TURN-OF-THE-NUT TIGHTENING OF ANCHOR BOLTS

A Thesis

by

JASON HALBERT RICHARDS

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

May 2004

Major Subject: Civil Engineering

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Major Subject: Civil Engineering

ABSTRACT

Turn-of-the-Nut Tightening of Anchor Bolts. (May 2004)

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Double-nut anchor bolt systems are used in the erection of traffic signal poles, high-mast luminaries, and other highway appurtenances. An absence of a tightening standard for such systems decreases the confidence in their performance under fatigue loading. Past research has shown that a tightening standard should include the development of preload in the anchor bolt sufficient to provide adequate resistance to fatigue failure. Preload should be measured by a turn-of-the-nut method.

Laboratory progressive tightening tests were performed in order to monitor the stress ranges occurring in the bolt at various locations of interest at various degrees of turn-of-the-nut tightness. Tests were performed on six diameters of anchor bolt ranging from 1 to 2-1/4 inches in diameter and two different categories of thread pitch: UNC and 8UN. Plots of stress range versus degree of tightness were developed for each test and evaluated to find the minimum degree of turn-of-the-nut at which stress range inside the nuts dropped below that outside the nuts. This shift was considered to be the principle theoretical indication of adequate performance. A fatigue test which saw failure outside the double-nut connection was set down as the practical indicator of adequate fatigue performance.

The 2 inch 8UN bolt was chosen as the critical specimen due to its overall low generation of preload during tightening tests. Theoretical testing showed that 1/24 turn-

of-the-nut would guarantee sufficient fatigue performance. Two practical fatigue tests of the bolt at that tightness saw one positive and one negative failure.

After actual lab tests, finite element modeling was used to investigate the behavior of the bolt. It was found that performance did not see improvement until 1/12 turn-of-the-nut.

After all results were considered, a standard of 1/6 turn-of-the-nut or refusal of tightening by specified methods was recommended, provided a minimum of 1/12 turn-of-the-nut was achieved. This value allows for ease of measurement, sufficient tightness, degree of safety, and has been shown in past testing not to cause failure through over-tightening. However, tightening to only 1/12 turn-of-the-nut still provided adequate performance.

DEDICATION

I offer this thesis up for the glory of Jesus Christ my savior. I further dedicate this thesis to my parents, Hal and Christi, to my brothers, Travis and Chad, and to my fiancée, Susan. Your love and support have never failed me.

ACKNOWLEDGEMENTS

I would like to thank Dr. Peter Keating for his guidance and patience, Dr. Ray James for his support, Dr. Harry Hogan for his encouragement and example, and Lee Christian and Scott Hubley for all of their hard work. Thanks also to Andrew Fawcett, Jeff Perry, and Matthew Potter of TMRF and Rick Seneff of the TTI Machine Shop. Credit is given to the Texas Department of Transportation for funding this research, and to TxDOT engineers Mike Smith and Jim Yang for their involvement in and support of this project.

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1 INTRODUCTION

In recent years, the Texas Department of Transportation (TxDOT) has sponsored research into the fatigue performance of anchor bolts. This research was undertaken for two reasons: one, a lack of tightening standards for double-nut connections used in roadside structures and two, the failure of cantilevered highway signs in various places across the country, particularly in Michigan. These two issues gave rise to questions about the practices used in Texas for assembling highway-side structures such as traffic signal poles, high-mast luminaries, and other similar highway appurtenances.

1.1 Definitions

A number of terms require definition, as an understanding of them is required for comprehension of the thesis. The following definitions were taken from the TxDOT *Manual of Testing Procedures*.

- Double-nut anchor bolt system – An anchor bolt with two nuts that sandwich the structure's base-plate. The bottom nut is positioned under the base plate to level, support, and provide the reaction for the force applied by tightening the top nut located above the base plate.
- Impact tightening – The tightening of nuts with a box end 'slug' or 'knocker' wrench and an approximate 16 lb sledgehammer. The wrench, matching the size of the nut to be tightened, is driven with the sledgehammer to rotate the nut. Wrenches used in testing associated with the current study are shown in Figure 1.1.

This thesis follows the style of the *Journal of Structural Engineering*.

Figure 1.1 – Slug wrenches

- Snug tight – The condition when the nut is in full contact with the base-plate. It may be assumed that the full effort of a workman on a 12 inch wrench results in a snug tight condition.
- Turn-of-the-nut-method – The tightening of a nut to snug tight condition then establishing reference positions by marking one flat on the nut with a corresponding reference mark on the base-plate at each bolt. The nut is then turned to the prescribed rotation from the referenced snug tight position. A nut marked with 1/24 turn-of-the-nut increments for reference is shown in Figure 1.2.

Figure 1.2 – Nut marked for turn reference

1.2 Problem history

To understand the purposes behind, and application of, the current study, it is necessary to understand the conditions that brought it about and the research leading up to it. The current study is a continuation of research and studies beginning in 1990. The failure of two cantilever highway signs in January and February of that year in Michigan prompted research into fatigue failure of anchor bolts in Michigan as well as across the country. Fatigue was found responsible in those failures, and issues were raised regarding proper tightening of the anchor bolