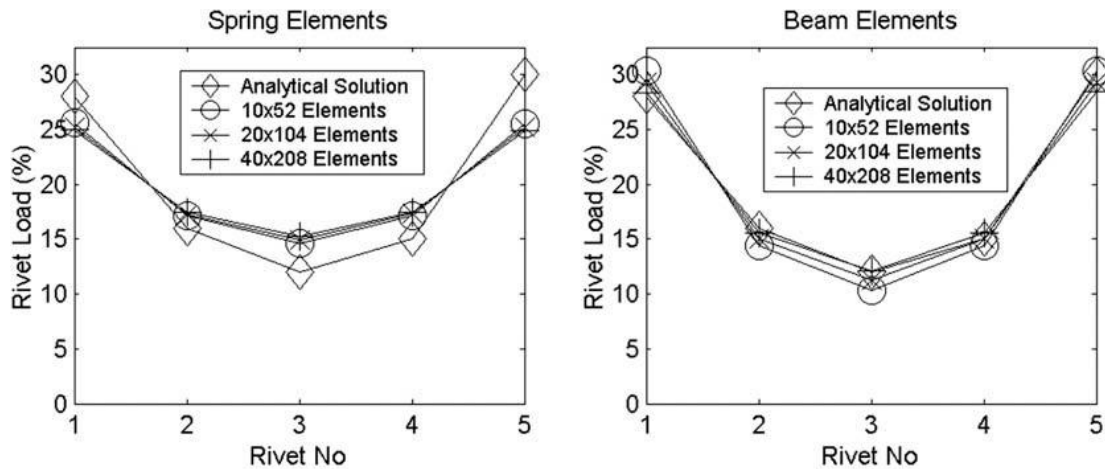


Figure 2-14: Rivet Load Distribution for FEM with Spring and Beam Elements (Naarayan, Kumar and Chandra)(2009)

It can be seen from the work of Naarayan, Kumar, and Chandra that there is an unequal sharing of load among the fasteners, again making clear that simplifying assumptions that all fasteners take equal load are quite inaccurate. Table 2-1 shows the results of Naarayan *et al*'s finite element studies in load sharing. Additionally, it is seen from the previous figures that mesh density does play a role in the load sharing in the finite element analyses. However, Naarayan *et al* considered this effect to be negligible.

Table 2-1: Rivet Load Distribution in Five Fastener Lap Joint (Naarayan, Kumar and Chandra) (2009)

Idealization	Rivet Load/Applied Load				
	Rivet 1	Rivet 2	Rivet 3	Rivet 4	Rivet 5
Analytical	0.28	0.16	0.12	0.15	0.3
Spring	0.25	0.17	0.15	0.17	0.25
Beam	0.28	0.15	0.12	0.15	0.28
RBE	0.25	0.17	0.15	0.17	0.25
MPC	0.29	0.15	0.11	0.15	0.29
Conventional	0.2	0.2	0.2	0.2	0.2

2.4.4 Explicit Fastener Elements

Explicit fastener elements are used by Cope and Lacy (2000) to model riveted single lap joints using the FRANC2D/L finite element software package. These elements are a combination of different elements, notably one dimensional springs, two dimensional adhesive elements, and contact elements. The purpose of the spring element is to discretely transfer the load between the two sides of the joint. The adhesive elements transfer the shear stresses in the region surrounding the connector, while the contact elements serve to add friction into nonlinear problems. These elements are used in different combinations based on the problem type and the analysis that is desired (Figure 2-15). Further development on these elements and FRANC2D/L is present in chapters three and four.

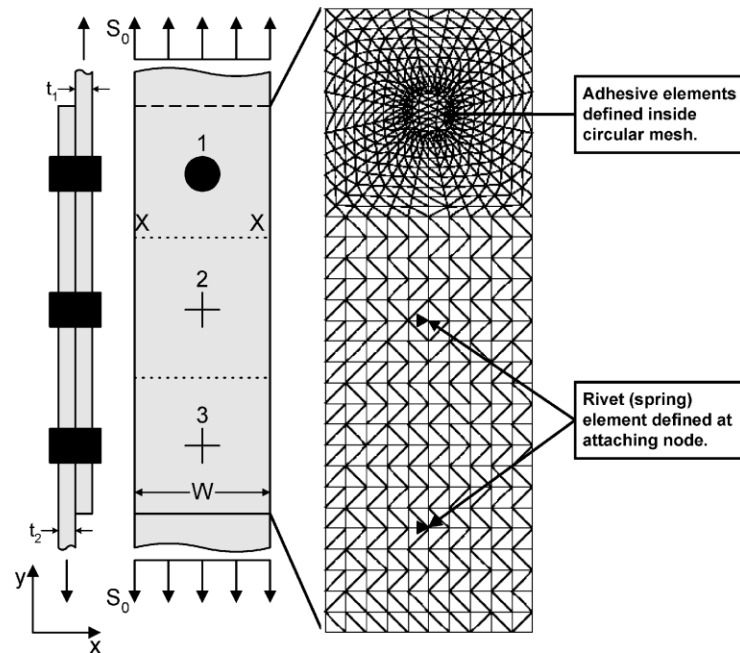


Figure 2-15: Single Lap Joint Modeled With Explicit Fastener Elements and Adhesive Elements (Cope and Lacy)(2000)

2.4.5 Three Dimensional Solid Elements

The final way that the fasteners can be represented is with a detailed model composed of solid elements. These models are well suited for introducing friction, fastener pre load, tolerance, etc, though the modeling and computational cost of using such a method often makes it impractical for even a small number of connectors. Due to the great level of detail, these models can not only handle the geometric nonlinearities from effects such as contact, but can also give detailed stress contours within individual fasteners, something that the simpler models cannot produce (see Figure 2-16) The modeling is generally done by modeling all connected surfaces with interface elements, while modeling the cores of the plates and fasteners with tetrahedral or brick elements. In the case of Fung and Smart (1994), the rivets were given orthotropic properties, with different expansion coefficients in the longitudinal and transverse directions. Examples of fasteners modeled as solids can be seen in the work of Fung and Smart in Figure 2-17 below.

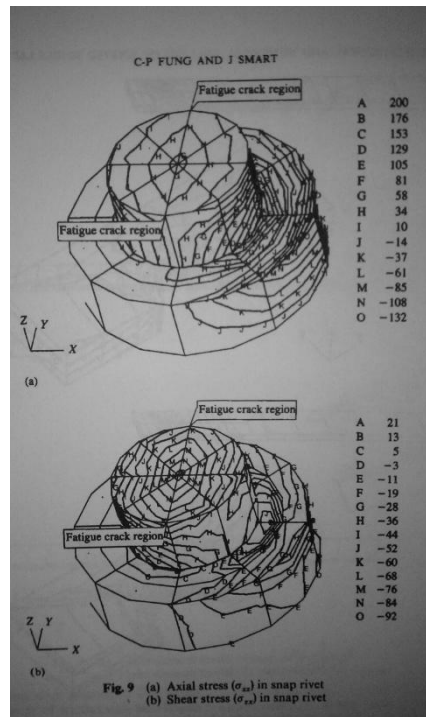


Figure 2-16: Axial and Shear Stress Contour Capabilities of Solid Element Modeling (Fung and Smart)(1994)

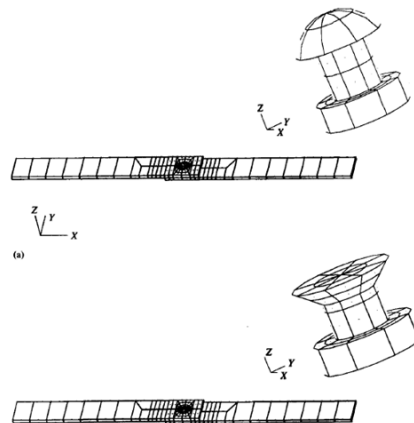


Figure 2-17: Three-Dimensional FEM of Countersunk and Snap Rivets in Single Lap Joints (Fung and Smart)(1994)

CHAPTER 3: JOINT THEORIES AND FINITE ELEMENT DEVELOPMENT

In order to build a foundation for the development and validation of a simplified connector model for mechanically fastened joints, classical joint analysis theories should be reviewed. First, a brief overview of bolted joint analysis is included, followed by background information for the analysis of bonded joints. This chapter will conclude by outlining the various approaches that can be taken in the way of finite element modeling of mechanical connectors and lay the groundwork regarding how they can be translated into a simpler connector pad region.

3.1 Bolted Joint Development

In order to move into the development of a connector pad that effectively replaces mechanical connections, a basis must be built in the behavior of mechanically fastened joints. Joints, by definition, are relatively complex regions of a structure where various parts of the structure connect. Changes in geometry, load path, and materials, are all facilitated within joints. These discontinuities can create stress concentrations, which often lead to the onset of initial failure in connected regions of structural systems.

3.1.1 Advantages and Disadvantages of Bolted Joints

Mechanically fastened joints are used in highly loaded structures. They are very capable of fastening thick walled structures due to the fact that there is hardly a limit to the diameter or length of the fastener used in the application. Given the sense of security and the overall reliability that mechanical connections provide, they are

often chosen for high risk applications such as connecting aircraft wings and fuselages.

Additionally, bolted or riveted connections are tolerant to environmental effects. However, the effects of corrosion are to be a point of concern in the case of using dissimilar materials for the fastener and the fastened parts. They have the advantage of having a repeatable (dis)assembly process, yet have a very high part count compared to a bonded joint, which only requires the adhesive (which may have up to three parts) and the adherends. Bolted connections are also relatively easy to assemble, requiring no special surface treatments while still ensuring immediate load bearing capability. Finally, bolted and riveted connections are relatively easy to inspect for damage, further emphasizing their reliability.

However, mechanically fastened connections do have several attributes that are cause for concern. For example, the holes that must be drilled in order to place the fasteners in create local stress concentrations that may initiate early onset of damage in the area surrounding the hole. To combat this phenomenon, the structure often needs to be thickened locally to guarantee a safe design. This added thickness, coupled with the high volume of metallic fasteners, add a large amount of weight to the structure, giving a low strength/weight efficiency ratio. Cost is also a concern with these types of connections, as they tend to be very labor intensive as well as each fastener must be placed individually.

3.1.2 Analysis of Mechanical Connections

The analysis of mechanically fastened connections requires special attention to many of the parameters that were outlined in the previous chapter, including connector edge distance, spacing, and diameter.

Bolted connections have a large number of different failure modes, each of which should be considered separately during an analysis or design (Figure 3-1). Each of these failure modes depends on a different set of strength or stiffness constants and geometrical setups ranging from the stiffness of the connecting plates to the spacing of the connectors.

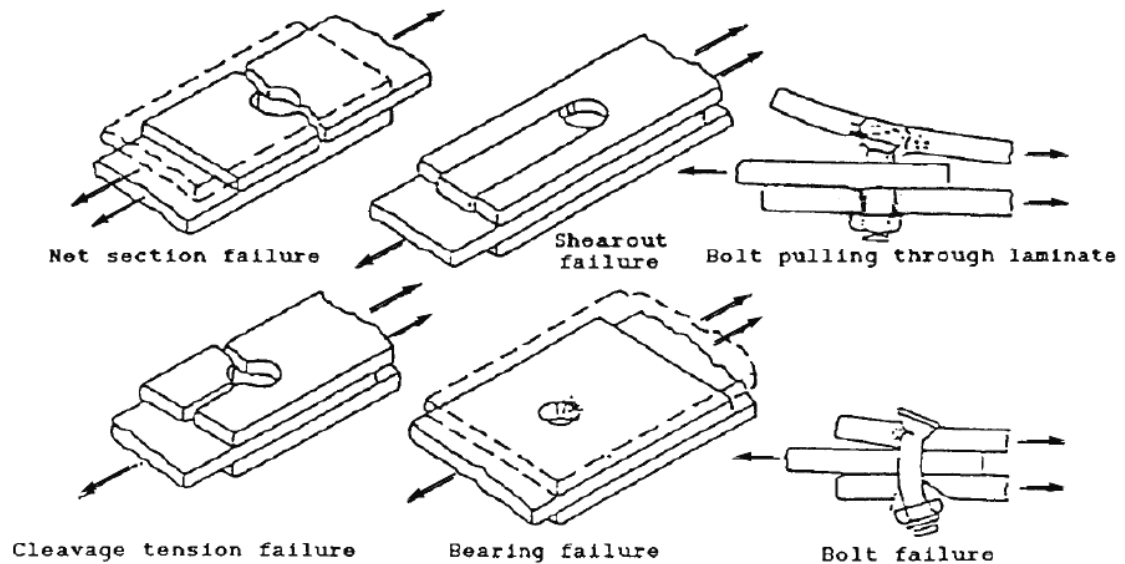


Figure 3-1: Various Mechanically Fastened Failure Mechanisms (Kim and Kedward)(2004)

First one must consider the configuration of the fasteners. A single row of connectors, for example, is analyzed somewhat differently than multiple rows (Figure 3-2). The applied load is to be transferred from one connecting plate to the fasteners

and finally into the other connecting plate through shear in the connectors (assuming a single lap joint). The amount of load taken by each fastener is a function of the spacing of the fasteners and is given by equation 3.1,

$$P = N_x S_y, \quad (3.1)$$

where P is the load taken by the single fastener, N_x is the distributed load acting on the element, and S_y is the connector spacing in the direction perpendicular to the load. In the case of a multi-row fastener configuration, the amount of load taken by each connector is dependent on the relative stiffness of the connecting members.

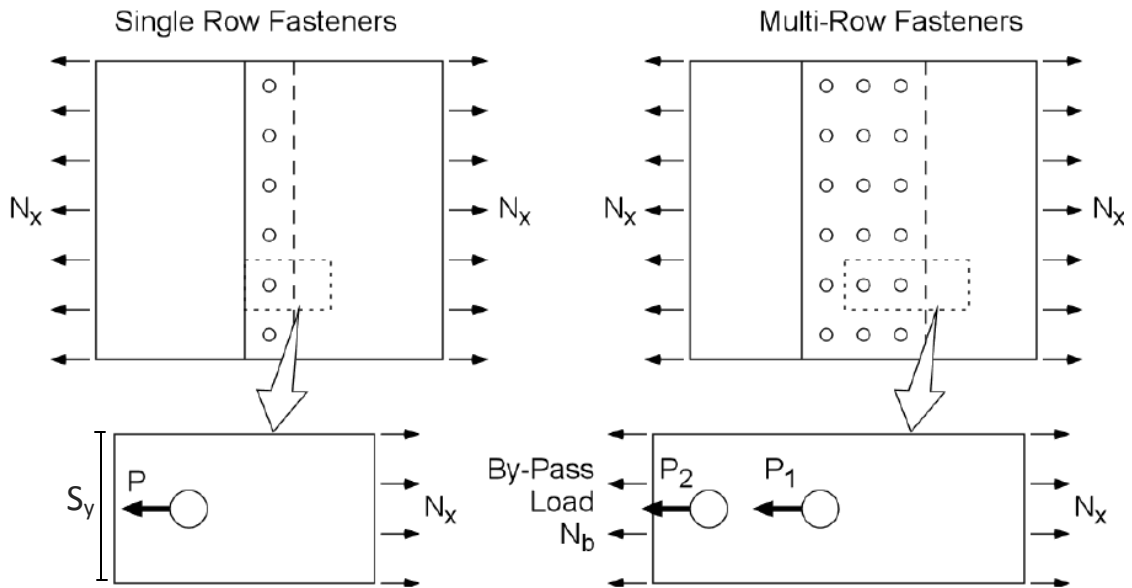


Figure 3-2: Illustration of a Single Row Versus Multi-Row Fastener Configuration

Niu (1999) gives a series of concise definitions of the various failure modes that exist in mechanically fastened connections (Figure 3-3 and Equations 3.2 a-c),

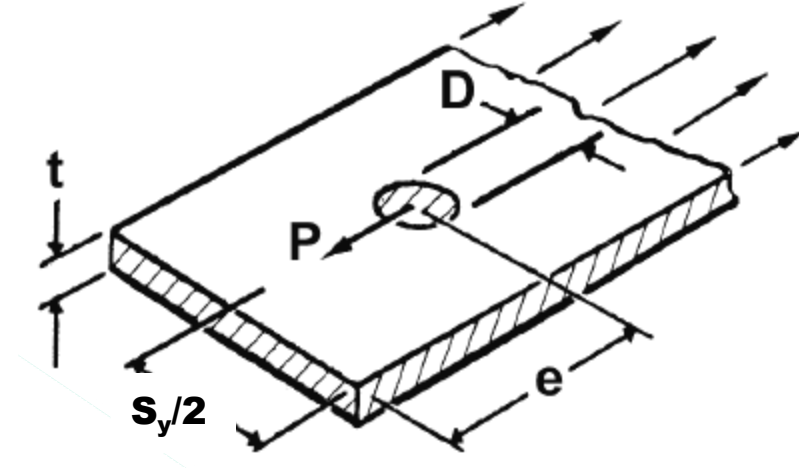


Figure 3-3: Representative Mechanically Fastened Element (Niu)(2009)

$$P^{br} = t d F^{bu} \quad , \quad P^s = 2 \left(\frac{e}{d} - 1 \right) t d F^{su} \quad , \quad P^t = 2 \left(\frac{S_y}{2d} - 1 \right) t d F^{tu} \quad , \quad (3.2 \text{ a-c})$$

where P^{br} , P^s , and P^t are the fastener loads in bearing, shear, and net tension respectively, d is the connector diameter, t is the plate thickness, e and S_y are the edge distance and spacing perpendicular to the load respectively. F^{bu} , F^{su} , and F^{tu} represent the ultimate strengths of the material in the bearing, shear, and net tension modes, respectively. Traditionally, analysis involves combining Equation 3.1 with those in 3.2 to find the maximum design strength of the joint as a whole.

3.2 Bonded Joint Development

Joint bonding is a process that occurs when a thin film of adhesive is placed between two pieces of material, eventually curing (hardening) and becoming an adhesive bond. For the purposes of finite element modeling, the connector pad being

developed will essentially act as a solid adhesive layer between two connected structures. For this reason, it is important to delve deeper into the analysis of bonded joints, so that an accurate representation of the behavior of the connectors can be achieved. This investigation will focus on the bonded joint modeling done by Volkersen (1938).

3.2.1 Advantages and Disadvantages of Bonded Joints

Like mechanically fastened connections, bonded joints also have pros and cons associated with the method and performance. Traditionally, adhesively bonded joints are used in less critical structural connections, though this is beginning to evolve with the more modern composite aircraft technology such as the Boeing 787 fuselage. Bonded joints are highly desirable due to their low weight due to the lack of metallic fasteners needed to connect the structural components, low cost of fabrication, and increase damage tolerance and strain to failure properties. In the modern world of aviation where weight savings are more important than ever, not only for efficiency but for cost, adhesives provide a worthy alternative to mechanically fastened joints. Additionally, bonded joints have a more continuous, flexible structure that has a high strength to weight ratio. Finally, the issue of corrosion found in mechanical joints is not an issue in bonded connections as dissimilar metals have a layer of adhesive acting as a buffer between them.

However, adhesively bonded joints, in comparison to mechanically fastened joints, are more difficult to fabricate in instances where the bond layer thickness is critical. Furthermore, in order to properly bond two surfaces, special surface

treatments are often required in order to guarantee adhesion. This means that adhesive bonding is a more thoughtful process that involves care in order to maintain the properties that have been designed for. Also, adhesives are often susceptible to environmental effects such as moisture absorption and temperature. Lastly, Adhesive joints are more difficult to inspect than mechanical joints, as visual inspection is often not an effective way to detect damage, as damage within the adhesive is not visible from the surface.

3.2.2 Analysis of Bonded Connections

The main method of load transfer in single lap adhesively bonded joints, as with all such connections, is shear transfer. In mechanically fastened connections, it is fairly easy to visualize that the load must pass from one part to the other through the discrete locations where the fasteners exist. With a more continuous connection region, like what is seen in a bonded joint, a shear lag model must be employed in order to depict the load transfer.

Shear lag states that a discontinuity within an axially loaded material or assembly must use shear in order to transfer the load. This is because at a discontinuity such as a crack or the end of an adherend, there exists a traction free boundary condition, inhibiting the section of material from opposing the load axially. To balance the axial load, the section must transfer the load via shear to the neighboring regions of the assembly or material (Figure 3-4 and 3-5).

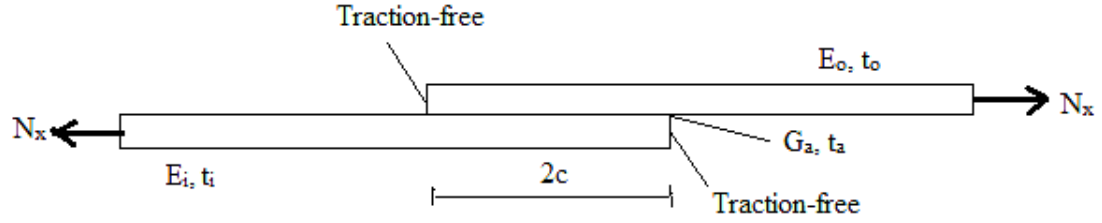


Figure 3-4: Representative Single Lap Bonded Joint

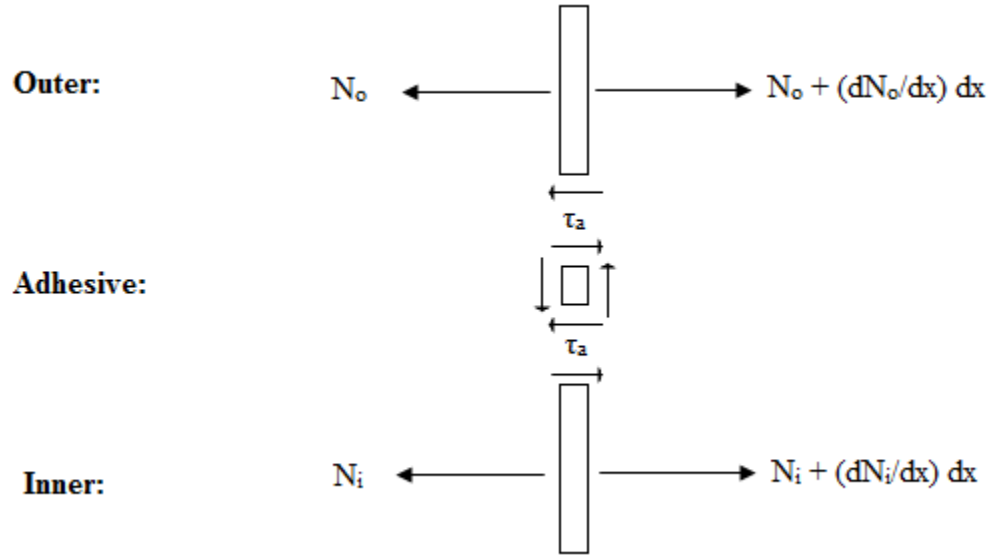


Figure 3-5: Load Transfer Via Shear in Bonded Joints

where E_o and E_i are the moduli of the outer and inner adherends, t_o and t_i are their respective thicknesses, G_a is the shear modulus of the adhesive which has thickness t_a . N_o and N_i are the loads in the outer and inner adherends respectively and τ_a is the shear that is transferring the axial load.

It is easily shown by balancing forces that:

$$N_i = N_x - N_o \quad (3.3)$$

$$\tau_a = \frac{dN_o}{dx} = - \frac{dN_i}{dx} \quad (3.4)$$

Using Equations 3.3 and 3.4, we can rewrite the derivative of the adhesive shear stress as:

$$\begin{aligned} \frac{d^2 N_o}{dx^2} - \lambda^2 N_o + C_o &= 0 \\ \lambda^2 &= \frac{G_a}{t_a} \left(\frac{1}{E_o t_o} + \frac{1}{E_i t_i} \right) \\ C_o &= \frac{G_a N_x}{t_a E_i t_i} \end{aligned} \quad (3.5 \text{ a-c})$$

The differential equation shown above has the general solution:

$$N_o(x) = A_o \cosh(\lambda x) + B_o \sinh(\lambda x) + \frac{C_o}{\lambda^2}, \quad (3.6)$$

where A_o and B_o are constants solved for with boundary conditions at the ends of the joints (i.e. no load at the free surface).

From equation 3.4, the above solution can be differentiated, yielding the adhesive shear stress profile:

$$\tau(x) = \lambda \left[\frac{N_x \cosh(\lambda x)}{2 \sinh(\lambda c)} + \frac{\sinh(\lambda x) \left(\frac{N_x}{2} - \frac{C_o}{\lambda^2} \right)}{\cosh(\lambda c)} \right], \quad (3.7)$$

where c is equal to half of the bond length. Further, from equation 3.6, the stress and strain in the connected plate is equal to:

$$\sigma_o = \frac{N_o}{t_o}, \quad \varepsilon_o = \frac{\sigma_o}{E_o}. \quad (3.8 \text{ a-b})$$

The strain can then be integrated along the overlap length to calculate the axial displacement of the plate. It should be noted that once the relationship from equation 3.3 is applied, analogous relationships to those in 3.8 are also valid.

The disadvantage to the Volkersen (1938) model for bonded joints is that the shear stress in the adhesive does not obey the traction free boundary conditions at the free edges of the adhesive. This over predicts the shear stresses at the free edges of the adhesive greatly, though the macro level solution still yields accurate results in most cases. Other models that allow for transverse shear displacement of the adherends or for the adhesive to behave plastically (Hart-Smith (1973)) have been developed to attempt to alleviate some of the inaccuracy in this model. While they are in fact a more accurate and complete representation of the behavior of the bond, the displacement and force results only experience minimal change for many common cases, such as the thin, single lap joints with which this project is concerned.

3.3 Finite Element Modeling Approach

The principles and theories outlined above are rooted into the finite element method that is employed during the computational modeling stage of the project. Two finite element software packages were employed for the modeling phase of the project, FRANC 2D/L, developed at Kansas State University and Cornell University that specializes in damage tolerance and layered plate analysis (i.e. joints), and FEMAP/NX NASTRAN, a finite element modeling and analysis program. This development will focus on four major types of elements that will all act to simulate the jointed portions of our nested angle specimens: a discrete spring element (CBUSH), a Timoshenko beam element, a solid element assembly have the same cross sectional area as the fastener, and a solid adhesive connector pad covering varying amounts of the jointed region (see Figure 3-6).

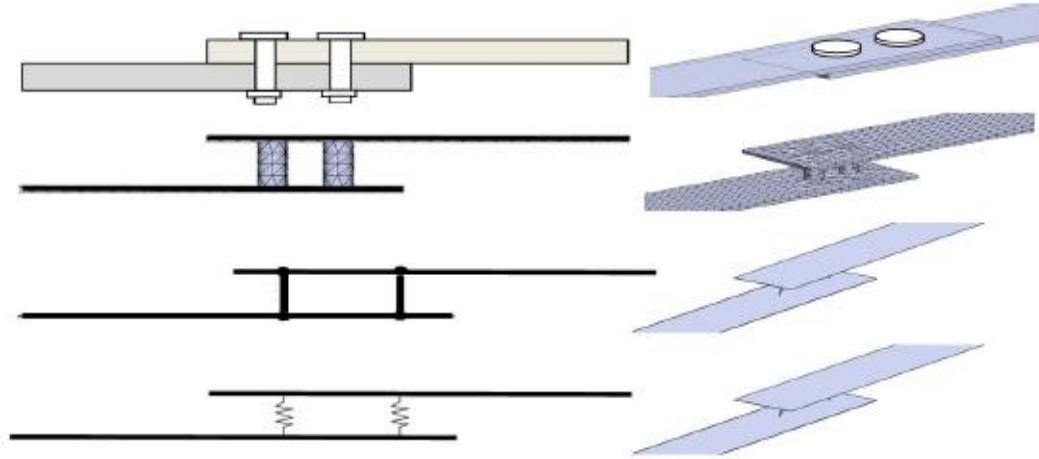


Figure 3-6: Various Modeling Techniques Varying from Entire Connector, to Spring Elements (Kosmatka, Brewer, Pun, and Hunt (2012))

3.3.1 Effective Material Property Approach

The basis of the finite element development for the current research is the stored energy in the system. Urugal and Fenster (1995) demonstrate that the general deformation for a beam can be decomposed with linear superposition into the sum of the term associated with the bending energy and the term associated with the shear energy. Cope and Lacy (2000) expand upon this idea, making it applicable to joints such as those in the current research. The assumption is made that the material model is purely linear elastic and that the fastener displacement is due to a similar combination of shear and bending terms. According to Cope and Lacy (2000), under unidirectional in plane loading, the fastener displacement can be written as a combination of shear and bending terms:

$$\Delta = \frac{Fh}{G_f A} + K \frac{Fh^3}{E_f I}, \quad (3.9)$$

where F is the load, h is the joint eccentricity defined above, E_f and G_f are the fastener axial and shear stiffnesses, respectively, and A and I are the fastener cross sectional area and moment of inertia, respectively. The constant K is dependent on the amount of bending relative to shearing that exists in the connector. Typical values are summarized in Table 3-1.

Table 3-1: K Factors for Solid Element Fastener Deflections

K	Description
0	For very stiff connecting plates relative to the fastener, the fastener undergoes all shearing with no bending
1/12	For double lap (or larger) connections (three or more sheets), the pin undergoes shearing and fixed-fixed bending
1/3	For one plate much thicker than the other, the fastener undergoes shearing and cantilever-like bending

Studies done by Swift (1984) demonstrate that the fastener displacement accounts for over 75% of the total displacement of the specimen. From these studies, the following empirical formula was developed for the fastener displacement:

$$\Delta = \frac{F}{Ed} \left(B + C \left(\frac{d}{t_1} + \frac{d}{t_2} \right) \right), \quad (3.10)$$

where E is the Young's modulus of the connected sheet, d is the diameter of the fastener, t_1 and t_2 are the thicknesses of the two connected sheets, and B and C are empirical constants determined by the material from which the fasteners are made. Table 3-2 summarizes these constants.

Table 3-2: Empirically Derived Fastener Constants (Cope and Lacy)(2000), (Swift)(1984)

Material	B	C
Aluminum	5.0	0.8
Steel	1.667	0.86
Titanium	4.0	0.82

Employing these constants, Equations 3.9 and 3.10 for the fastener displacement can be set equal and the effective E and G for the fasteners can be solved for. It should be noted that an isotropic fastener material is assumed for this step in the derivation. Cope and Lacy (2000) expanded on the principles developed by Swift (1984) to develop the following expressions for E_{eff} and G_{eff} :

$$E_{eff} = E \frac{hd}{A} \frac{\left(2(1+\nu) + K \frac{Ah^2}{I}\right)}{\left(B+C\left(\frac{d}{t_1} + \frac{d}{t_2}\right)\right)}, \quad G_{eff} = E \frac{hd}{A} \frac{\left(1 + K \frac{Ah^2}{2(1+\nu)I}\right)}{\left(B+C\left(\frac{d}{t_1} + \frac{d}{t_2}\right)\right)}, \quad (3.11 \text{ a,b})$$

where h is the joint eccentricity, A is the fastener cross sectional area, and ν is the Poisson's Ratio of the connected sheet. Using these new properties, the fastener can effectively be modeled using an anisotropic material property, where the standard connector stiffnesses E_f and G_f would control the axial and torsional stiffnesses of the fastener, and E_{eff} and G_{eff} would control the modeled fasteners transverse bending and shear, respectively (Kosmatka, Brewer, Pun, and Hunt (2012), Swift (1984)). It should be noted that in each of the following modeling approaches, the element density is scaled to reflect the added total mass of the fasteners within the connection region. The scaling is done based on the area over which each approach will spread

the discrete fasteners out (i.e. a single node for beams and springs, and larger areas for the connector pad).

3.3.2 Solid Element Modeling Approach

For this method, the fastener is modeled using solid elements that connect the two opposing connected sections. The sheets are modeled using plate elements and modeled at the midplane of the physical sheet. With this convention, the eight node brick elements span the gap equal to the joint eccentricity of $h = \frac{(t_1 + t_2)}{2}$, where t_1 and t_2 are the thicknesses of the two connected sheets (Figure 3-7) and have the effective properties from Equation 3.11. In order to combat the fact that the plate elements that compose the angles have additional rotational degrees of freedom compared to the solids, the rotations were constrained, allowing strictly the displacement in the axial direction. If the angles were also composed of solid elements, this mismatch would not exist.

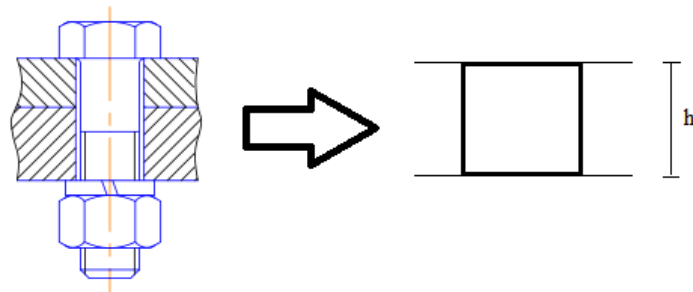


Figure 3-7: Solid Element Representation of a Fastener (http://en.wikipedia.org/wiki/Bolted_joint)

3.3.3 Beam Element Modeling Approach

Rather than modeling each fastener as a group of solid elements, the cylindrical section of the fastener can be modeled using a beam element in modern finite element codes, connecting the two necessary nodes with 6 degrees of freedom per node. Like the solid elements, the beam would span a distance h , equal to the distance between the midplanes of the plates (Figure 3-8). In most practical cases, the length of this beam element will be very small in comparison to the connector diameter. For this reason, the Timoshenko beam theory must be employed in order to account for shear deformation of the pin. Similar to the solid element approach, the mismatch in degrees of freedom between the plate elements and beams must be accounted for, by applying an extra constraint to the beam elements to account for the out of plane bending degree of freedom that doesn't exist in the plate. The E_{eff} and G_{eff} in Equation 3.11 are used for the material properties to model the shear transfer behavior, along with a shear correction factor associated with circular sections ($k = 9/10$) and the section properties of the cylindrical pin model (A, I) (Cowper (1966)).

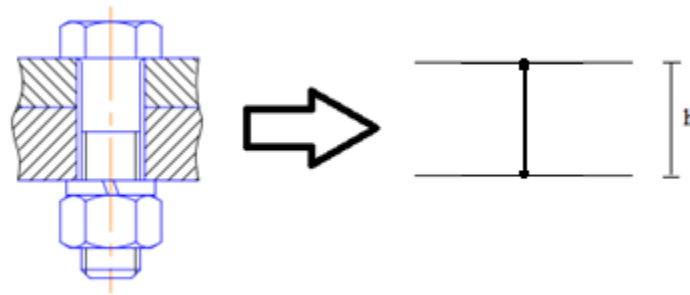


Figure 3-8: Beam Element Representation of a Fastener (http://en.wikipedia.org/wiki/Bolted_joint)

3.3.4 Spring Element Modeling Approach

Rather than using a beam element, a spring element can be used to model the fastener using the same method of connection (two nodes with six degrees of freedom) as the beam. Such a representation would be similar to Figure 3-8 above. Both the effective E and G from Equation 3.11, and the standard fastener E_f and G_f are used in the strength of materials derivation of the extensional (k_e), shearing (k_s), bending (k_b), and torsional (k_t) stiffnesses of the spring element (Equation 3.12 a-d):

$$k_e = \frac{E_f A}{h}, \quad k_b = \frac{4E_{eff} I}{h}, \quad k_s = Ed \frac{1}{\left(B + C \left(\frac{d}{t_1} + \frac{d}{t_2} \right) \right)}, \quad k_t = \frac{G_f J}{h}. \quad (3.12 \text{ a-d})$$

3.3.5 Multiple Fastener Modeling Approach via Connector Pad

In any given flight critical aircraft structure, there exists hundreds of fasteners arranged with varying densities and configurations. For this reason, it is extremely impractical to individually model each fastener using any of the aforementioned approaches during the initial design/analysis phase. Especially true during the preliminary design phase, where changes and design evolution are commonplace, it is desirable to model fasteners and mechanically fastened connections in a global fashion, while still maintaining the accuracy of the analysis model. To do so, a region of connectors can be simplified into one connector pad that is defined by a thin layer of solid elements, much like an adhesive would be. Like the approaches that came before, the connector pad would occupy the space between the midplanes of the opposing sheets that are made of plate elements. Figure 3-9 shows the conversion of a group of uniformly distributed fasteners into an equivalent connector pad. Increasing

the number of fasteners within a connection region, or changing the connector diameter or material properties are in turn reflected in the shear and bending properties of the pad associated with that region of fasteners.

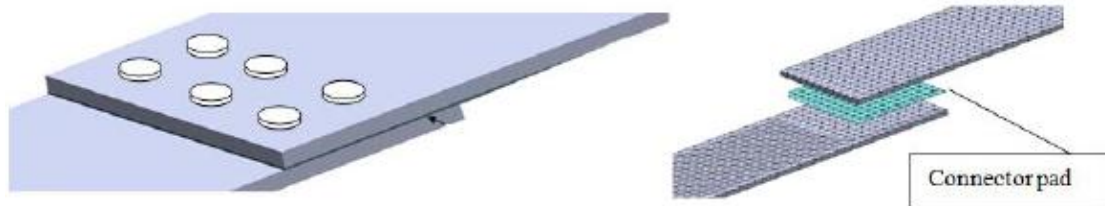


Figure 3-9: Multiple Fasteners Converted into an Equivalent Pad (Kosmatka, Brewer, Pun, and Hunt) (2012)

In the case that the fasteners are not as neatly distributed as those in Figure 3-9, multiple connector pads can be defined to reflect the properties of the fasteners within a slice of the connection region. For example, in sub regions with a higher concentration of fasteners, the resulting connector pad properties would be higher than those outside of the immediate area (Figure 3-10).

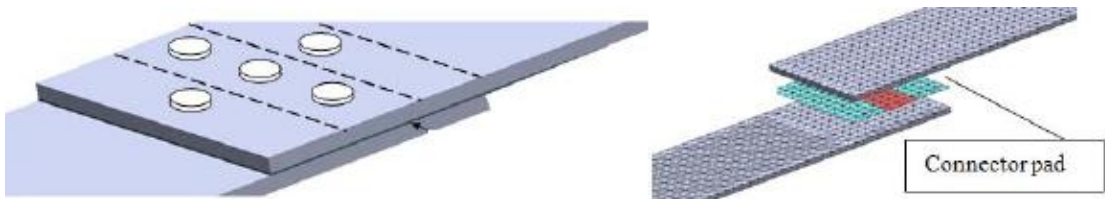


Figure 3-10: Effect of Fastener Density on Connector Pad Definition (Kosmatka, Brewer, Pun, and Hunt) (2012)

Using this approach, the connector pad's axial and shear moduli (E_{pad} and G_{pad}) would be defined by knocking down the stiffness of the fasteners with a ratio of the areas over which the modulus will be active (i.e. The connector pad will cover a larger area than the fastener, and therefore will have a lower modulus). See Equation 3.13:

$$E_{\text{pad}} = \frac{1}{A_{\text{pad}}} \sum_{i=1}^n E_{\text{eff}}^i A^i, \quad G_{\text{pad}} = \frac{1}{A_{\text{pad}}} \sum_{i=1}^n G_{\text{eff}}^i A^i, \quad (3.13 \text{ a,b})$$

where n is the number of fasteners in the overlap region.

However, in such a formulation, the load is potentially able to transfer continuously throughout the entire overlap length as opposed to discrete locations such as the fasteners (see Figure 3-11). If the overlap regions are defined such that they are long, a different formulation may be needed. In other words, the properties generated in this formulation apply to the areas where fasteners would actually exist, with a higher density of fasteners behaving better than a lower density, as it would be more similar to the continuous load transfer mechanism.

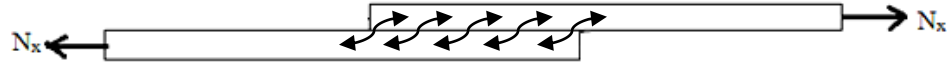


Figure 3-11: Unrealistic Continuous Load Transfer

CHAPTER 4: FINITE ELEMENT ANALYSIS AND RESULTS

In this chapter, a finite element study was performed to understand the tradeoffs between connection type and spacing with the resulting displacement.

4.1 Finite Element Methodology and Setup

The assembly that was modeled and tested for this research was called a nested angle assembly. Each of the specimens was comprised of two aluminum angle brackets, one smaller than the other, that were nested together and fastened with one of several types of fasteners. Figures 4-1 and 4-2 detail the configuration.

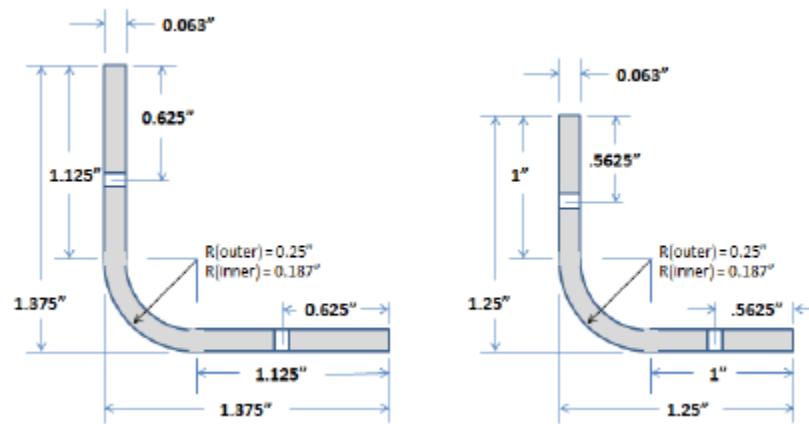


Figure 4-1: Large and Small Angle Cross Sections (Kosmatka, Brewer, Pun, and Hunt) (2012)

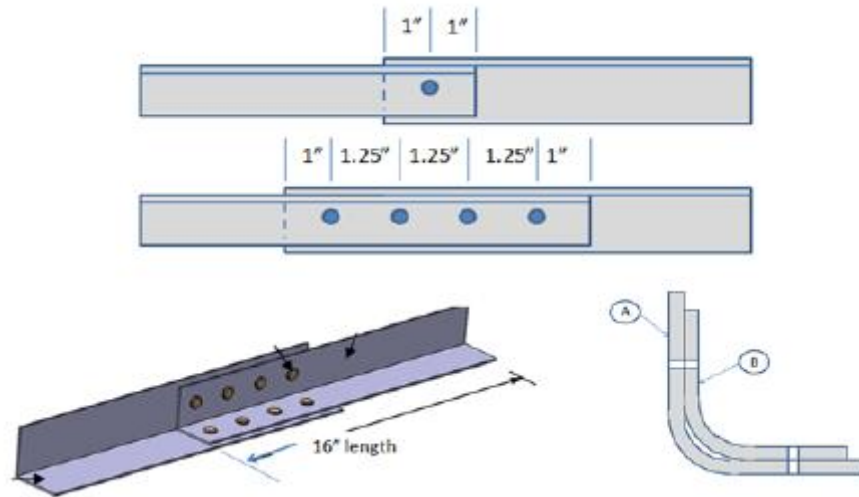


Figure 4-2: Two and Eight Fastener Test Specimens in Nested Configuration (Kosmatka, Brewer, Pun, and Hunt) (2012)

4.1.1 The Specimens

The geometry was modeled from direct measurements of the specimens which had a range of two to eight fasteners. Each of the individual angles had a uniform thickness of $0.0625 \left(\frac{1}{16}\right)$ inches and was 16 inches long. These individual angles were then fastened into the nested configuration shown in chapter one. Due to varying numbers of fasteners maintaining the spacing requirements from chapter two, the overlap length, and therefore, the overall length of the physical specimen varied from 30.25 inches to 26.25 inches. Overall, three different sets of mechanically fastened specimens were fabricated; four Hi-Lok specimens (two, four, six, and eight fasteners), four specimens with stainless steel bolts, and four specimens with aluminum POP rivets. Additionally, a set of four adhesively bonded specimens was fabricated in an attempt to replicate the connector pad models and show different behavior between fastened and adhesive models. For each of the specimens that was

made, the longitudinal edge distance of the fasteners is one inch and the spacing between fasteners is 1.25 inches. Each of the fasteners, regardless of the type, has a $\frac{3}{16}$ inch pin diameter. It should be noted that the ratio of the aforementioned spacing and diameters is equal to 6.67, which is much larger than the industry standard of 4. The Hi-Lok fasteners are composed of two separate parts: the pin and the collar (Figure 4-3). The pins used for the nested angle specimens are HL-18 solid steel pins, and the collars are HL-70 2024 aluminum. The bolted specimens use solid 18-8 stainless steel hex head bolts and nuts. The POP rivets are made of 5056 aluminum and are not completely solid in the shear area cross section like the other two fastener types. This will be discussed in more detail later in this chapter. It should also be mentioned that one specimen was tested with 18-8 stainless steel POP rivets, to investigate differences in behavior. The adhesively bonded specimens used the same aluminum angles and were bonded to have the same range of overlap lengths as the other specimens. The adhesive that was used was Hysol EA 9394 in conjunction with 0.005 inch diameter 3M Scotchlite Glass Beads to control the bondline thickness.

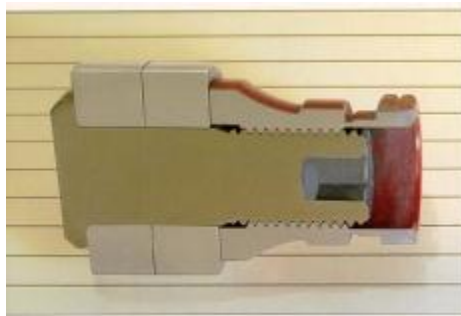


Figure 4-3: Hi-Lok Fastener Section View (Hi-Shear Corporation)

4.1.2 Modeling the Specimens in FEMAP

The finite element models were made in the FEMAP preprocessor and analyzed with the NX NASTRAN solver. While the angles are each 16 inches long, a custom set of grips was designed to hold the assembled nested angles in the testing machine. The connection on each side of the nested angles takes four inches. For the purposes of computer modeling, this reduces the length of each angle to 12 inches, as there is a fixed boundary condition for the remainder of the distance.

The angles were each modeled using four node bilinear plate elements. The mesh size was set to 0.1 inches in both directions in order to be able to represent the radius of the fastener with reasonable accuracy using solids while still maintaining minimal computational effort. The solid elements that were used to represent the fasteners were trilinear brick elements. Since a majority of the displacement would come from shear rather than bending, the beam elements, as mentioned in chapter three, follow the Timoshenko model in order to allow for shear deformation. Finally, CBUSH elements from FEMAP's spring/damper library are used for the spring elements. These elements are further outlined in chapter three as well.

The modeling process for each of the overlap lengths (two, four, six, and eight fasteners) began by modeling a full connector pad. In a similar fashion to Figure 3-8, two connection zones were defined. One zone was defined between the midpoints of the outermost fasteners and was assumed to include all of the fasteners, including the halves that technically lay outside (see Figure 4-9). This zone took on the properties defined by Equation 3.11, and therefore represented knocked down versions of the

effective E and G defined by Swift (1984). The second zone was defined as the region outside of the fasteners to the edge of the overlap region. For this zone, since no load would be transferred in a mechanically fastened specimen, the E and G are set very low as an approximation. For preliminary design, this method can be modified in order to restrict the fasteners to smaller strips within the overlap, with the properties only changing based on the change in area. As a short aside, the physical adhesive specimens were modeled with the aforementioned model, with all overlap regions defined with the same properties for Hysol EA9394 ($E = 615$ ksi, $G = 212$ ksi). This modification leads to the second main model, where the fastener areas are modeled by solids. For this mesh size, three solid elements were chosen to represent the fastener, yielding an area that is accurate within 10%. The group of solid elements is placed at the location where the fastener is assumed to be. This is the inherent benefit in the connector pad and thin modification thereof, in which the fasteners are only required to exist in the connector pad area, and not at an exact location, making the preliminary design process easier. Using the three element solid model as a guide, the spring and beam models can be easily created using the existing nodes (and choosing the most central for the fastener) and simply deleting the solids. The beam element uses the effective E and G from Equation 3.11 and the actual cross sectional area of the fasteners. The spring elements use the properties defined in Equation 3.10 for the six degrees of freedom.

The aforementioned modeling process needs to be modified slightly to represent the POP riveted specimens. The reason for this modification is that the cross

section of the rivets is not solid like those of the Hi-Lok rivets and the 18-8 stainless steel bolts. This is due to the nature of the blind riveting process, which pulls a mandrel head into a hollow section, widening the flanges and securing the fastener, but breaking the mandrel and leaving behind a hollow cross section in the shear plane (see Figure 4-4). Therefore, in order to model these fasteners, a solid cross section was assumed with the same cross sectional area as the hollow POP rivet. The wall thickness of the pop rivets 0.00225 inches, yielding a cross sectional area of 0.00068035 in^2 . Finding the equivalent solid cross section gives a radius of 0.014716 inches. Using this method, all of the development in chapter three and the preceding paragraphs is applicable.

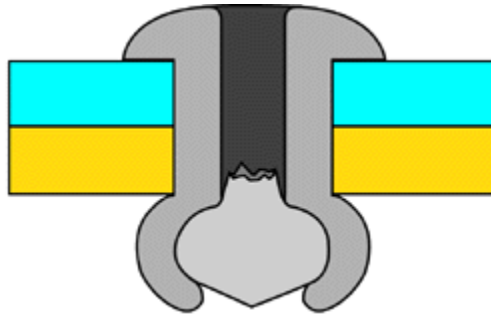


Figure 4-4: Blind Rivet with Hollow Cross Section in the Shear Plane (Marson and Gesipa)(2012)

Loads and boundary conditions were applied through the centroid of the joined section. This was done by placing rigid spiders between the free and fixed end edges of the angles and a node at the centroid. In this way, the bending of the specimens due to eccentric loading can be minimized. Additionally, sliders were placed along the longitudinal edges of the angles in order to prevent out of plane bending behavior.

Figures 4-5 through 4-8 show the full meshes of the two, four, six, and eight fastener models, each using CBUSH elements in order to show where the fasteners exist within the overlap region. Figures 4-9 through 4-11 all are close up side views of the connection region of the eight fastener specimen, each using a different connection type in order to demonstrate how each fastener type goes into the connection region. Bear in mind that models with each connection type were made for each overlap length, and not just this limited collection.

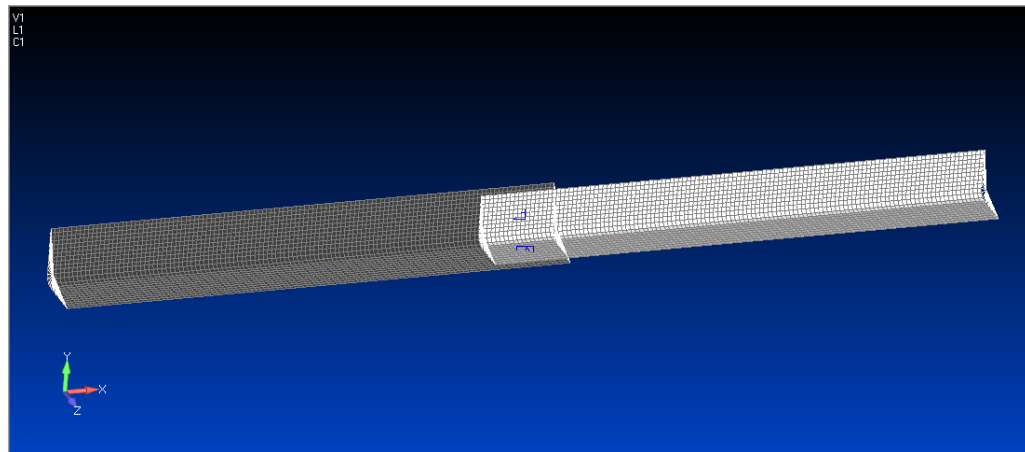


Figure 4-5: Two Fastener Finite Element Mesh with Spring Elements

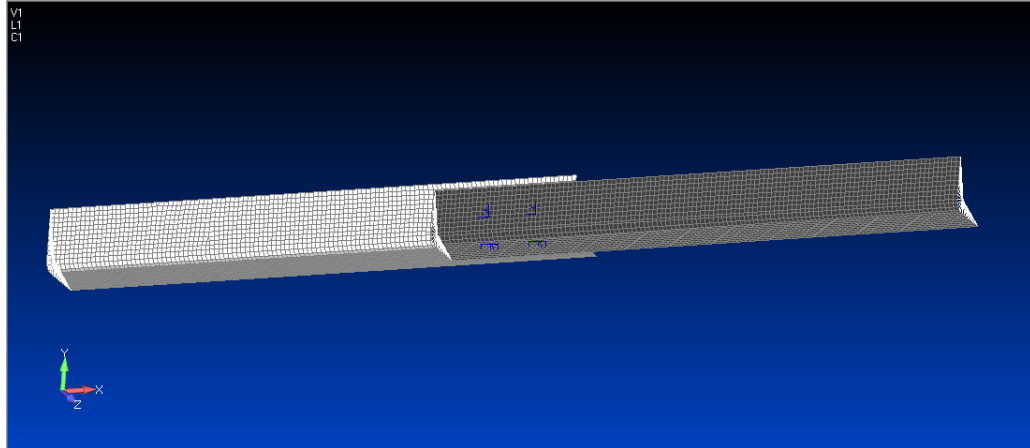


Figure 4-6: Four Fastener Finite Element Mesh with Spring Elements

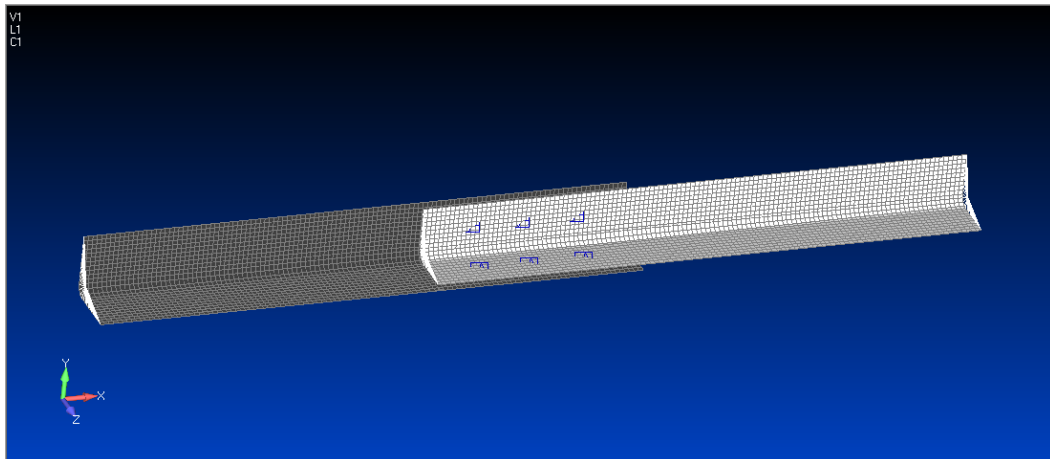


Figure 4-7: Six Fastener Finite Element Mesh with Spring Elements

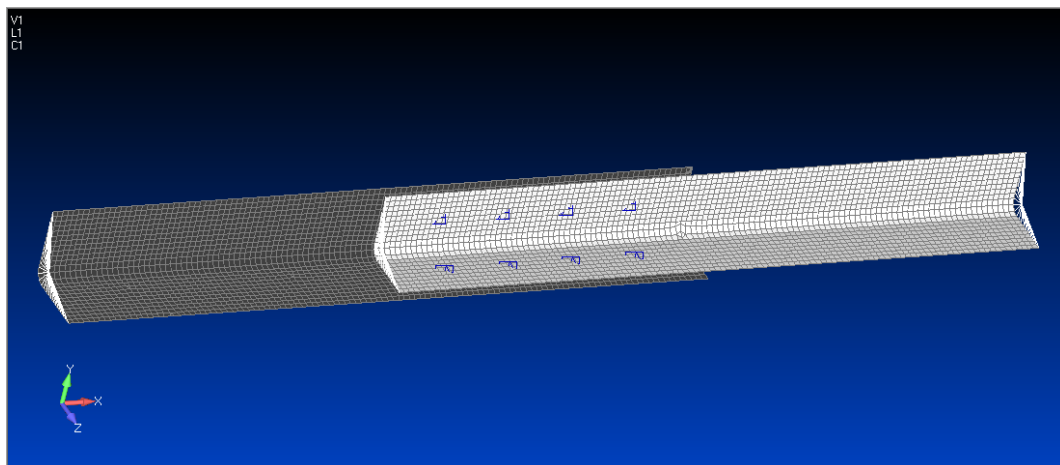


Figure 4-8: Eight Fastener Finite Element Mesh with Spring Elements

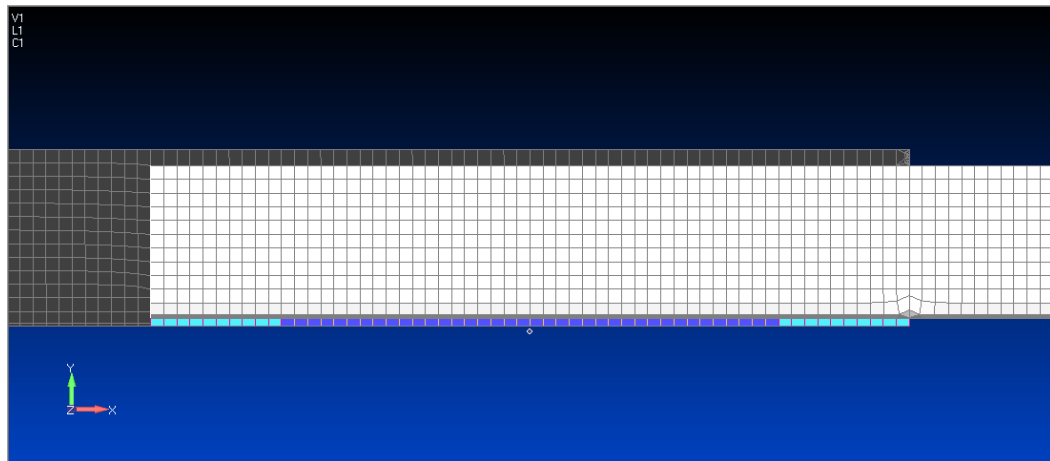


Figure 4-9: Eight Fastener Connection Region with Two Connection Zones

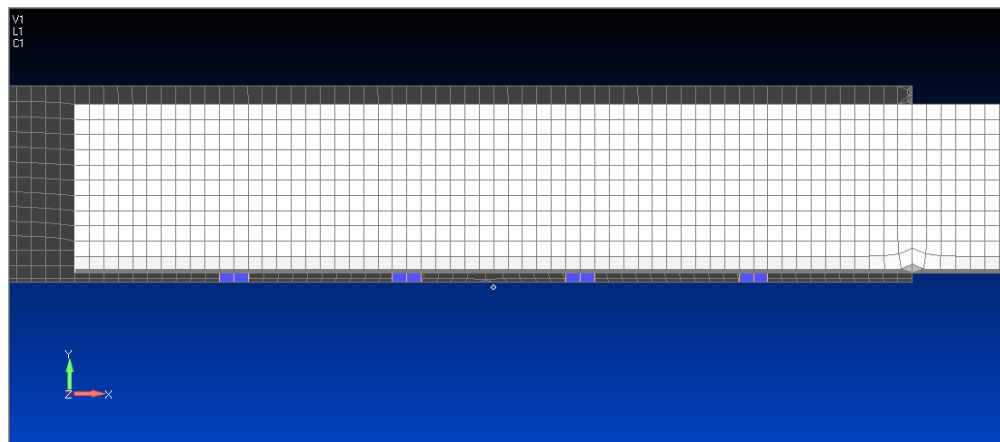


Figure 4-10: Eight Fastener Connection Region with Solid Element Fastener Approximations

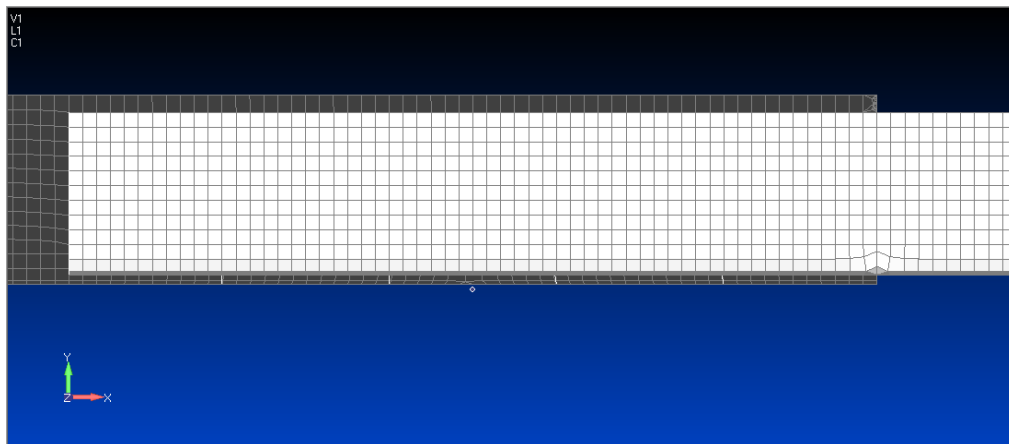


Figure 4-11: Eight Fastener Connection Region with Timoshenko Beam Elements

4.1.3 Modeling the Specimens in FRANC2D/L

As mentioned in chapter three, Cope and Lacy (2000) show the development of an explicit rivet element to be used within FRANC 2D/L. In order to compare this element with the fastener model developed by the testing done by Swift (1984), several analysis models were made in FRANC 2D/L as well. The rivet element is based on the development done by Swift (1984) and is contained within Equation 3.9 using a value of $K = 1/3$.

The first challenge arises from the fact that by its nature, FRANC 2D/L is exclusively a 2D modeling software. Therefore, a model was made in FRANC 2D/L and FEMAP to see if a flattened nested angle would compare favorably to its more realistically shaped counterparts in FEMAP. Figure 4-12 shows flattened FRANC 2D/L mesh that uses the same materials, geometry, and fastener density/configuration as the actual eight fastener specimens (as well as the associated FEMAP models). Therefore the only true difference between the FRANC 2D/L and FEMAP models is the bend in the angles, or lack thereof.

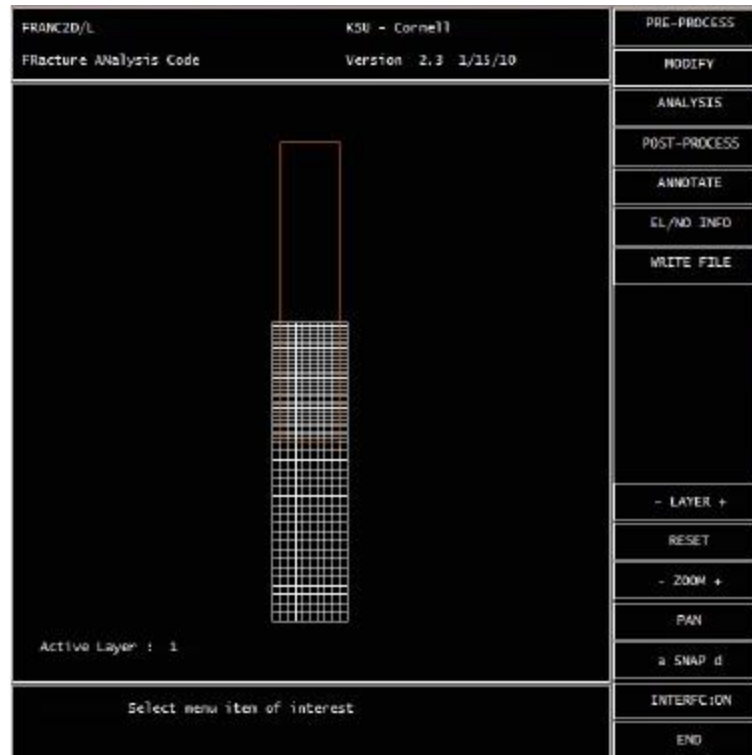


Figure 4-12: Flattened FRANC 2D/L Nested Angle Model

Using the Volkersen (1938) model described in chapter three, an equivalent shear modulus can be found. From the relationship that $\tau = G\gamma$, where τ is the shear stress calculated from the development in chapter three, and γ is equal to the relative displacement of the joint divided by an artificially chosen thickness of 0.001 inches (this assumed thickness eventually was changed to be equal to the distance between the midplanes of the plate elements defining the nested angles). The relative displacement of the joint could be measured from the FRANC 2D/L model, allowing for the simple calculation of an equivalent G of 11095 psi for this case. An adhesive layer with these properties was created in both FRANC 2D/L and FEMAP and both analyses were run to check for equivalence (see Figures 4-13 and 4-14).

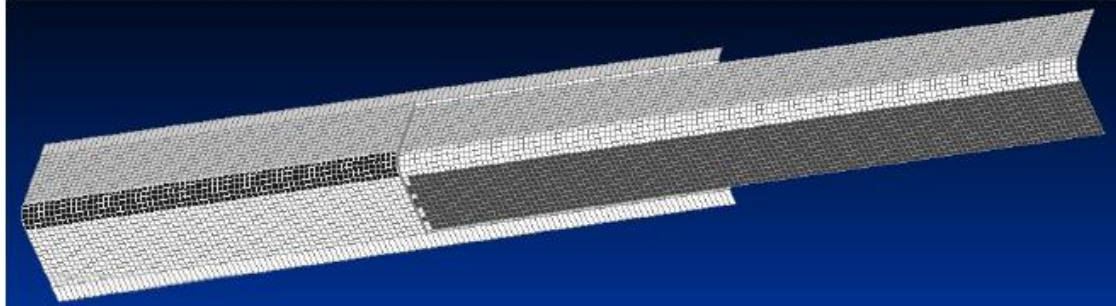


Figure 4-13: FEMAP Equivalent G Verification Mesh

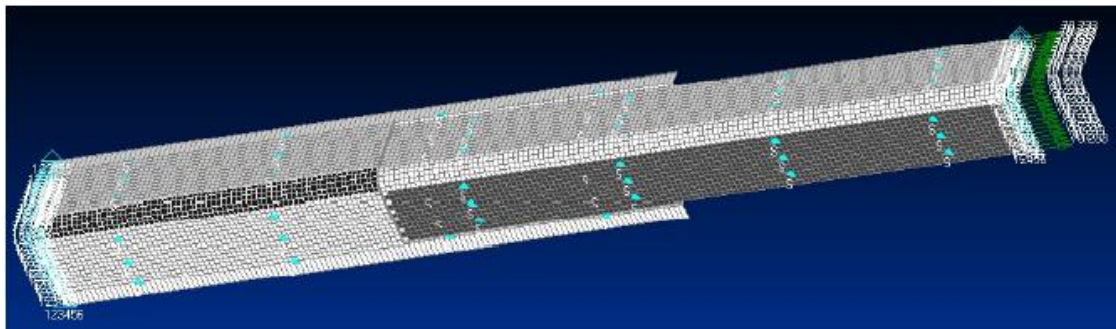


Figure 4-14: FEMAP Equivalent G Model Boundary Conditions and Loads

Figures 4-15 and 4-16 show the results of these finite element analyses. It can be seen that the total axial displacement matches with 0.5% at 0.019 inches. It has not been demonstrated whether or not the angle bend has an effect on the stresses in the nested angle, but for displacement analyses, FRANC 2D/L is shown to be a valid analysis method.

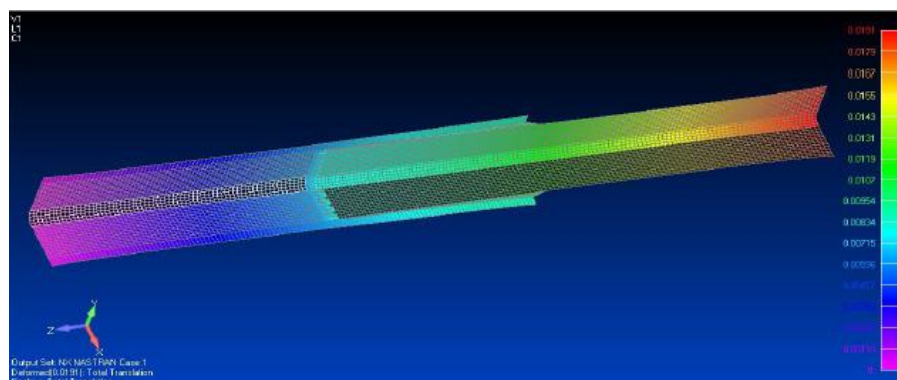


Figure 4-15: FEMAP Equivalent G Results

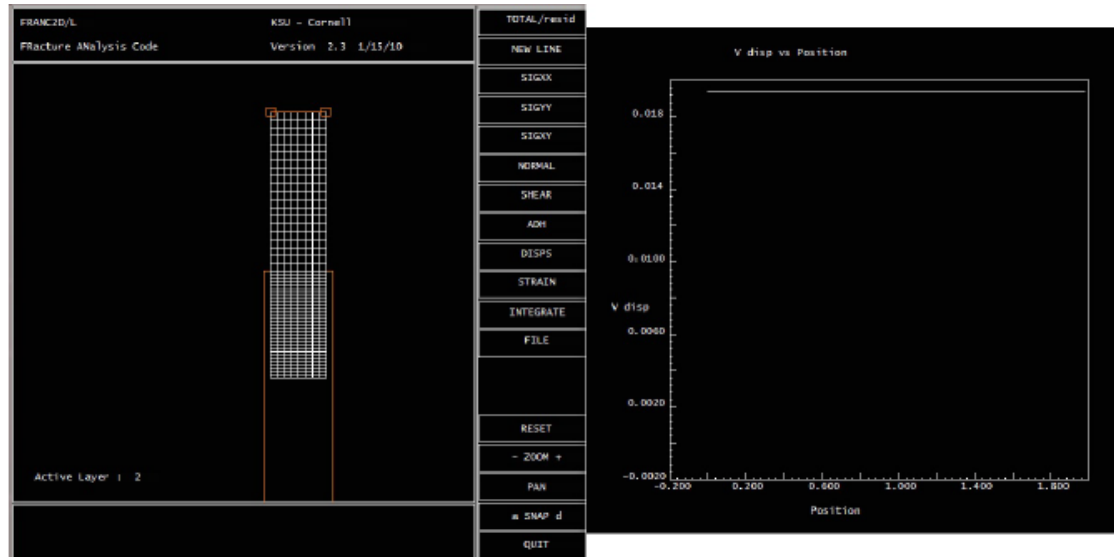


Figure 4-16: FRANC 2D/L Equivalent G Results

4.1.4 A Representative FRANC 2D/L Model

As discussed, FRANC 2D/L is a valid method for analyzing these nested angle specimens for overall joint displacements. While FEMAP/NX NASTRAN was the primary analysis method, models were made in FRANC 2D/L as well in order to test the previous assertion with real specimens and test data. The model was made using 88 elements in the width direction to accurately capture the variation in displacement across the section, 23 elements in the longitudinal direction in the overlap area, and only six elements in the longitudinal direction outside of the connection region in order to fit with the memory allocation limits of FRANC 2D/L. Figure 4-17 shows the eight connector model in FRANC 2D/L.

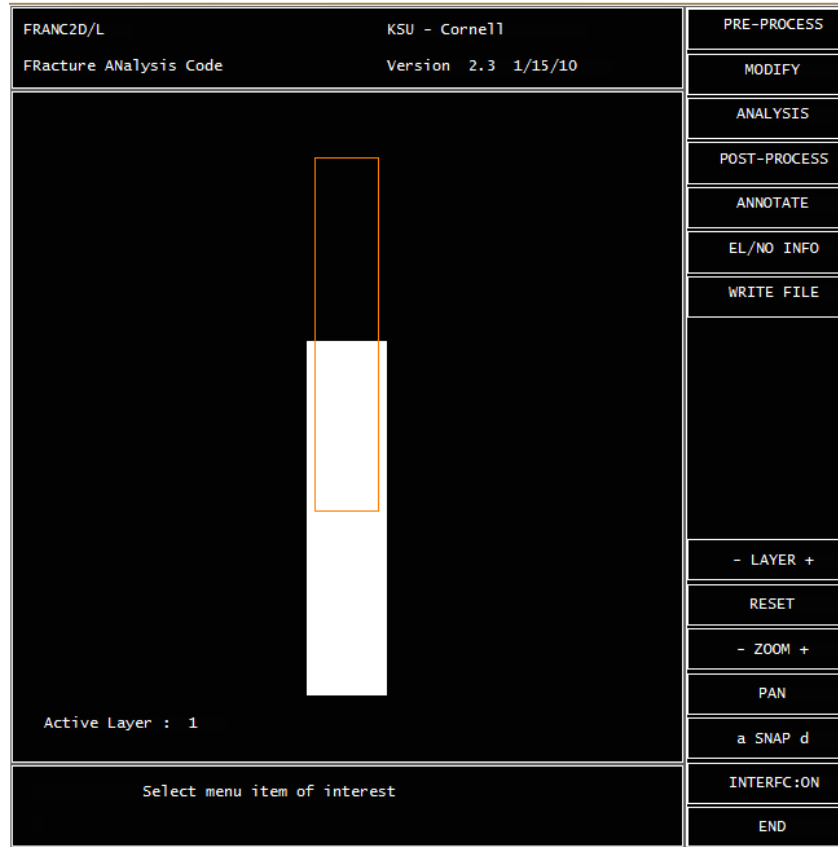


Figure 4-17: FRANC 2D/L Model of Eight Connector Specimen

Upon applying the same axial load and boundary conditions, the analysis was completed and displacements measured at each fastener.

4.2 Finite Element Results

As stated in the previous section, there were 48 models created in FEMAP that were to be compared to the test results. 24 of these models represent the Hi-Lok and stainless steel fastener specimens and the other 24 represent the POP riveted specimens. Table 4-1 contains the values for E_{eff} , G_{eff} , k_e , k_b , k_s , and k_t for each of the fastener types that were modeled using Equations 3.11 and 3.12. Note that the POP rivet values appear much stiffer than the Hi-Lok. This is due to the fact that the rivet's

cross sectional area and inertia are so small in comparison to the other two fastener types, that, by the design of Equations 3.11 and 3.12, some of the values get very large. For the calculations of the E_{eff} and G_{eff} , a value of $K = 1/12$ was used. This is consistent with most common applications and joint behaviors Swift (1984). Table 4-2 contains the results for the total displacements of the Hi-Lok and stainless steel fastener specimens. Table 4-3 contains the results for the POP rivet models. Figures 4-18 and 4-19 show the results from these tables in bar graph form as a visual comparison between modeling techniques.

Table 4-1: Effective Fastener Properties for FEM

Property\Fastener Type	Hi-Lok	18-8 Stainless Steel	POP Rivet
E_{eff} (psi)	1,833,000	1,833,000	152,437,624
G_{eff} (psi)	689,100	689,100	57,306,211.86
k_e (lb/in)	12,811,504	12,811,504	108,856
k_b (lb/in)	7117.319	7117.319	54,809.17
k_s (lb/in)	37,121,813	37,121,813	191,326.53
k_t (lb-in/deg)	21,654.52	21,654.52	677.755

Table 4-2: Hi-Lok and 18-8 Stainless Steel Model Displacement Results in FEMAP

Number of Fasteners	Type of Fastener	Axial Displacement [inches] at 1000 pounds
2	Diameter Defined Strip (3.3.5)	0.0184
	Radius Defined Strip (3.3.5)	0.0178
	Solids (3.3.2)	0.0178
	Beams (3.3.3)	0.0184
	Springs (3.3.4)	0.0183
4	Diameter Defined Strip (3.3.5)	0.0158
	Radius Defined Strip (3.3.5)	0.0153
	Solids (3.3.2)	0.015
	Beams (3.3.3)	0.0152
	Springs (3.3.4)	0.0152
6	Diameter Defined Strip (3.3.5)	0.0144
	Radius Defined Strip (3.3.5)	0.0139
	Solids (3.3.2)	0.0133
	Beams (3.3.3)	0.0135
	Springs (3.3.4)	0.0135
8	Diameter Defined Strip (3.3.5)	0.0128
	Radius Defined Strip (3.3.5)	0.0126
	Solids (3.3.2)	0.0122
	Beams (3.3.3)	0.0123
	Springs (3.3.4)	0.0123

Table 4-3: Aluminum POP Rivet Model Displacement Results in FEMAP

Number of Fasteners	Type of Fastener	Axial Displacement [inches]
2	Diameter Defined Strip (3.3.5)	0.0152
	Radius Defined Strip (3.3.5)	0.0153
	Solids (3.3.2)	0.0159
	Beams (3.3.3)	0.021
	Springs (3.3.4)	0.0209
4	Diameter Defined Strip (3.3.5)	0.0136
	Radius Defined Strip (3.3.5)	0.0137
	Solids (3.3.2)	0.014
	Beams (3.3.3)	0.0166
	Springs (3.3.4)	0.0165
6	Diameter Defined Strip (3.3.5)	0.0123
	Radius Defined Strip (3.3.5)	0.0124
	Solids (3.3.2)	0.0126
	Beams (3.3.3)	0.0144
	Springs (3.3.4)	0.0143
8	Diameter Defined Strip (3.3.5)	0.0109
	Radius Defined Strip (3.3.5)	0.011
	Solids (3.3.2)	0.0115
	Beams (3.3.3)	0.013
	Springs (3.3.4)	0.013

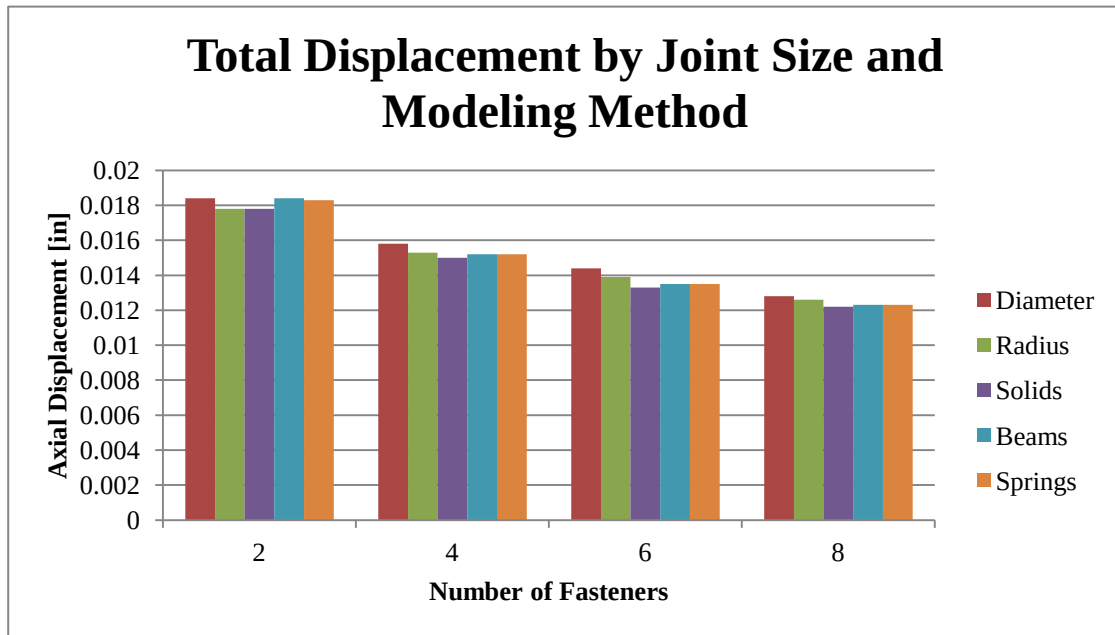


Figure 4-18: Hi Lok and Stainless Steel Displacement Estimates

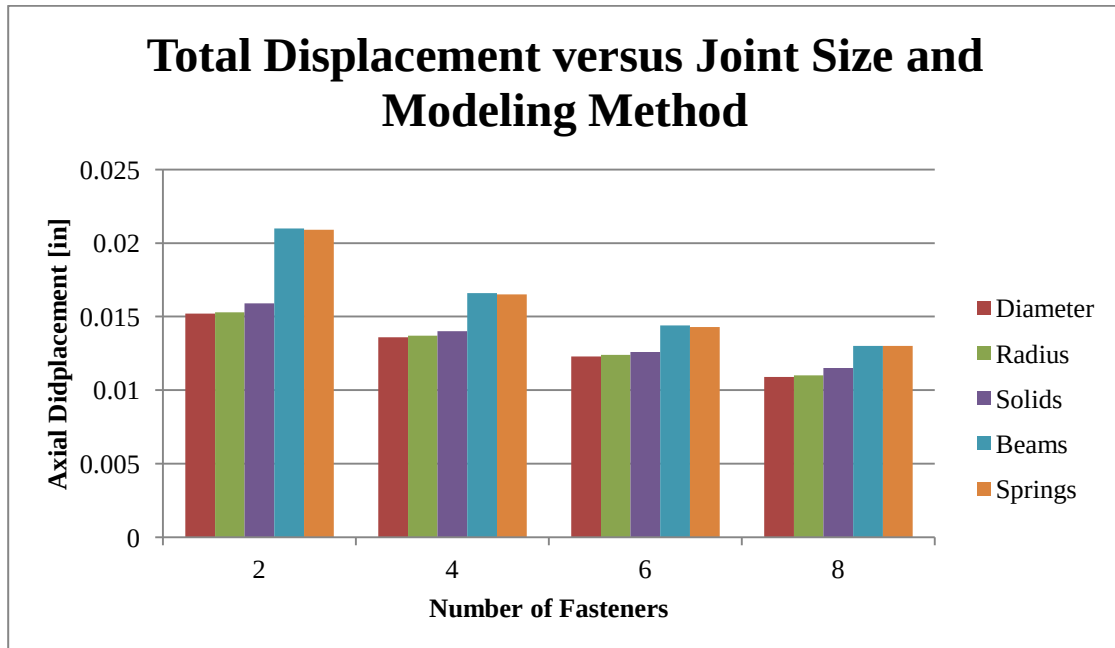


Figure 4-19: POP Rivet Displacement Estimates

It can be clearly seen from the data tables and graphs that both the diameter and radius definitions of the connector pad strips compare within about five percent in almost every case. Therefore, it can be assumed that the method of knocking down the E_{eff} and G_{eff} with the ratio of the areas of the bolt and pad is a reasonable approach to modeling such joints.

CHAPTER 5: EXPERIMENTAL TESTING AND RESULTS

The previous analytical approaches were validated by performing a series of simple tensile tests.

5.1 The Physical Specimens

Chapter four contains the details of the specimen and fastener geometry. This section serves primarily as a summary of the variation in the specimens that were tested. The first type of fastener is the Hi-Lok fastener. Four specimens were fabricated using this type, each with more fasteners than the last, ranging from two to eight. The same fastener configurations were fabricated using the 18-8 stainless steel fasteners and the POP rivets. On the stainless steel bolted specimens the torque setting on the fasteners was 8 lb-ft in accordance with standard bolt torque setting specifications.

The adhesively bonded samples were fabricated to compare the displacement behavior with that of mechanically fastened connections. As mentioned in the previous chapter, the bondline thickness was controlled on these specimens with 3M Glass Beads with a thickness of 0.005 inches. By clamping the specimens together with thick aluminum angles on either side of each flange, the clamping force was more evenly distributed, helping to avoid adhesion concentrations. Per the mixing directions on the Hysol, a mass ratio of 6:1 adhesive to hardener was used, with only 0.25 grams (0.3% by weight) of the glass beads for the total amount of Hysol. 60 grams of Hysol were mixed with 10 grams hardener for these specimens, with some adhesive left over.

Since there was a large number of specimens to be tested, a naming scheme was created in order to easily keep track of the samples and convey quickly the sample of interest. The convention is as follows:

$$NXXY-ZZ$$

where N stands for “Nested,” XX represents the fastener type, Y represents the number of connectors, and ZZ represents the specimen number. The possible values in the XX field are “ HL ,” “ SS ,” “ PR ,” and “ AD ,” which stand for “Hi-Lok,” “Stainless Steel,” “POP Rivet,” and “Adhesive” respectively. The value of Y varies between 2, 4, 6, and 8, based on the number of fasteners. In the case of the adhesive specimens, Y takes the value of the number of specimens corresponding with the overlap length (i.e. if the overlap length is 5.75 inches, Y is 8). Finally, the value of ZZ represents the specimen number of that type. Some of the samples tested had duplicates so that repeatability could be tested or other behaviors may be observed. Therefore, values of 01, 02, 03, et cetera are acceptable values of ZZ . From here on, specific samples will be referred to with this naming convention.

5.2 The Test Setup

The test setup for this series of quasi-static axial tests is relatively simple, though there are several important considerations that require special preparation.

5.2.1 *Preparing the Specimens*

Each specimen required special attention before it could be tested in order to ensure that the test was able to capture all of the behavior it was intended to. As mentioned in chapter one, a custom set of grips was designed in order to minimize the effects of bending in the joined region. However, since it was uncertain whether or not the grips would function as planned, the bending behavior of the joints was a desired output of the tests. In order to capture this behavior, axial strain gages were placed on the specimens. In order to see where the gages needed to be placed, a practice test was performed with far more gages connected than future specimens would need. In this way, the areas of highest bending could be identified, and gage placement could be restricted to those regions for greater gaging efficiency. The test case was performed on the NHL6-01 specimen, which was equipped with 20 gages. Figure 5-1 shows 10 gages; the other 10 are directly opposite of those shown. By placing the gages on opposing sides, the bending behavior can be seen at the points of the gages. Also, notice that there are two set of far field gages on the larger angle section. Comparison of these two strains serves to ensure that stress concentrations from the fasteners are not reaching out as far as the gages (i.e. if they match, then there is no stress concentration factor at the gage group closer to the joint, as there certainly is not one at the farthest group). Also, by having the far field gages, the level of bending behavior can be seen as a function of the location along the specimen. The gages are numbered in the same fashion, regardless of the specimen, starting on the outer surface and labeling from left to right, top to bottom, and then following the

same convention on the inside surface (See Figures 5-2 and 5-3). Differences in strain gage readings will be denoted by coupling the two opposing gages together with a hyphen (-), as seen in the results of this trial test (Figures 5-4 through 5-8).

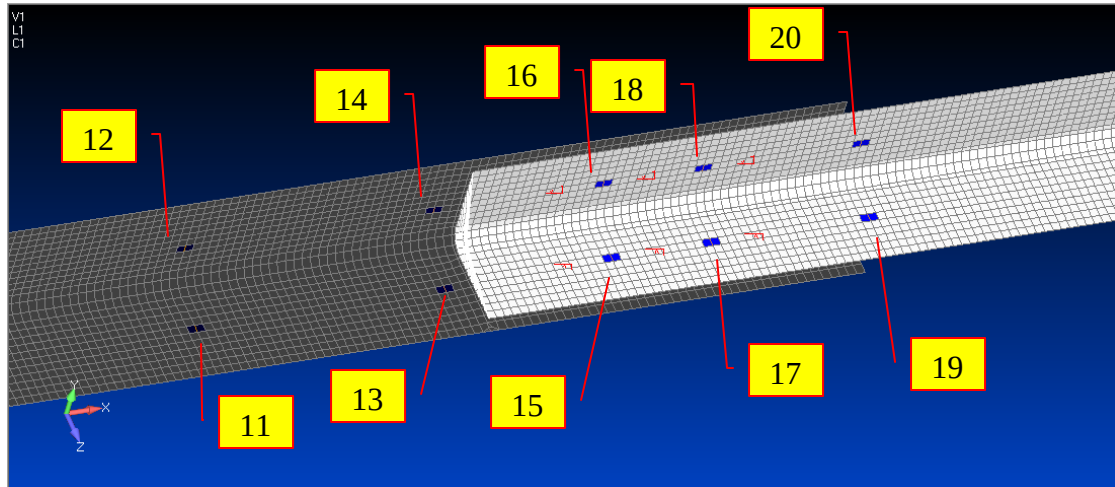


Figure 5-1: Strain Gage Locations for NHL6-01 (Fasteners in Red)

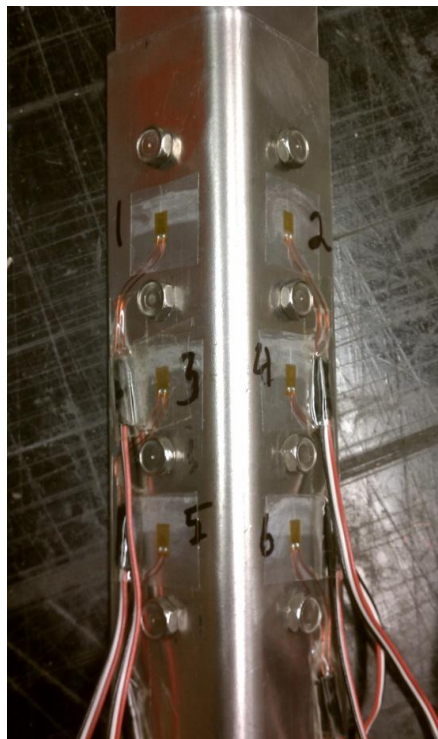


Figure 5-2: General Strain Gage Labeling Convention on Outside Surfaces

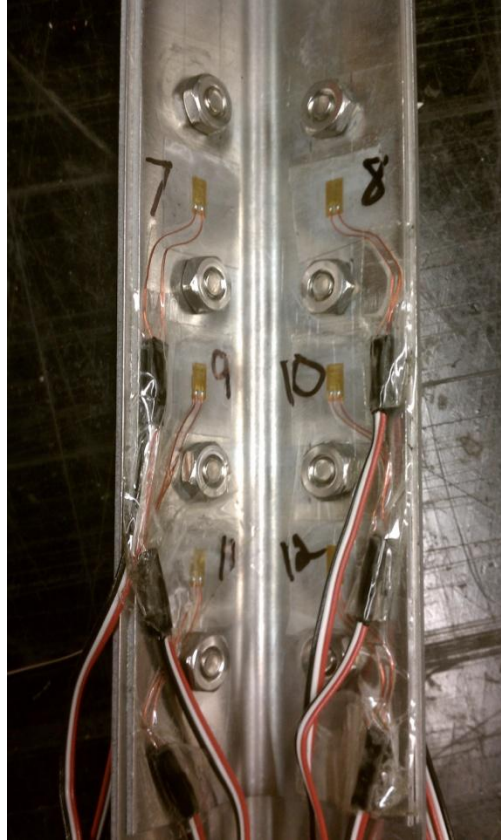


Figure 5-3: General Strain Gage Labeling Convention on Outside Surfaces

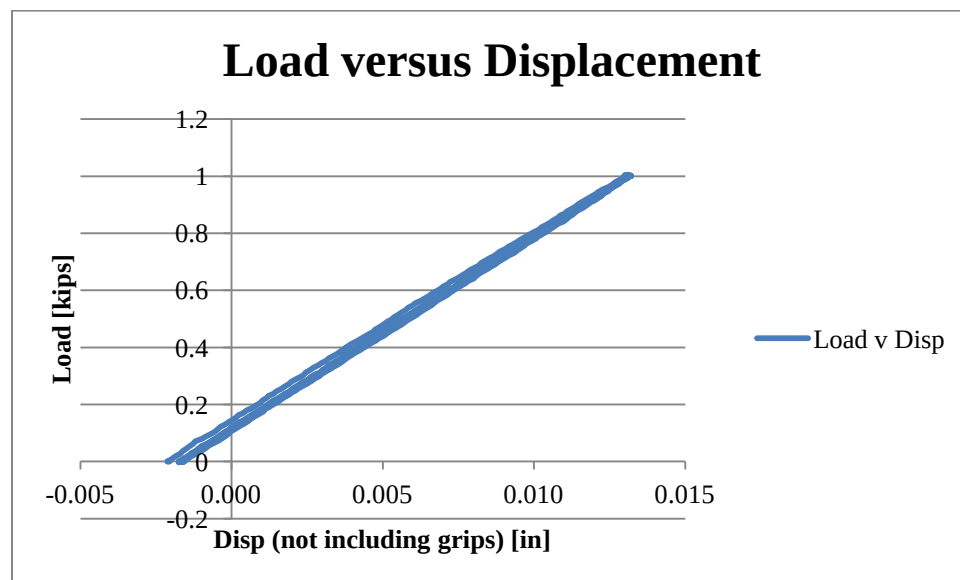


Figure 5-4: Load versus Displacement Cycles for NHL6-01

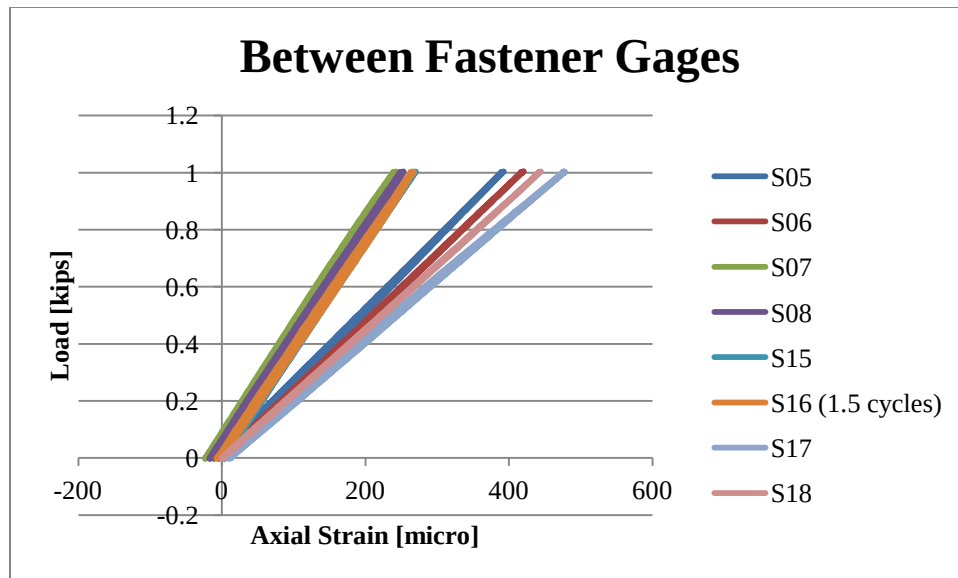


Figure 5-5: Axial Strain Data Between Fasteners

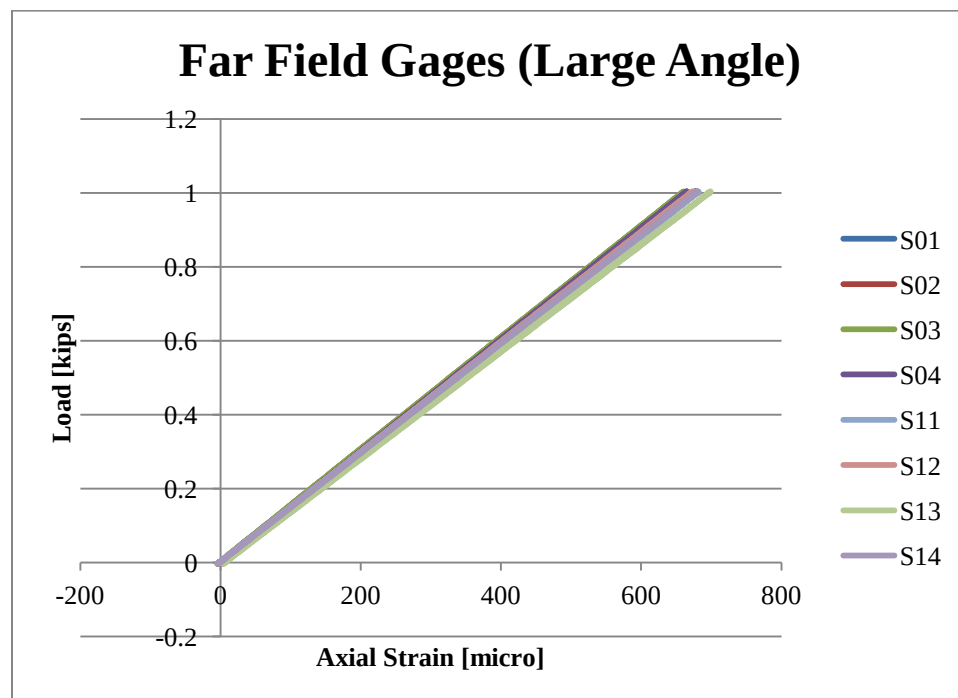


Figure 5-6: Far Field Axial Strain for Large Angle

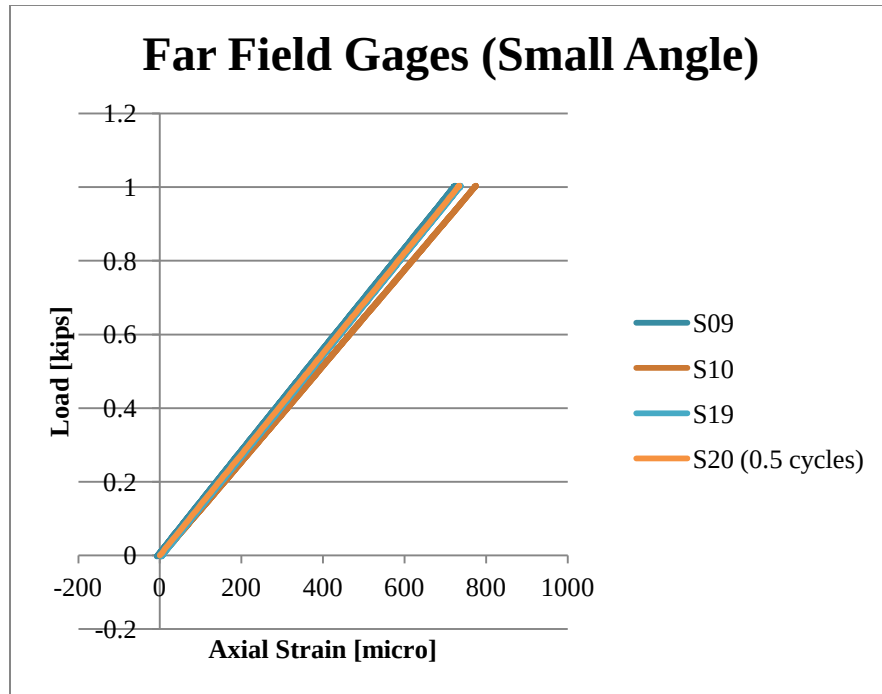


Figure 5-7: Far Field Axial Strain for Small Angle

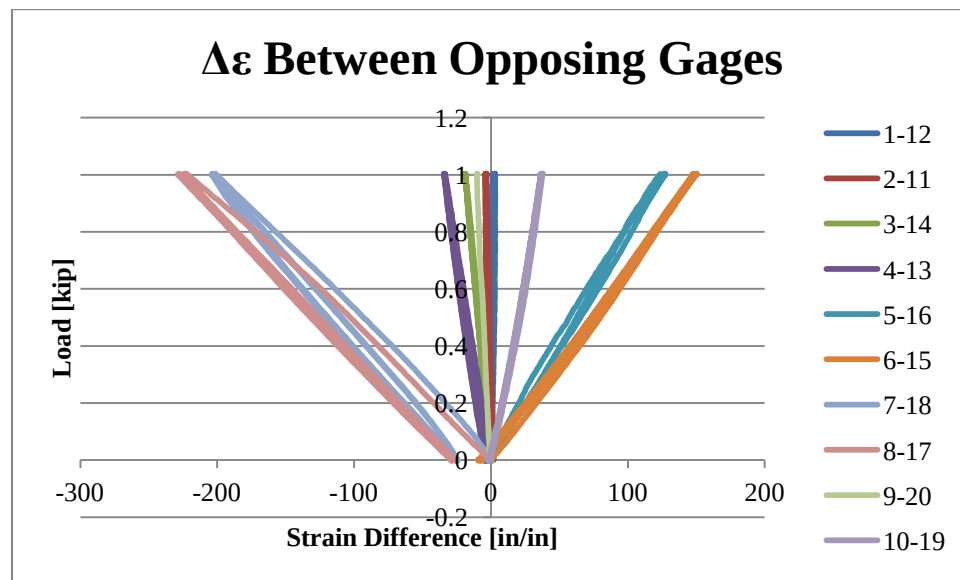


Figure 5-8: Opposing Strain Differences for NHL6-01

The preceding figures display the behavior of the test sample. First, it should be noted that all tests and analyses are being done in the linear elastic regime. This can be seen in the loading cycles in Figure 5-4 as well as the axial strain data in those

following it. For this test, the two primary concerns were the bending behavior and the far field gages on the larger angle so that the number of gages needed on subsequent tests could be determined. Looking at Figure 5-6, it is easy to see that each of the eight gages reads a very similar strain, with only a 5% maximum variation in the readings. This demonstrates that the bending component of the strain is much smaller than that of the axial component. This is contrasted with Figure 5-5, where the gages show a slightly larger spread, demonstrating more bending contribution. Additionally, from Figure 5-8, it can be seen that the maximum curvature is present with the pairs 5-16, 6-15, 7-18, and 8-17. On this particular specimen, these gages are those between the fasteners (as seen in Figure 5-5), indicating that the maximum bending behavior occurs within the overlap region. Figure 5-9 shows the radius of curvature defined by each pair of gages. Since analytically the strains are proportional to the curvature multiplied by the total thickness, the ratio of the radius of curvature to the thickness can be computed as the inverse of the strain difference, like so:

$$\frac{R}{t} = \frac{1}{\epsilon_{top} - \epsilon_{bottom}} \quad (5.1)$$

where R is the radius of curvature, and t is the total specimen thickness. Using these pieces of information it is reasonable to only place gages between the fasteners in all subsequent tests, knowing that the smallest radius of curvature is on such a large order compared to the thickness (the smallest radius is approximately 70,000 times greater than the thickness). This way, if the bending behavior is to become important, the most important data will have been gathered. These between fastener gages also

provide a sanity check that the load is being transferred properly between the two connected angles.

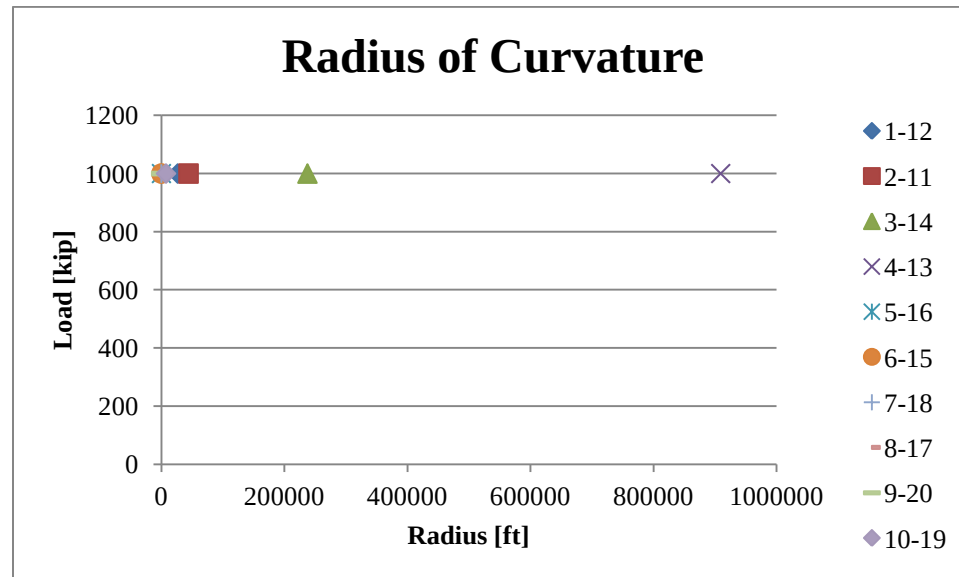


Figure 5-9: Radius of Curvature for Test Specimen

With the so called practice sample test complete, the gages could be placed on the remaining specimens. The gages used were Micro Measurements brand 3 lead axial strain gages with a gage factor of 2.14 and a nominal resistance of 350 Ω . The specimens were labeled, marked, and scribed such that the marks were indented slightly into the aluminum. They were then sanded using 600 grit sandpaper, and cleaned thoroughly with MEK. At this stage, the gages were applied using a proper CN strain gage adhesive, and the connections were checked with a multi meter to ensure that none of the internal wiring was compromised during the application process.

5.2.2 *Preparing the Test Machine*

Once the specimens were outfitted with strain gages and labeled with sample names and gage numbers, they were ready to be tested. The test machine is a 110 kip MTS axial testing machine with software for data acquisition on a local desktop computer (see Figures 5-10 and 5-11).



Figure 5-10: 110 Kip Material Testing System Machine



Figure 5-11: Native Data Acquisition Software

This native data acquisition software is able to read and report not only load and displacement data directly from the load cell and actuator, but is fully compatible with strain gage equipment. The beige boxes to the left of the computer in Figure 5-11 are the strain gage hookup locations and balancing equipment. However, many of the specimens needed more than the available number of hardware channels on this system. For this reason, focus shifted from using the native data acquisition to using an alternative software package written by Dr. Christopher Latham at UCSD. With this program, an external data conditioning cabinet containing 40 hardware channels could be connected to a desktop computer and used to read the strain data. Then, in the interest of consolidation, the load and displacement data from the native software could be transferred to the secondary data acquisition program through two standard BNC cables. Figure 5-12 shows the secondary data acquisition system that was used, including the cabinet and the desktop.



Figure 5-12: Secondary Data Acquisition System

Preparing the machine for testing is a relatively straightforward process. Compressed air is needed for the hydraulic system that powers both the actuator and the grips that hold the specimen in place. Once a large air compressor was started, a water pump is turned on to cool the system and maintain working temperatures. The machine itself has interchangeable grips that allow for specimens with varying cross sectional shapes and sizes to be tested. None of the grip styles would have worked well with the nested angle configuration, as the flat grips would have had a tendency to flatten the angles. Therefore, the grip style that accommodated these tests most was a notched grip that allowed for a 1.25 in diameter circular cross section. In order for this to work though, a set of customized end pieces needed to be designed to fit the angles and also the machine (Figure 5-13). The design process of these end pieces is included in the Appendix.



Figure 5-13: End Piece Grips (with Fasteners)

The end pieces were attached to each of the specimens using grade 8, 5/8 in, 1/4 in diameter bolts with 20 threads per inch. Using a bolt torque calculator by FUTEK Advanced Sensor Technology, Incorporated, the recommended torque setting was found for these bolts to be 10 lb-ft. Using a torque wrench, this level of torque was included on each of the 20 fasteners holding the end pieces to the angles to ensure maximum strength.

The specimens were placed into the machine and held in place using the hydraulic grips. Figure 5-14 shows a representative specimen in the machine and ready to be tested. Using the native MTS software to control the cross head and the actuator, the specimen is put into a zero load state, within the tolerance of the machine. For the best accuracy at the relatively small loads that these tests were done at (to stay in the linear elastic range), the setting on the machine are changed to the 0.8

inch range and the 11 kip range, as opposed to the default 8 inch and 110 kip range that would be used at higher load and displacement levels.



Figure 5-14: Representative Specimen in the Test Machine

The strain gages are each connected to one quarter of a Wheatstone bridge, which is in turn excited by a DC voltage supply. Using the conditioning cabinet, the gages can be individually zeroed at the null point of measurement. In other words, the

voltage output in each gage can be initially dialed into zero such that when strains occur, the Wheatstone bridge is thrown out of equilibrium due to the change in resistance in the gages, producing a strain reading.

The loading procedure was such that the linear elastic range was the main focus of the study. After several computational and experimental studies, a maximum load of 1000 pounds of force was selected for the studies. At this load, each of the specimens would be able to handle the load elastically. The most common loading procedure was a cyclic loading to 1000 pounds three times with 30 second holds at the top and bottom to ensure that good readings were taken at both the maximum and minimum load states (see Figure 5-15).

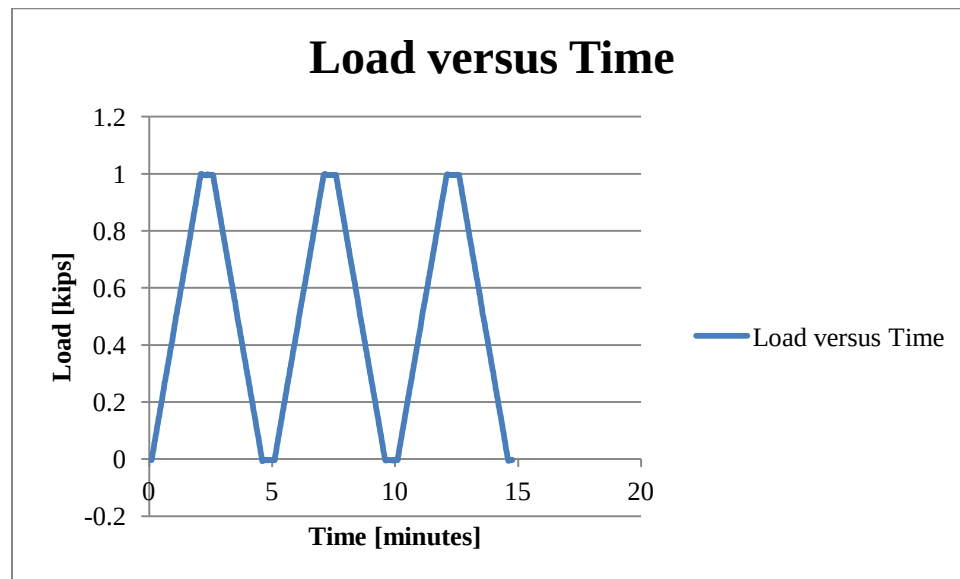


Figure 5-15: Typical Load versus Time Profile

5.3 Experimental Results

As mentioned previously, the primary concern for these studies is the macro-level displacement of the joint. Stresses and strains in the area immediately surrounding the fasteners were not of any concern since, for this project, only the preliminary design of flight structures is being considered.

5.3.1 End Piece Displacement Correction

The MTS test machine records the load data via a load cell located in the upper cross beam, while collected the displacement data from the displacement of the lower actuator as the test progresses. In order to correctly capture the behavior of the joint of interest, an experiment was done in order to understand exactly how much of the reported displacement was coming from each deformable section of a sample. For each specimen, there are five deformable sections: 1) the upper end piece connection, 2) the aluminum between the end piece and the central joint, 3) the central joint itself, 4) the aluminum between the central joint and the lower end piece, and 5) the lower end piece connection. Sections two and four can be calculated analytically simply by knowing the cross sectional area, Young's modulus, the length of the section, and the load. The cross sectional area was found using SolidWorks from the original drawings from which the angles were made. These sketches can be found in the Appendix. The third section was measured using a displacement transducer placed on the principal axes on either side of the angle (see Figure 5-16). Having these three sections measured, the displacement associated with the end piece connections, and

therefore not modeled in the finite element models, can be found as the difference between the experimental machine displacement and the sum of the displacements associated with sections two, three, and four.

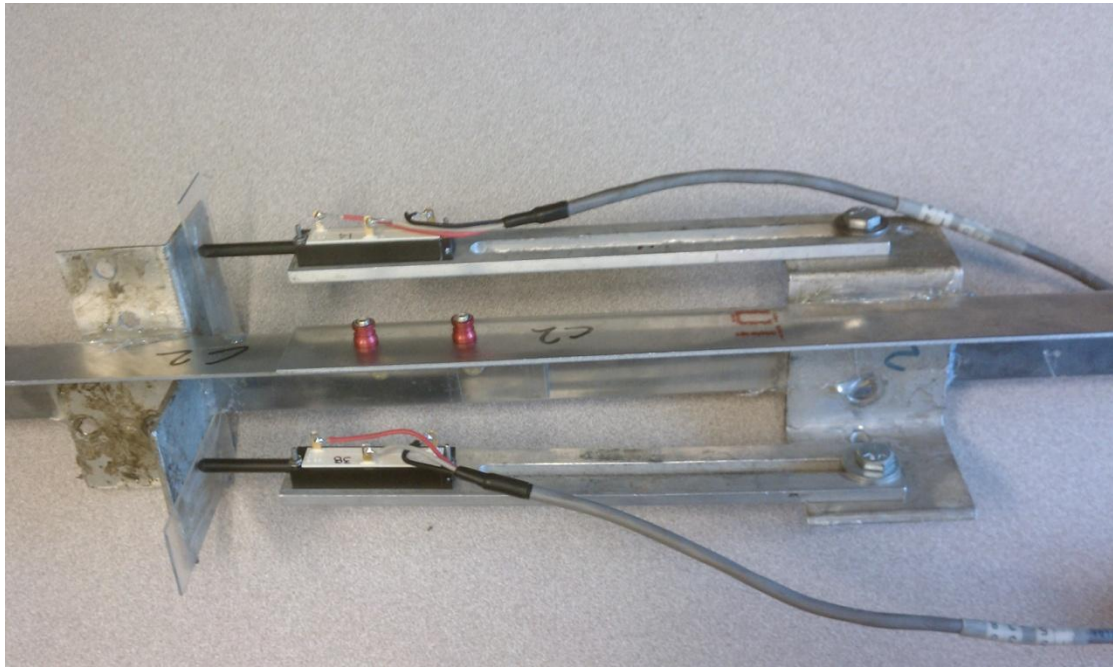


Figure 5-16: NHL4-01 with Displacement Transducers

Figure 5-17 shows the load versus displacement data for NHL4-01, which included the two aforementioned displacement transducers, called “Pot 1” and “Pot 2” in the data. The load cycle is the same as shown in Figure 5-15, and the average of the maximum displacements was taken for each of the transducers as well as the machine displacement (see Table 5-1). In the table, the average of the two transducer readings was taken as the section three displacement. Section two had a cross sectional area of 0.016337 in^2 and a length of 4.75 inches, yielding a displacement of 0.00297 inches at a 1000 pound load. Section four displaced 0.00454 inches at the same load as the cross sectional area is smaller (0.14762 in^2) and the section longer (6.8125 inches).

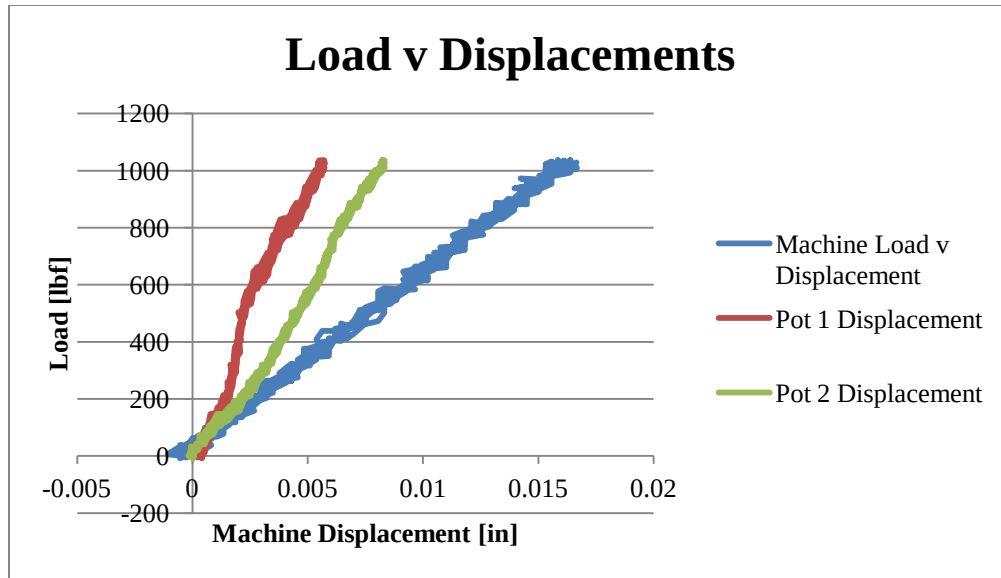


Figure 5-17: Load versus Displacement with Displacement Transducers

Table 5-1: Displacement of Deformable Sections for NHL4-01

Displacement Device	Average Value of the Displacement [in]
Machine Displacement	0.016516
Pot 1	0.005640
Pot 2	0.008283
Section 2	0.00297
Section 4	0.00454

Subtracting the average displacement of the two transducers (0.00696 inches) and the two aluminum sections from the machine displacement yields an end piece displacement of 0.002046 inches at 1000 pounds. For the purposes of these studies, this displacement will be assumed as linearly varying between 0.002046 inches at 1000 pounds and 0 inches at 0 pounds. This displacement is subtracted from all of the

experimental data in order to capture the displacement of the regions that were modeled in the finite element models. In the plots of the load versus displacement in the following section, it should be noted that this offset is present.

5.3.2 Results of the Experiments

In this section, the results of the tests will be presented, organized by number of fasteners. Figures 5-18 through 5-25 show both the load versus displacement data for the entire loading cycle as well as just the initial slope of the curve (between 0 and 250 pounds of force) in order to see the differences in overall joint stiffness. It should be noted that the specimen NAD 6-01 is not included as it failed prematurely due to a manufacturing error. Additionally, two POP rivet specimens were tested with the two fastener configuration, one with aluminum rivets, and one with stainless steel fasteners. The original test of the aluminum fasteners demonstrated that the specimen was not capable of carrying the 1000 pound load, so a stronger fastener was used in order to be able to make comparisons to the other fastener types at the 1000 pound load level. Finally, it should be noted that the displacements all begin starting at -0.002046 inches. This is not due to unbiased data, but rather a result of the subtraction of the calculated displacements associated with the end piece grips and is a constant value subtracted from all of the data.

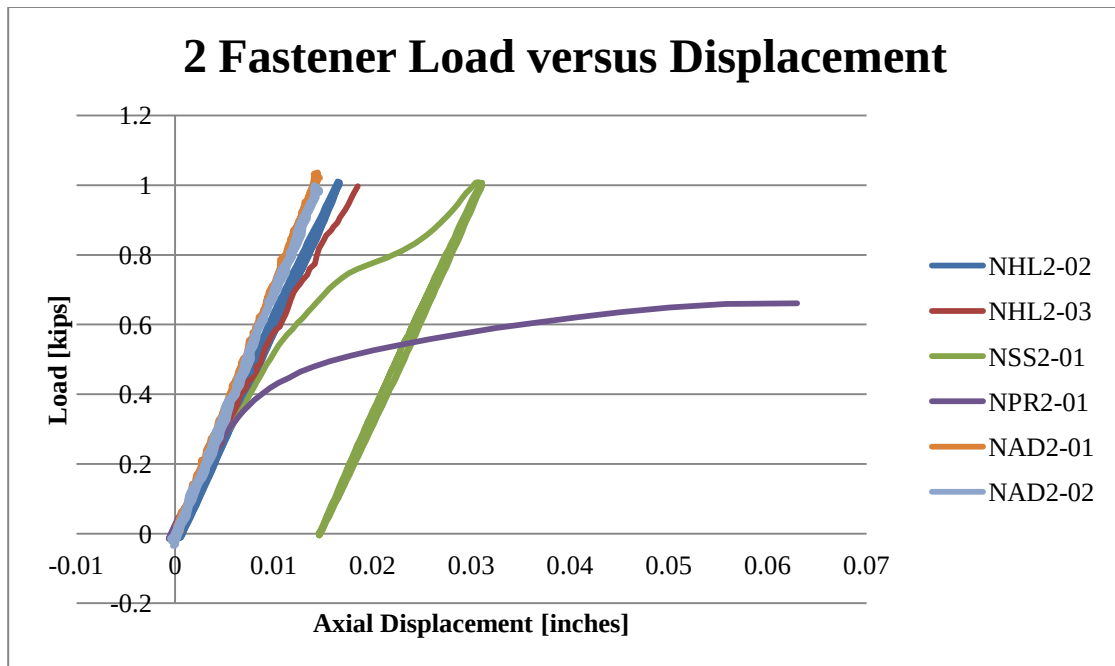


Figure 5-18: 2 Fastener Load versus Displacement Data

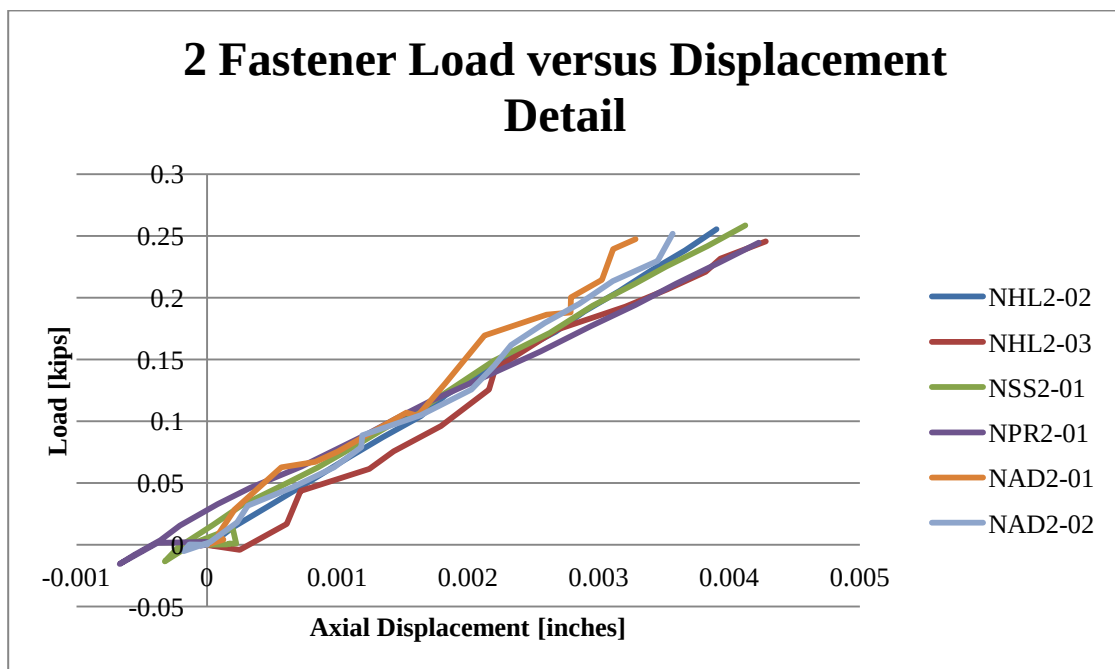


Figure 5-19: 2 Fastener Load versus Displacement Slopes

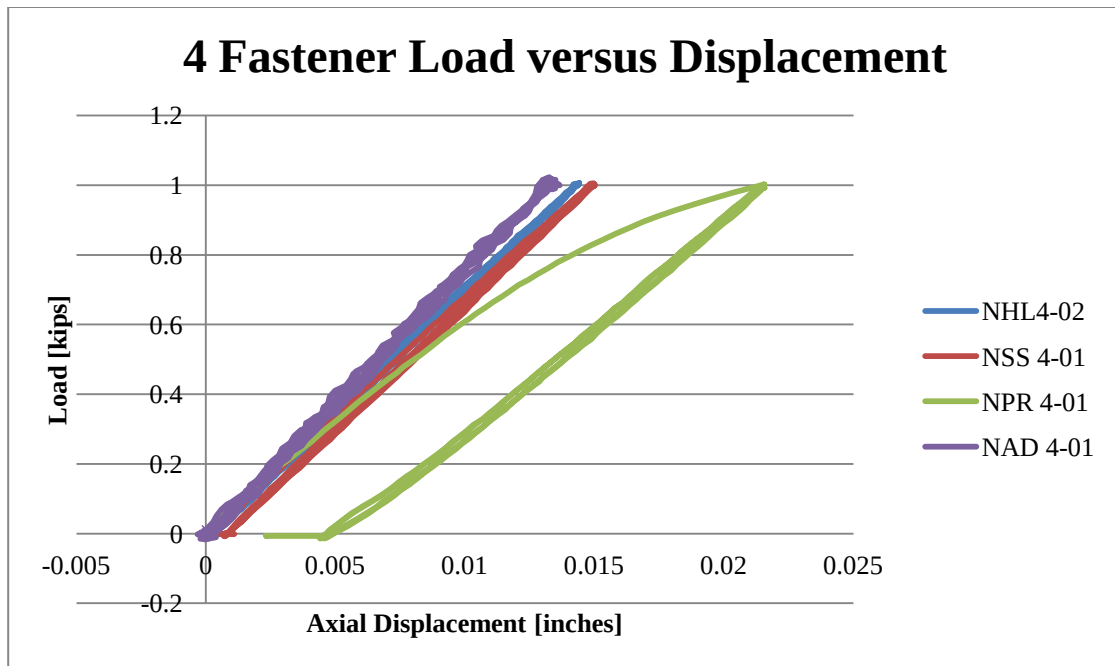


Figure 5-20: 4 Fastener Load versus Displacement Data

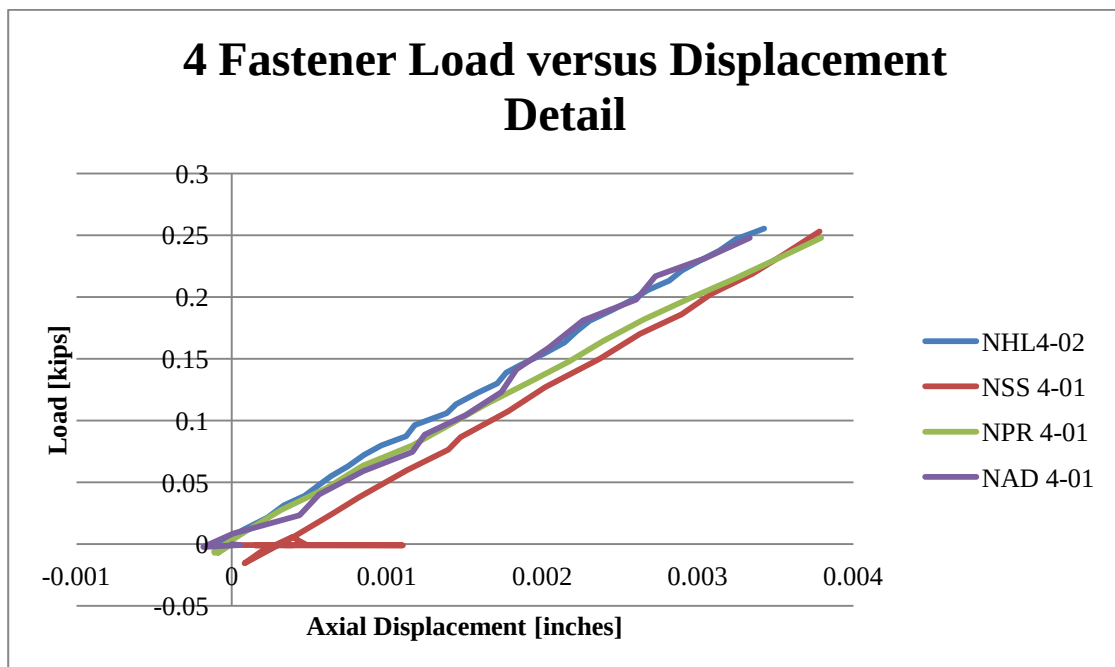


Figure 5-21: 4 Fastener Load versus Displacement Slopes

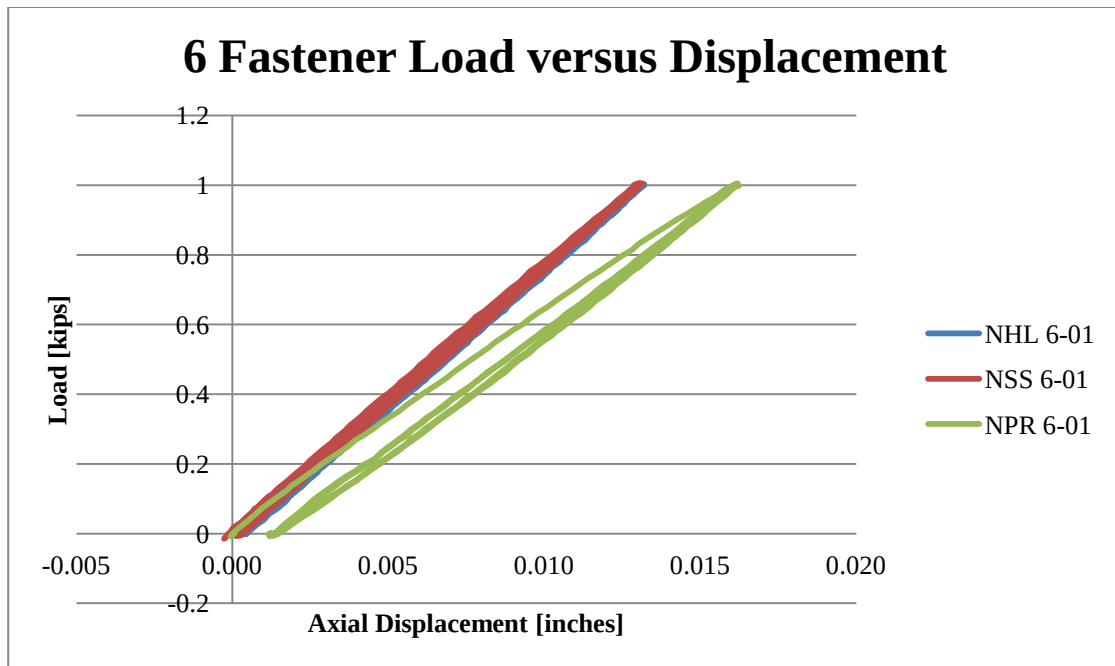


Figure 5-22: 6 Fastener Load versus Displacement Data

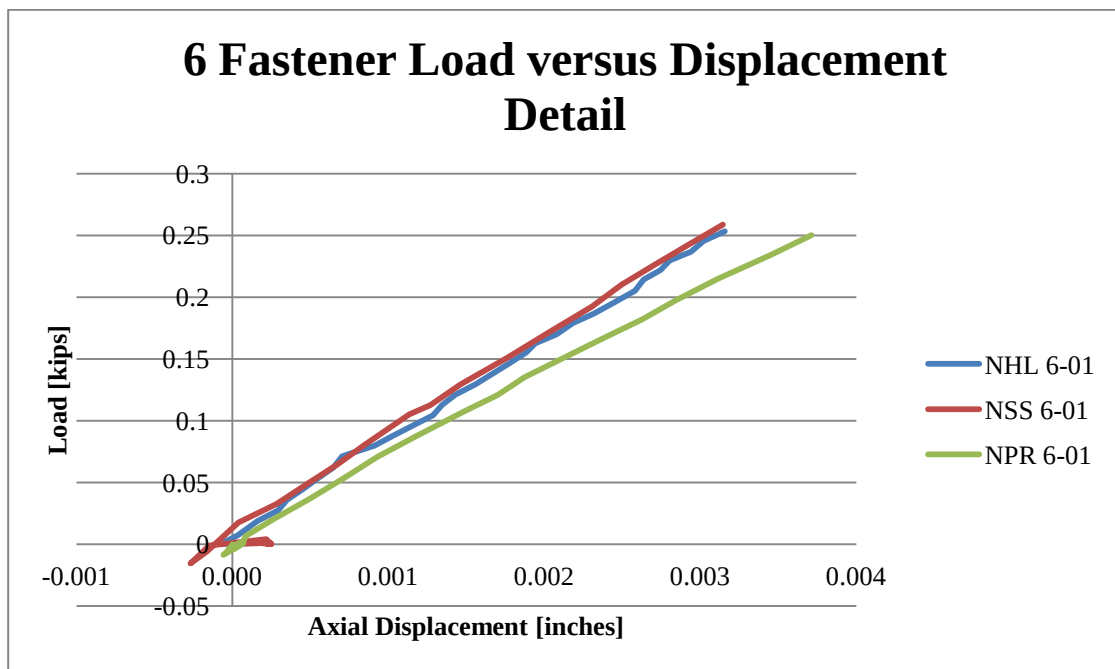


Figure 5-23: 6 Fastener Load versus Displacement Slopes

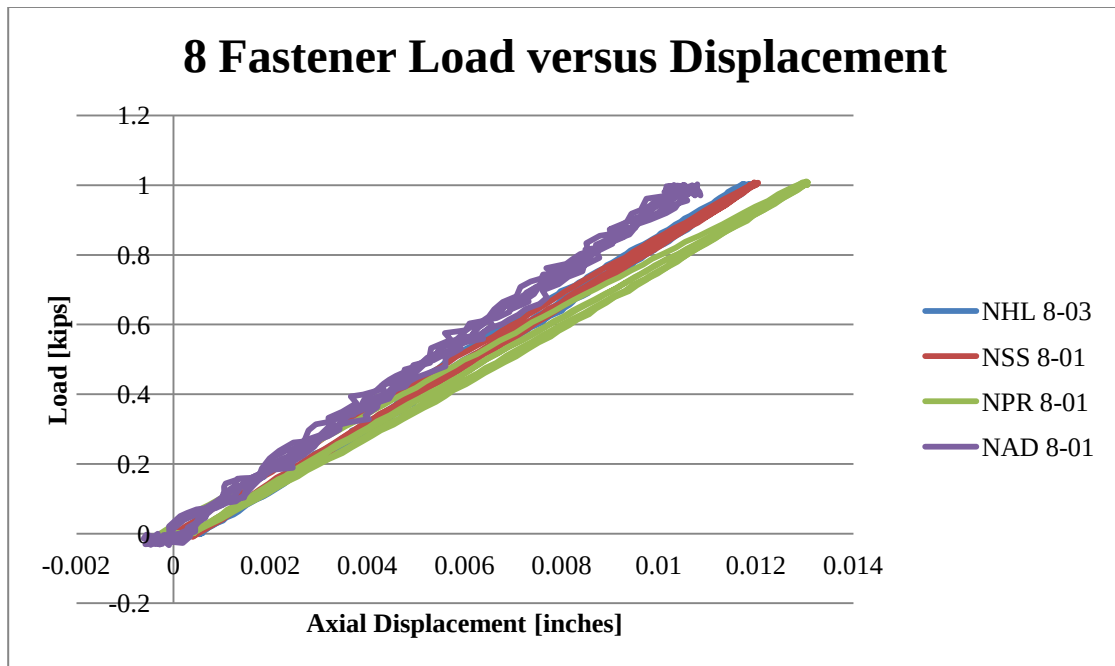


Figure 5-24: 8 Fastener Load versus Displacement Data

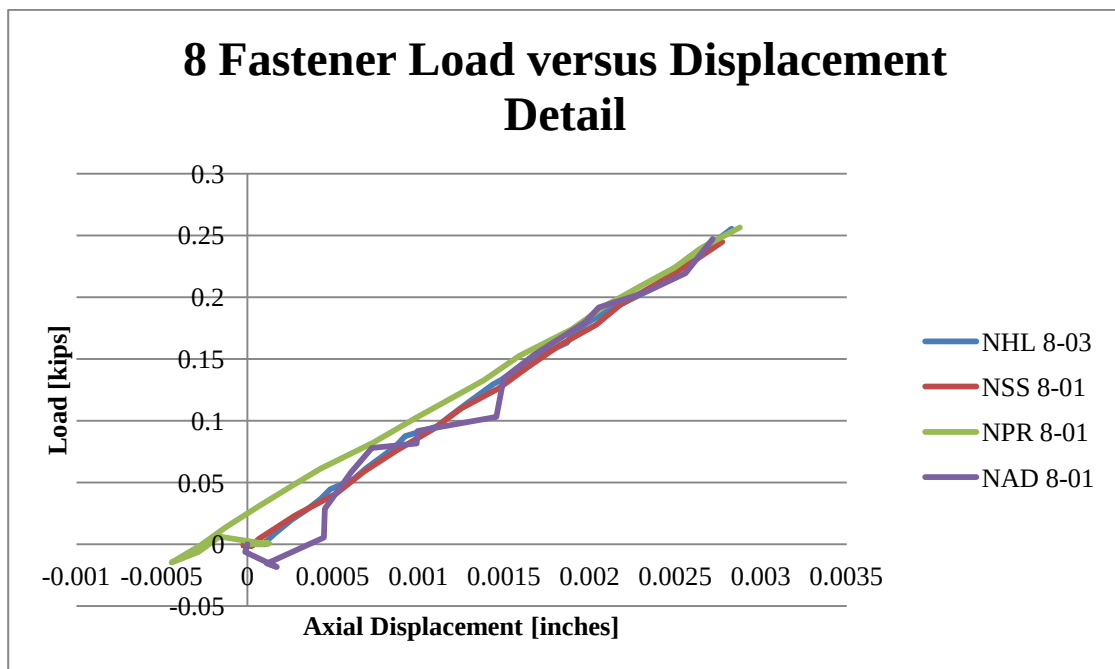


Figure 5-25: 8 Fastener Load versus Displacement Slopes

There are a few important observations to be made from looking at the test data. Firstly, for the larger number of fasteners, the overall stiffness of the joint due to

the mechanical connectors is very similar between the different types of fasteners. This is interesting considering the fact that different materials are used for each of the fastener types. However, as the number of fasteners in a group decreases, larger discrepancies can be seen between the stiffness of the joint. For example, while the POP rivet specimens always have the lowest stiffness (smallest slope on the plot), the difference between its specimen's stiffness and the next stiffest specimen becomes larger for the specimens with fewer connectors in the local group. Second, it should be noted that the adhesively bonded specimens behave slightly differently than the mechanically fastened joints. As seen in the plots above, the adhesive specimens were always the stiffest of the bunch. This is due to the much increased load transfer area compared to the finite load transfer zones of the fasteners as well as the properties of Hysol EA 9394. Additionally, the data for the adhesive specimens is often noisier than that of the mechanical fasteners. This is likely due to the differential stretching of the adherends described in chapter three and the larger load transfer area.

5.4 Experimental Correlation with Computational Modeling

5.4.1 Correlation between FEMAP Modeling and Test Results

As mentioned in previous discussion, the goal of these studies was to understand and simulate the global displacement behavior of the joints with the type and number of fasteners within the connection region as the primary parameter. Therefore, in the assessment of the quality of the modeling methods, the overall joint displacements will be compared. It should be noted again that the displacement associated with the end pieces is subtracted from the test data in order to represent that

only the area between the end piece connections was modeled in the computational analyses. Table 5-2 shows the comparison between the test results and the various modeling techniques that were employed. NPR2-01 and NSS 2-01 were analyzed within its linear range at 300 pounds of force in order to make a valuable comparison. Similar analysis was done for NPR 4-01, except with 500 pounds. However, the displacement due to the end piece connections was calculated at 1000 pounds, and therefore would reduce the displacement by too much for lower loads. It will be assumed that the displacement due to the end piece connection varies linearly from 0 to 0.002046 inches at 1000 pounds, making the value equal to 0.0006138 at 300 pounds and 0.001023 at 500 pounds.

The results of the testing and finite element analysis show several interesting trends regarding the modeling and behavior of the nested angle joints. Firstly, we will look at the elements that have been used in previous research, such as the solid, beam, and spring models. In general, each of these models accurately predicted the global behavior of the joint to within 10%, with a majority of the correlations falling within 5%. With these favorable results, the modeling methods are validated.

Additionally, one can compare the performance of the two groups of pad elements that were defined based on the radius or the diameter of the associated fasteners. Generally, both methods worked quite well, coinciding with the current state of the art modeling methods in nearly every case. In several cases, these elements outperformed elements such as the Timoshenko beam or CBUSH, particularly in the shorter overlap specimens. While neither definition of the pad strips was always more accurate, the radius definition tended to outperform the diameter definition. It can be seen that the predictions were not as accurate for the POP rivet specimens. This is due to the tested behavior of these specimens, as they tended to displace much less linearly than either of the steel fasteners, experiencing local yielding of the fasteners' thin walls with very low loads. Additionally, the cross section of the POP rivet is less accurately modeled than those of the Hi-Lok or 18-8 fasteners, as the mandrel is broken off inside the casing leaving a relatively unknown hollow cross section. As described in chapter four, an equivalent solid cross section was calculated to try to maintain the modeling techniques used for the Hi-Lok and stainless steel connectors. While this equivalent section may be accurate, it does not

account for the increased deformability of the thin walled POP rivet fastener under load. This explanation may describe why many of the predicted displacements were much less than the test behavior. The full connector pad did yield reasonably good predictions for the adhesive specimens. This is due to the models ability to imitate the continuous load transfer that the adhesive specimen experiences. The correlation in this area works to validate the modeling of the adhesive joints in the computer, as well as to demonstrate the difference in behavior of the adhesively bonded specimens and the mechanically fastened specimens. It should be noted again that the adhesive models were simply given the properties of Hysol EA 9394 as inputs. Figures 5-26 and 5-33 demonstrate graphically how the various modeling techniques match up with the test data. It should be noted that in general, the models all lay essentially in line with the test data. Additionally, notice that FRANC 2D/L studies were completed for the NHL and NSS specimens. The NPR specimens were omitted due to the high degree of nonlinearity in the testing, which FRANC 2D/L does not consider.

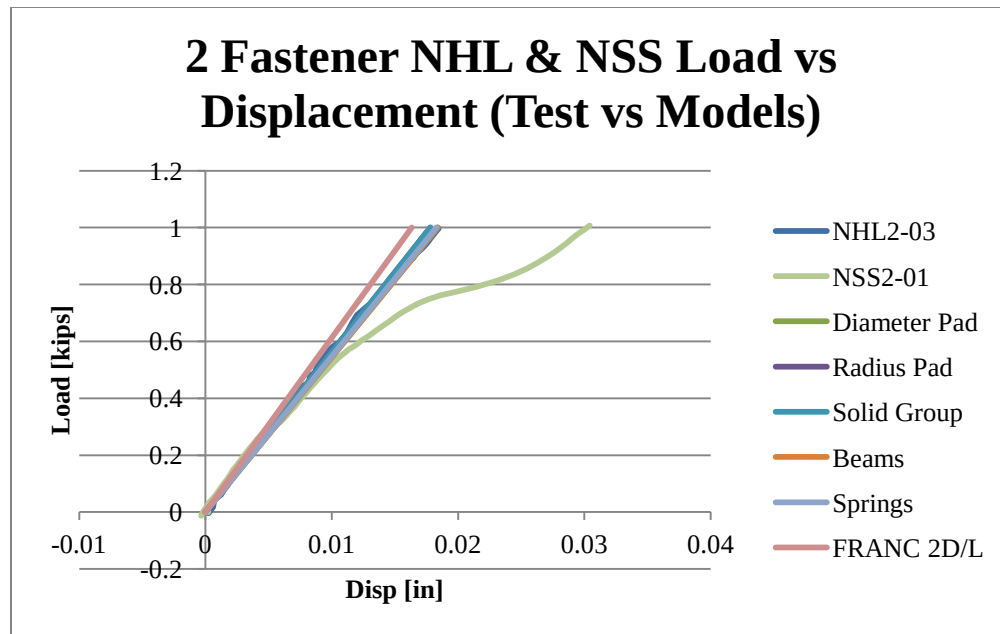


Figure 5-26: Load vs Displacement Comparison for Various Modeling Techniques for 2 Fastener Specimens (NHL and NSS)

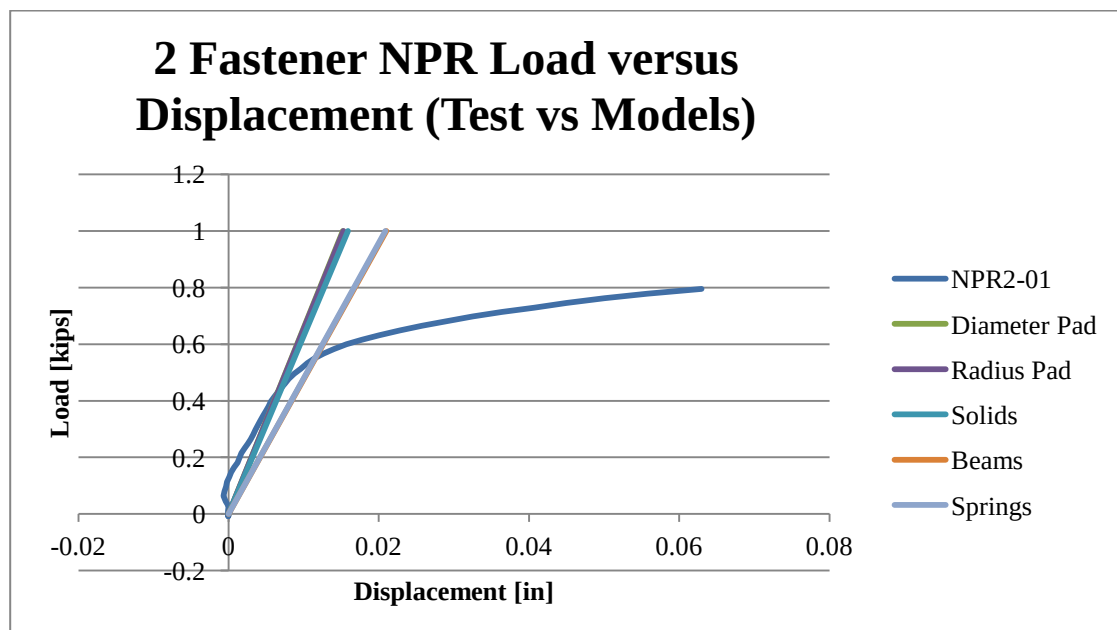


Figure 5-27: Load vs Displacement Comparison for Various Modeling Techniques for 2 Fastener Specimens (NPR)

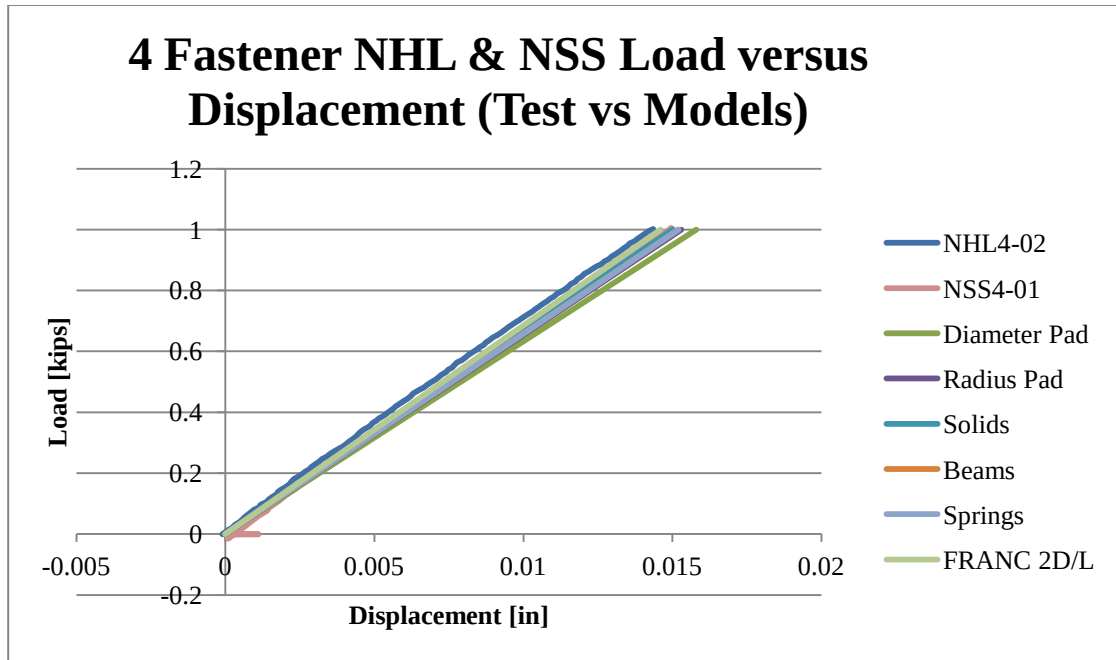


Figure 5-28: Load vs Displacement Comparison for Various Modeling Techniques for 4 Fastener Specimens (NHL and NSS)

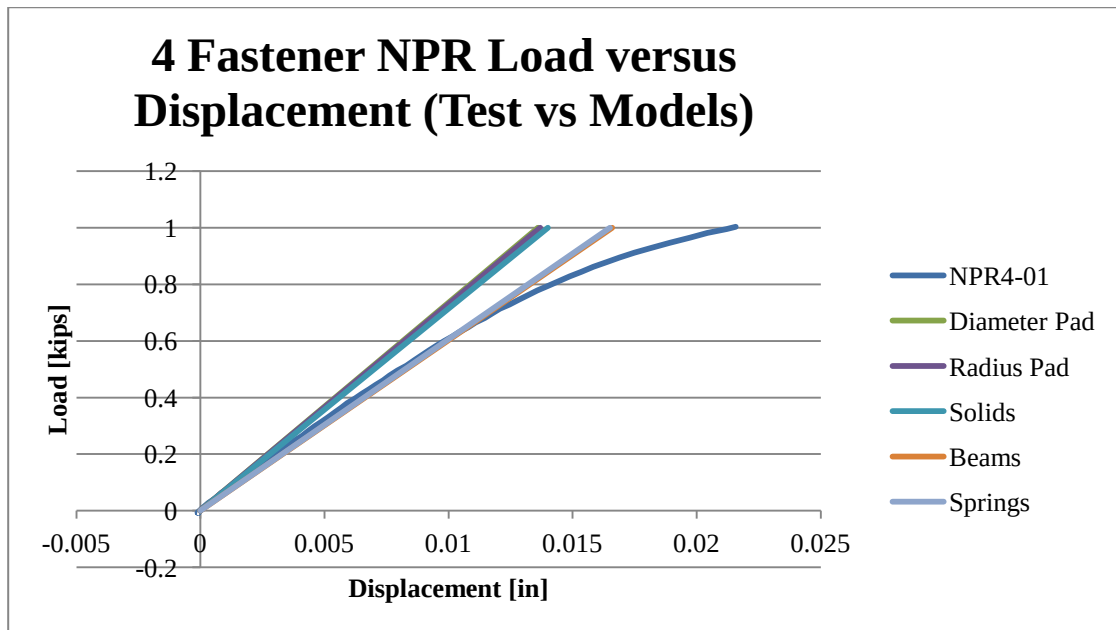


Figure 5-29: Load vs Displacement Comparison for Various Modeling Techniques for 4 Fastener Specimens (NPR)

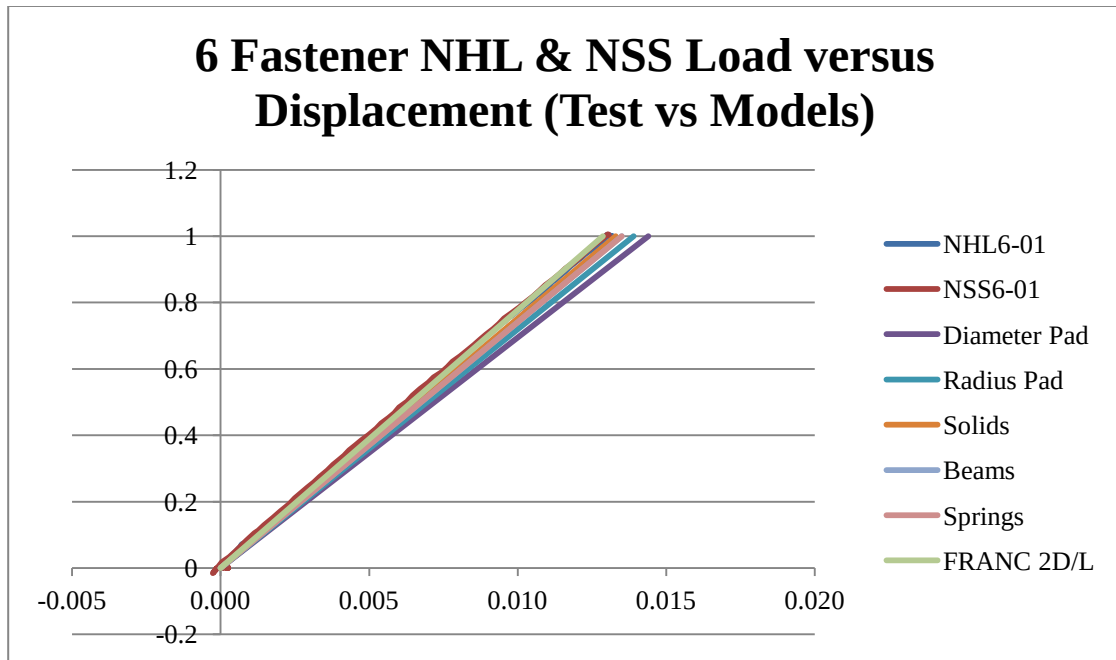


Figure 5-30: Load vs Displacement Comparison for Various Modeling Techniques for 6 Fastener Specimens (NHL and NSS)

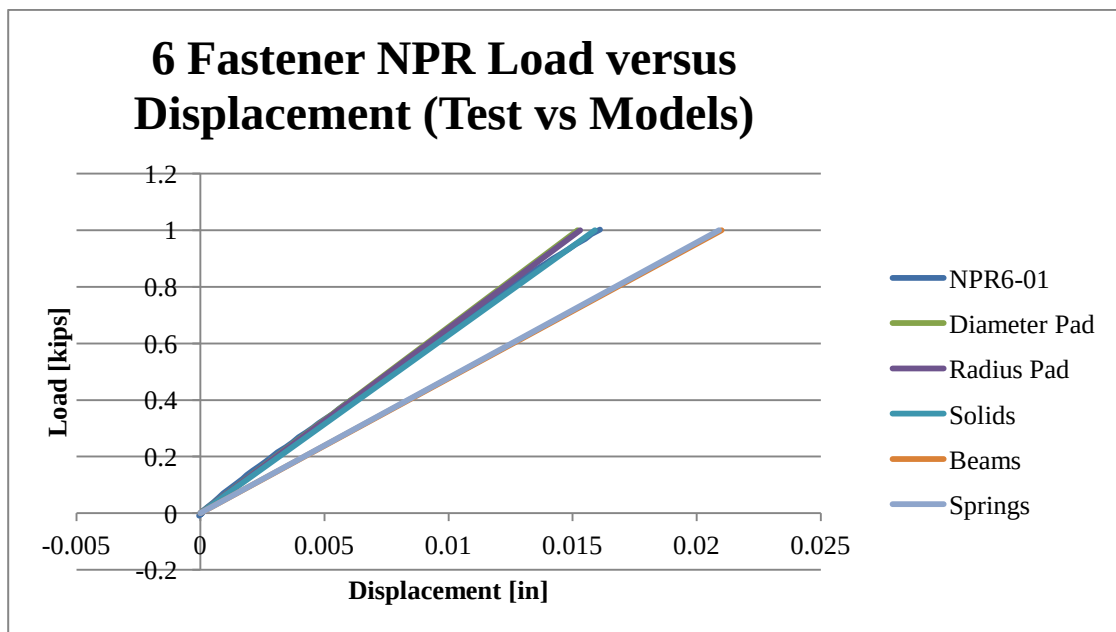


Figure 5-31: Load vs Displacement Comparison for Various Modeling Techniques for 6 Fastener Specimens (NPR)

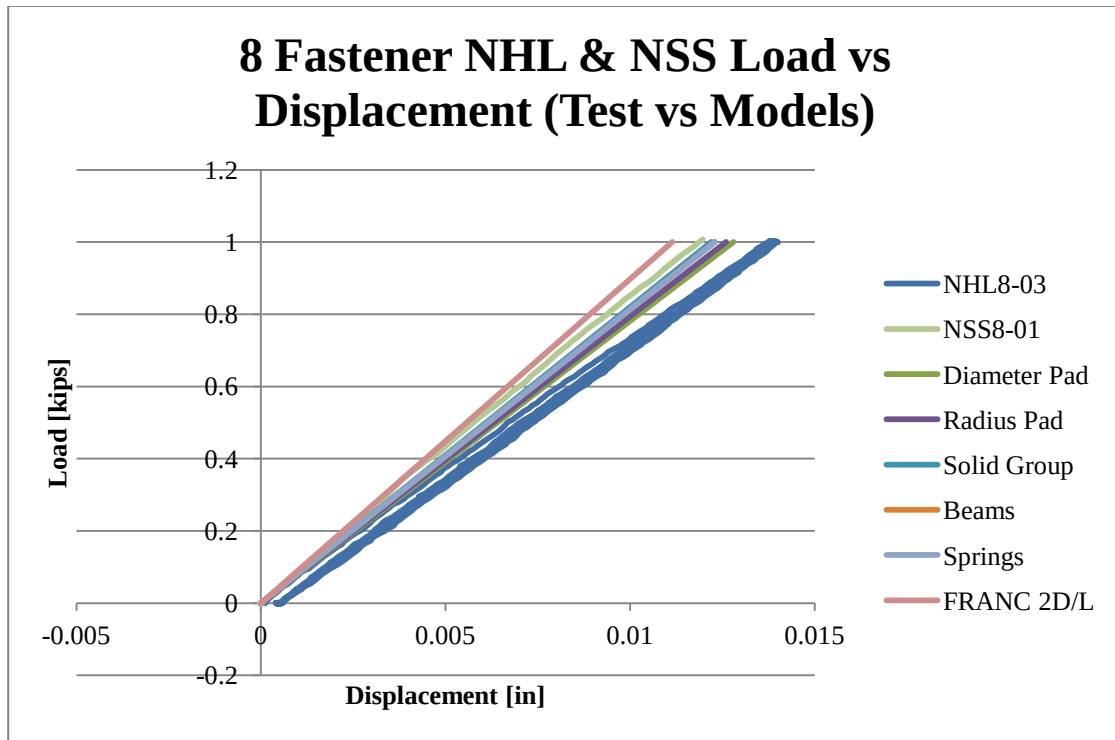


Figure 5-32: Load vs Displacement Comparison for Various Modeling Techniques for 8 Fastener Specimens (NHL and NSS)

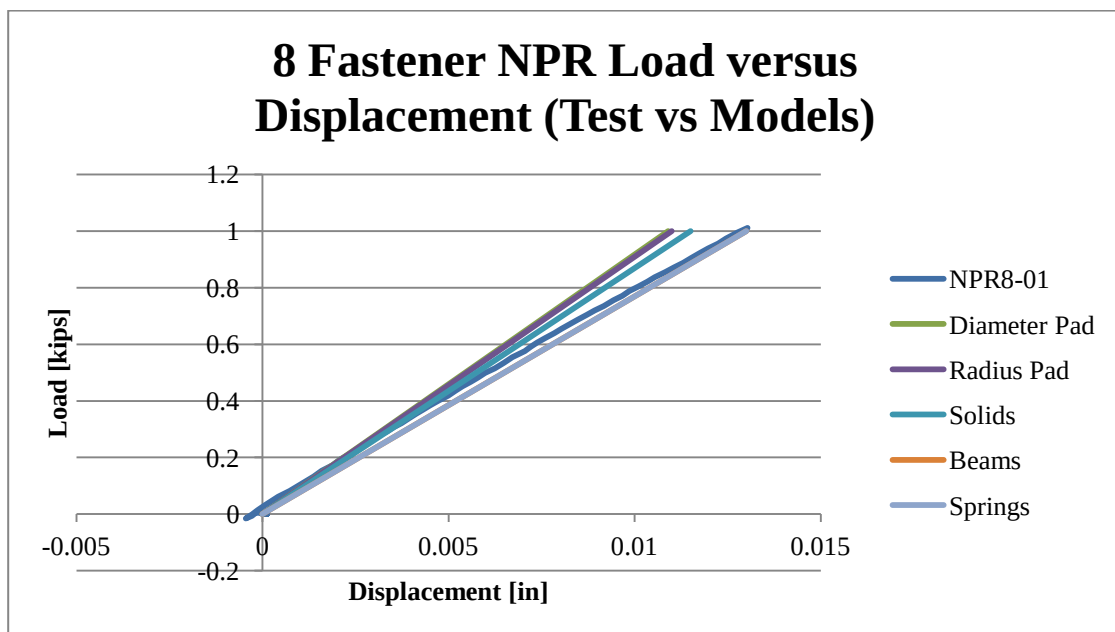


Figure 5-33: Load vs Displacement Comparison for Various Modeling Techniques for 8 Fastener Specimens (NPR)

5.4.2 Correlation between FRANC 2D/L Modeling and Test Results

The data for FRANC 2D/L can be seen in the plots preceding this section, however, one representative case will be addressed. In Figure 5-34, row one through row four represent the longitudinal displacement of each of the four rows of fasteners, with row one being the bottommost set and row four being the uppermost. In the plot, “Free End” represents the total joint displacement that has been the point of interest and comparison. It should also be noted that the dips in the data represent the locations of the fasteners within the section.

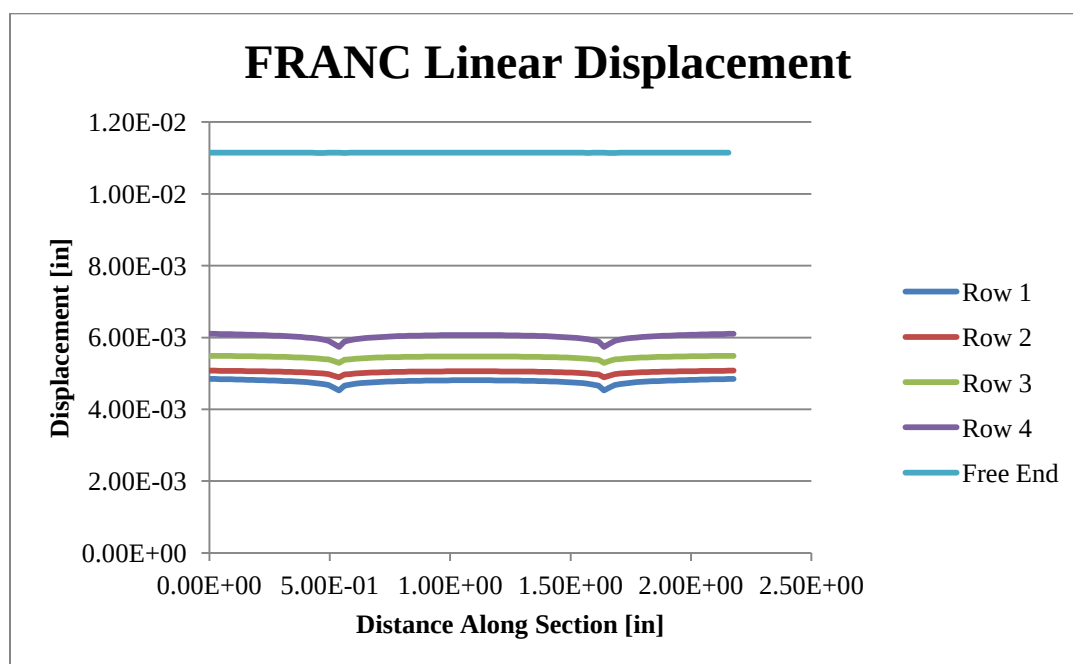


Figure 5-34: Various Section Displacements of FRANC 2D/L Eight Fastener Model Under 1000 lbf Load

It can be seen that the overall joint displacement for is equal to 0.01114 inches at the 1000 pound load. Referencing Table 5-2 shows that the displacement of NHL8-03, the specimen for which this model was created, was 0.012 inches. This represents accuracy to 7.167% of the test, which is roughly the accuracy obtained by the radius

defined connector pad for the same specimen. This quick test verifies that, in terms of displacement, such a model can be employed with a fair amount of accuracy.

5.5 Linear Versus Nonlinear Experimental Behavior

As a precursor to future development in the research, several tests were completed that investigated the onset of yielding in the joints. Future implementations of the models used throughout the current research will need to know whether or not the onset of yielding results in plasticity, or simply nonlinear elasticity that could be brought on as a result of joint geometry, preloading, friction, et cetera. Figures 5-35 and 5-36 show two specimens that were cycled into the nonlinear regime. Both of these specimens demonstrated plasticity, shown by the rate and direction of unloading, parallel to but offset from the linear load path. The offset is equal to the plastic (permanent) displacement in the specimen that occurs first in the area surrounding the fasteners, or in the case of the POP rivets, within the fastener. It should be especially noted that there were three load cycles in the linear regime during the test of NHL 8-03, and the linear response is shown lying in line with the first nonlinear load cycle.

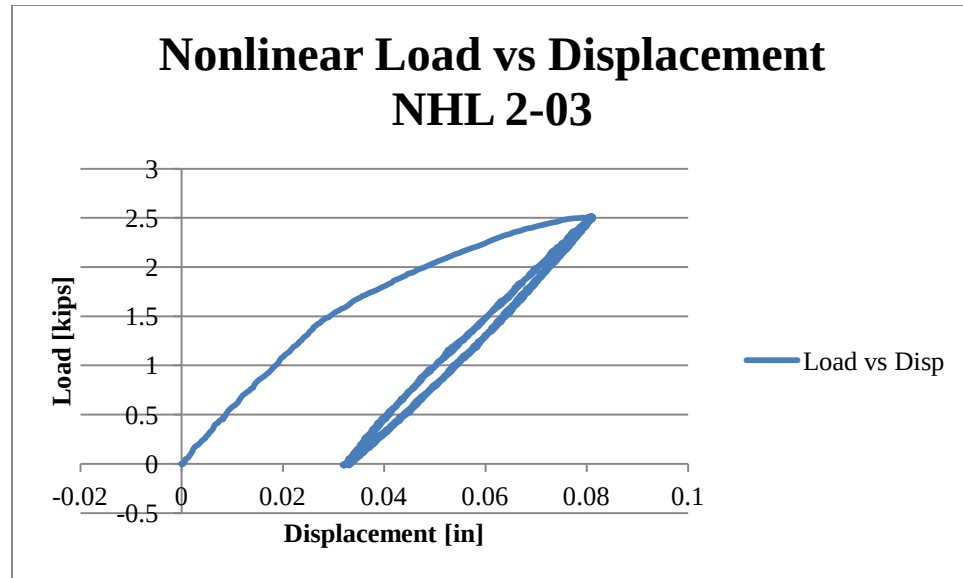


Figure 5-35: Linear and Nonlinear Response of NHL 2-03

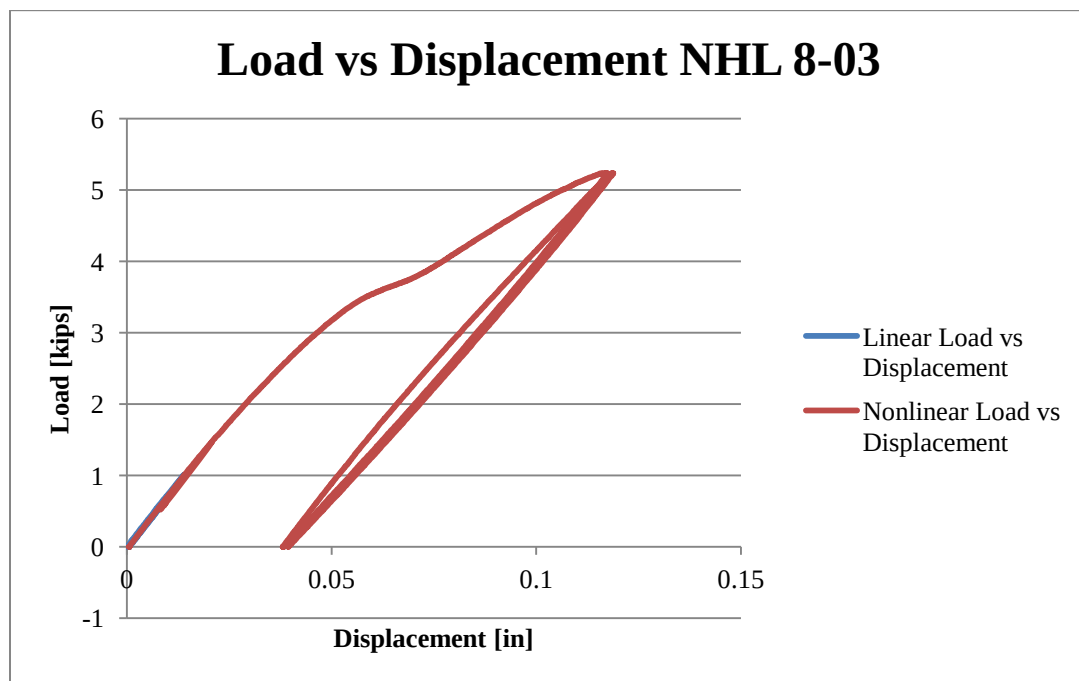


Figure 5-36: Linear and Nonlinear Response of NHL8-03

Since plasticity has been shown to occur in both extreme cases in terms of fastener number, and both with the most state of the art fastener type that was considered in the current research, it can be inferred that nonlinear plasticity models including material nonlinearities, contact, preload, or interference will need to be incorporated into future developments of the connector pad.

CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS

This research project has yielded a great deal of data and information regarding the behavior of mechanically fastened joints. In collecting and analyzing the data, several simplifications were able to be made to the current state of the art modeling techniques. Along with these developments come some additional suggestions that would act to further improve the simplification process, making it more general and complete.

6.1 Conclusions Drawn

The first goal for this research was to validate the models that represent the current state of the art methods of modeling mechanical fasteners in flight structures. As discussed primarily in chapter three, these models included those by Swift (1984), Cope and Lacy (2000). Through his extensive experimental work, Swift (1984) developed an approach for the effective moduli of mechanical fasteners that has been used in many research projects since, including the work of Cope and Lacy (2000). This project was able to further validate the solid, beam, and CBUSH models for the effective moduli by implementing material models into FEMAP according to such a development. Each of these three methods was able to accurately match the overall joint displacement, proving these modeling techniques to be accurate. The accuracy that was obtained varied slightly based on the fastener type, but averaged about 95% accuracy. This accuracy assumed that the value of K , which dictates the amount of bending displacement expected in the fasteners, was equal to $\frac{1}{12}$. The overwhelming success of these established models with state of the art materials helped build

confidence to move farther in the research. In other words, by proving that these models could be implemented competently, the next step could be taken in the direction of modifying such a development into a connector pad that would make the Swift (1984) model even more powerful.

The connector pad development follows directly from the idea of the effective fastener moduli outlined in chapter three. However, while the equivalent solid model still required the individual treatment of each fastener, the connector pad would theoretically allow more freedom to the engineer by allowing entire groups of fasteners to be treated together, simultaneously. This presents the opportunity for large savings in preliminary design modeling time and ease. Modeling of faster groups as discrete strips of solid elements defined by either the radius or the diameter of the fasteners contained within the strip proved to be a very effective method of modeling groups of connectors. The data correlation was equally as good as the previously established modeling methods, such as beam and CBUSH elements. Modeling the fasteners as an effective stiffness over the entire overlap length using this method would likely prove to be much too soft, yielding joint displacement predictions much larger than those observed in the test.

An additional observation of the research was that flattening the nested angles into a more traditional layered configuration does not appear to affect the analytically predicted displacement behavior considerably. This is shown in chapter four with FRANC 2D/L, in which a nested angle specimen was modeled as a more traditional single lap joint. This model was shown to match the displacement of the more exact

FEMAP model, showing that the displacements are not sensitive to specimen geometry in these cases. This small, but interesting observation may further be able to simplify the preliminary design process for similarly bent sections by allowing them to be modeled as simple flat plates. However, this has only been confirmed as accurate for the overall displacement of the specimen and is only true in analysis models, not physical testing. If stress between fasteners is the point of interest, then additional research must be conducted to verify whether this simplification is accurate still.

Additionally, it was found that the bending behavior that was exhibited by the joints did not affect the overall displacement of the joint to a great degree. This is shown in chapter five when the test specimen was analyzed. The radius of curvature of the specimen at different locations in the specimen was proved to be very large compared to the thickness, with a minimum radius around 350 feet. For future specimens that are loaded quasi statically within the linear regime, bending behavior of the nested angles need not be included as shown by this test.

Finally, it was shown throughout the research that the various connection types operated quite differently than one another in general. For example, the POP rivets, being made of aluminum, were shown to yield much larger displacements than either the Hi-Lok or stainless steel fasteners. Not only is this suggestive of the difference in materials, but also the difference in cross sectional geometry. Though both fasteners have the same outside diameter, the POP rivets act as thin walled tubes, while the other two fastener types both have solid cross sections. Different from all three of these is the Hysol EA 9394 adhesive. The adhesive by nature transfers the load

differently than the mechanical fasteners as it allows for continuous, rather than discrete shear transfer of the load. Hysol was shown to be stiffer than any of the other fastener types. This is specific to the configurations studied in the current research, including, but not limited to: the spacing of the fasteners, the size of the fasteners, edge distances, and geometry.

6.2 Recommendations for Future Research

In order to create a more complete and correct connector pad model, several additions could be included in the development to account for industry standards in geometry, modeling, and manufacturing.

Firstly, it would be correct to more stringently consider the edge distance requirements that are outlined in chapter two. In its original development, the connector pad represents a possible area in which connectors may be present. However, when implementing the connector pad, only the edge distance requirements in the direction of load were really considered. This consideration came in the form of the two connection regions outlined in chapter three, where the center connection region had the properties of all of the connectors within the overlap region, and the two outside pads had very low moduli to account for the zero load transfer condition that follows from the edge distance requirements. In order to create a more complete development, the connector pads could be shortened in the width direction (perpendicular to the loading direction). This would not change the calculation of the equivalent pad properties in any way except changing the pad area, though the method of knocking down the effective moduli by the fastener-pad area ratio remains the

same. This addition to the model is essentially cosmetic, but it enforces the industry standards for connector edge distance, making the preliminary design process more accurate.

Another consideration for future research would be to include the effects of fastener preload and interference on the displacement and global behavior of the joint. It could be postulated that preloading the fastener would increase the friction between the connected angles and therefore reduce the displacement compared to those specimens that are free from preloaded fasteners. Similarly, the effect of interference fits of the fasteners can be considered. It is unknown whether these effects would increase or decrease the stiffness of the specimens. It may be argued that local plasticity would occur sooner around the fasteners and may increase the displacement of the joints, though testing would be necessary to verify this hypothesis. Adding these considerations into the model would increase the accuracy of the displacement predictions in many applicable state of the art fasteners, such as the Hi-Lok fasteners already considered or standard bolts such as the 18-8 stainless steel fasteners that were tested for this research.

Late in the development of this project, it was discovered that the finite element code Hyperworks contains an explicit Hi-Lok fastener element. This was of great interest as this element is a combination of rigid elements, beams, and springs (Sampson (2012)). Further comparison of the current modeling techniques with this new commercially available Hi-Lok finite element is recommended to remain a relevant state of the art research subject.

Further considering the bending of the joint due to the eccentricity of the loading could create a more realistic and complete model. In this research, the custom end pieces were designed in a way that minimized the bending behavior of the joints. However, this is a highly idealized loading condition that would not generally be the case in flight structures. It was shown in the test results that greater amounts of bending can be observed in short overlap specimens and configurations. Therefore, though bending behavior was shown not to effect the displacements of the joints too drastically, investigating the bending behavior as a function of joint length could bring extra realism to the findings of this project. Related to bending is the symmetry of the joints. If, instead of the single lap nested angle configurations, double lap configurations were studied, the effects of bending would theoretically be eliminated due to the lack of eccentric loading mechanisms. This effect should be studied; not only for bending purposes, but also to test the connector pad methods developed in the research and validate the previous work of the solid, beam, and CBUSH elements.

The nature of the connector pad begs for an extension that allows for the modeling of very large numbers of fasteners by a single connector pad that spans the entire overlap region. A modeling technique such as this would allow the engineer to know only the number, size, and material of the fasteners in order to change the properties of the equivalent connector pad. Such a model would fall directly from the Volkersen model for bonded joints (chapter three) by using the analytically calculated displacement of the jointed region to specify the stiffness of the joint. This would include not only the layer of solid elements making up the connector pad, but also the displacement due to axial stretching of the top and bottom connected materials.

Finally, nonlinear effects should be considered in future work. It was shown that joints of this type exhibit local plasticity of the fastener holes and the connectors themselves in some cases. The nonlinearities are therefore not simply due to nonlinear elasticity, and need to be considered differently, with the use of nonlinear material models in the regions surrounding the fasteners. Other nonlinearities to consider would be the effects of contact/friction, fastener preload, and interference fits of the connectors.

APPENDIX I: END PIECE DESIGN

As a way of trying to control the bending behavior of the joints due to eccentric loading, a custom set of end piece grips was fabricated. The goal of these pieces was to move the line of action of the load from the centerline of the angles to the centroidal axis of the joint. This would greatly reduce the amount of bending exhibited by the joint and remove such a variable from the connector pad development.

Chapter two explains eccentric loading further than what will be seen in this section, however, the basics should be understood in order to grasp the motivation behind the design of the grips. Figures AI-1 and AI-2 show the cross sectional geometry of the large and small angles, respectively. For more detailed dimensions, Table 4-1 should be consulted. If the angles were to be loaded directly, the load's line of action would pass through the centerline of each respective angle. Not only would this create a force-moment couple, but it would also cause bending behavior as the load is not passing through the centroid of the cross section.

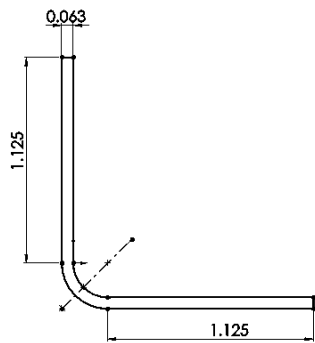


Figure AI-1: Large Angle Cross Sectional Geometry

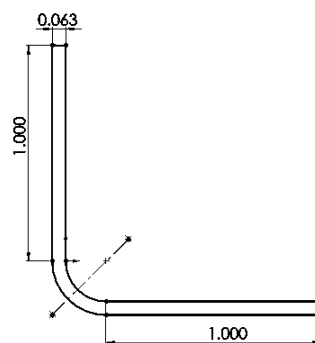


Figure AI-2: Small Angle Cross Sectional Geometry

Figure AI-3 shows the cross section of the center part of any given nested specimen. Note that two holes are included in the section to account for the loss of area due to the fasteners presence when calculating the centroidal location of the cross section. In the figure, the centroid is shown as an asterisk with the distance labeled from the outside of the large angle.

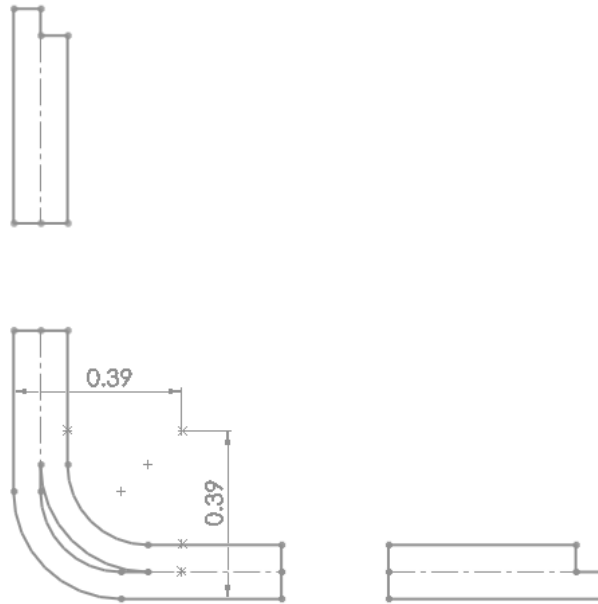


Figure AI-3: Nested Section Centroid Location

Now that the problem and the cross sections are fully defined, the end piece grips are designed. The general design idea was to design end pieces that could be bolted to the specimens to ensure that the central joint would be weaker and be tested appropriately. Therefore, the goal was to over-design the end pieces, as well as the bolts that attach them to the specimen. The maximum size of the end piece cross sections was 1.25 inch diameter circular sections, a limit coming from the size of the grips on the testing machine. As such, the pieces were to be fabricated from 1.25 inch

diameter round steel stock in order to give the maximum area to create an effective cross section, and to give high strength and stiffness compared to the aluminum angles. From the guidelines in chapter two for edge distance and spacing, we can calculate what the theoretical minimum values for edge distance and spacing should be. The value of e (edge distance) was found to be equal to 0.375 inches, or twice the fastener diameter and the value of s (spacing of fasteners) was found to be 0.625 inches. Since the small angle has less space to drill into while still following these guidelines, the design of the small end piece was taken as the critical case. Additionally, since the goal was to over-design these structures, and since the loading was only to be in the axial range of the specimens, the aforementioned guidelines were taken as exactly that; guidelines, and not unbreakable rules for this case.

The cross section design challenges came in the maximization of three attributes: 1) the amount of drillable space on the end pieces, 2) the lining up of the centroid of the end pieces with the centroid of the joint, and 3) the fit of the end pieces in the angled sections. These three challenges were maximized with an iterative process using SolidWorks. Ultimately, three steps were taken to ensure that each of these conditions was met: 1) the design had to be symmetric about the axis bisecting the angle cross sections in order to ensure that the centroid lies on that same line, 2) two opposing corners of the section were filleted to meet the inner radius of the angles such that they fit together correctly, and 3) the dimensions were adjusted in order to both maximize the drillable space and match the centroids with the central combined section centroid. The two sections needed only to be scaled versions of one another.

Since the centroid of the combined section is farther from the surface of the large angle, the large end piece required a larger section. Figure AI-4 shows the final shape of the end piece cross section. For the larger section, the total distance between the left and right (or top and bottom) surfaces is 0.66 inches, and for the smaller section, 0.52 inches.

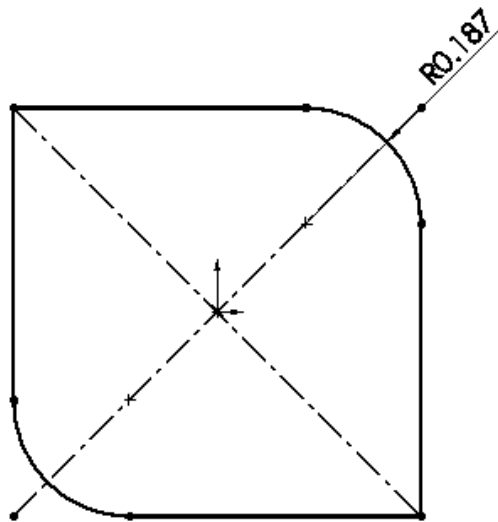


Figure AI-4: End Piece Cross Section

For analysis purposes, it was assumed that the aluminum angles would behave similarly to steel, and the AISC Steel Design Manual was used to aid in the strength calculations. Ten $\frac{1}{4}$ inch Grade 8 steel bolts were used to fasten each end piece to the angles in order to ensure that their strength would exceed that of the central fasteners. Using the values of e and s from above as guidelines, the bolting pattern in Figure AI-5 was calculated. This configuration satisfies all of the requirements except for the edge distance in the transverse direction. Additionally, the requirement for the edge distance from a rolled edge in chapter two was neglected under the assumption that the

bolt spacing would be large enough to prevent premature net section failure under such small loads. The length displayed in the figure was chosen in order to satisfy the spacing and edge distance guidelines.

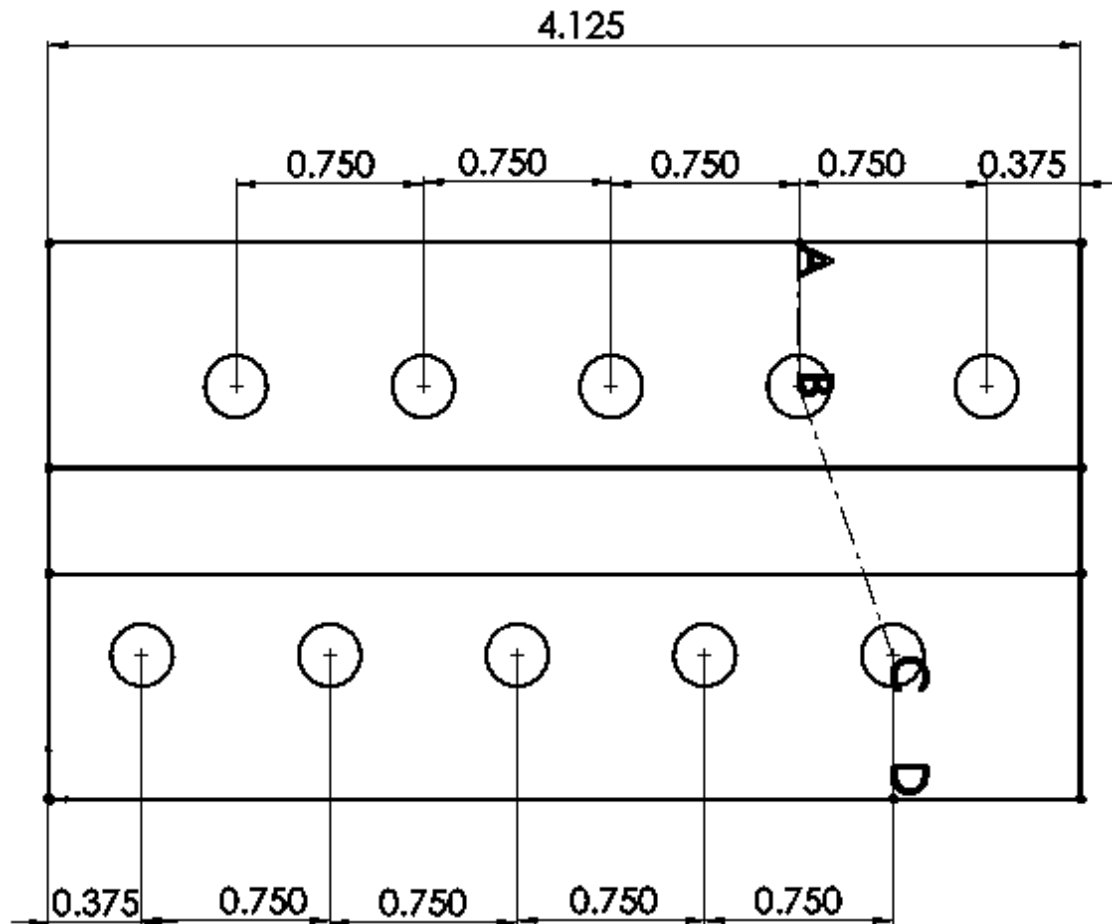


Figure AI-5: End Piece Bolting Configuration (Dimensions in Inches)

In terms of end piece joint strength, three cases need to be considered: 1) bolt shear strength, 2) bolt bearing strength, and 3) net section failure of the angle. Shear strength of the fastener is dependent upon the cross sectional area of the bolt, as well as F_{nv} , a property of the aluminum angles. In Equation AI.1, ϕ is the shear strength reduction factor equal to 0.75 and R_n is the strength of the connection.

$$\phi R_n = 0.75(F_{nv}A_b) \quad \text{AI.1}$$

Bearing strength is considered next and is defined as the load at which the aluminum surrounding the bolt fails due to the bolt pressing against the bolt hole under load. Consequently, the bearing strength is a function of the thickness, t , and ultimate strength, F_u of the angle. In Equation AI.2, L_c is the “clear distance” and is equal to $e - h/2$, where e is defined as the bolt diameter, d plus 1/16 inch.

$$\phi R_n = 0.75(1.2tF_uL_c) \leq 2.4tdF_u \quad \text{AI.2}$$

Finally, net section failure of the aluminum must be considered. This calculation requires a new quantity called d' , which is defined in Equation AI.3. Note that in the equation, s is equal to the spacing in the longitudinal direction, and g is the spacing in the transverse direction (reference Figure AI-5). The variable d' is a quantity which represents the path that a crack would travel through transversely from one edge of the angle or angle to the other. For this analysis, the path $abcd$ (Figure AI-5) yielded the smallest value of d' and is therefore the critical case. The net area of the section, A_n , was then found from subtracting the area of the bolt holes and the area associated with this variable d' from the gross area (taken from SolidWorks). Finally, the strength is found by Equation AI.4:

$$d' = d - \frac{s^2}{4g} \quad \text{AI.3}$$

$$\phi R_n = 0.75(F_u A_n) \quad \text{AI.4}$$

For this configuration, d' is equal to 0.0246, which yields a value of A_n of 0.125 in². This area is plugged into Equation AI.4 to find the load at which net section failure will occur. Table AI-1 contains the strength values for each of the three

checks. It can be seen that net section failure is the critical case, and consequently, the angles had an ultimate load equal to the net section failure load.

Table AI-1: End Piece Strength Checks

Strength Criteria	Ultimate Load [pounds]
Bolt Shear	35300
Bearing	4960
Net Section	4687.5

Since the critical load for the end piece connections is much higher than the load that the specimens were to be tested to (1000 pounds), the end piece design is considered suitable for this application. The final design geometry is shown in Figures AI-6 and AI-7. Again, only the size of the section changes between the larger and smaller angles.

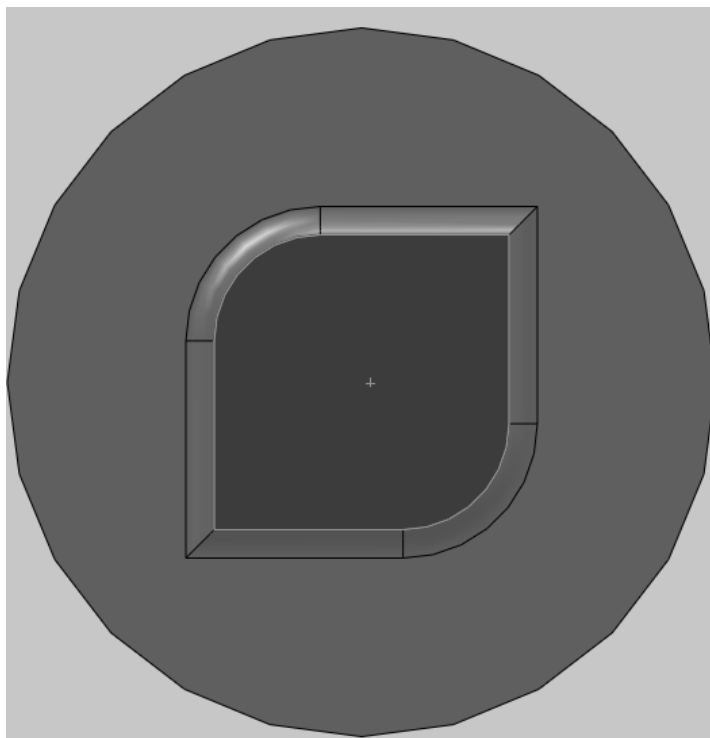


Figure AI-6: End Piece Design Cross Section

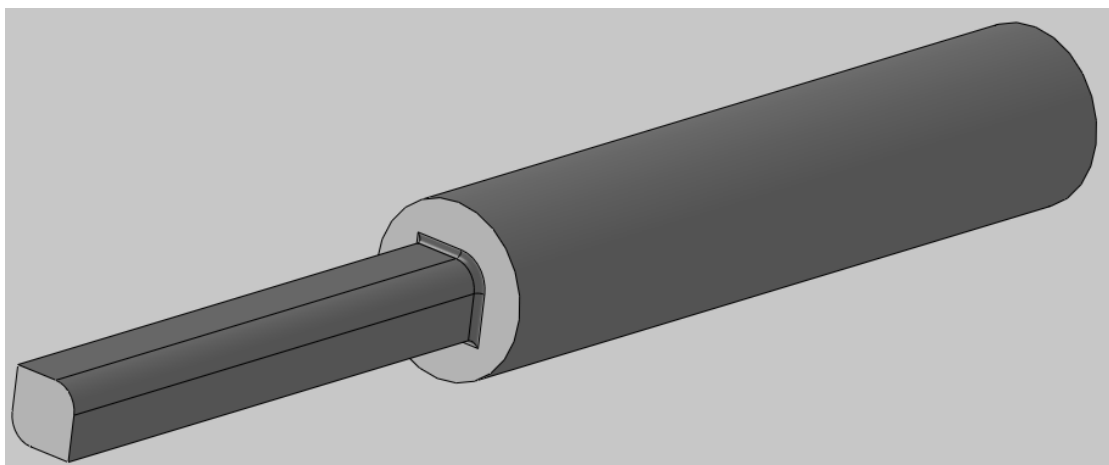


Figure AI-7: End Piece Design Isometric

REFERENCES

- American Institute of Steel Construction. *Steel Construction Manual*. American Institute of Steel Construction, 2006.
- Chen, CS, PA Wawrzynek and AR Ingraffea. "Crack Growth Simulation and Residual Strength Prediction in Airplane Fuselages." Hampton, VA: NASA/CR-1999-209115, 1999.
- Cope, DA and TE Lacy. "Modeling Mechanical Fasteners in Lap Joints for Stress Intensity Determination." *4th Joint DoD/FAA/NASA Conference on Aging Aircraft*. St. Louis, Missouri, 2000.
- Cowper, G. R. "The Shear Coefficient in Timoshenko's Beam Theory." *Journal of Applied Mechanics* 33 (1966): 335-340.
- Cutler, John. *Understanding Aircraft Structures*. Third. Malden: Blackwell Science, 1999.
- Fung, CP and J Smart. "A Experimental and Numerical Analysis of Riveted Single Lap Joints." *Proc. Instn. Mech. Engrs. Part G* (1994): 79-90.
- Hart-Smith, LJ. "Adhesively-bonded single-lap joints." CR 112236. 1973.
- Hi-Shear Corporate. *Hi-Lok Fastening System*. n.d. 2012.
- Kim, Hyonny and KT Kedward. "Joining and Repair of Composite Structures." *ASTM STP 1455*. West Conshohocken, PA: ASTM International, 2004. 127-160.
- Kosmatka, John, et al. "Simplified Methods for Automated Modeling of Riveted/Bolted Connections in Flight Structures." *AIAA 2012 -1450*. Honolulu, 2012.
- Mackerle, J. "Finite Element Analysis of Fastening and Joining: A Bibliography (1990-2002)." *Int. Jour. Pressure Vessels and Piping* 80 (2003): 253-271. Document.
- Marson and Gesipa. *Bolt Products Inc*. n.d. 05 2012.
- Naarayan, S S, D.V.T.G P Kumar and S Chandra. "Implication of unequal rivet load distribution in the failures and damage tolerant design of metal and composite civil aircraft riveted lap joints." *Engineering Failure Analysis* 16.7 (2009): 2255-2273. Document.

- Niu, Michael Chun-Yung. *Airframe Structural Design: Practical Design Information and Data on Aircraft Structures*. Hong Kong: Hong Kong Conmilit, 1999.
- Reithmaier, L.W. *Standard Aircraft Handbook for Mechanics and Technicians*. Sixth. New York: McGraw-Hill, 1999.
- Rosenfeld, S. "Analytical and Experimental Investigation of Bolted Joints." Washington: National Advisory Committee for Aeronautics, 1947.
- Sampson, Stuart. *Simulating Complex Fasteners and Connections – HiLock Fastener*. 10 09 2010. 04 2012.
- Segui, William. *Steel Design*. Toronto, Canada: CL Engineering, 2007.
- Seshadri, B R and J C Newman Jr. *Residual Strength Anaylses of Riveted Lap-Splice Joints*. Washington D.C.: NASA, 2000.
- Swenson, Daniel and Mark James. "FRANC2D/L: A Crack Propagation Simulator for Plane Layered Structures: Version 1.4 User's Guide." 1997.
- Swift, T. "Fracture Analysis of Stiffened Structures." *Damage Tolerance of Metallic Structures: Analysis Methods and Application*. ASTM STP 842, 1984. 69-107.
- Ugural, A C and S K Fenster. "Energy Method for Deflections." *Advanced Strength and Applied Elasticity*. Upper Saddle River, NJ: Prentice-Hall, 1995. 221-223.
- Volkersen, O. *Die Nietkraftverteilung in zugbeanspruchten Nietverbindungen mit*. Luftfahrtforschung, 1938.
- Wikipedia. *Bolted Joint*. n.d. <http://en.wikipedia.org/wiki/Bolted_joint>.
- Younger, John E., Raymond H Rice and Nairne F Ward. *Structural Design of Metal Airplanes*. New York: McGraw-Hill, 1935.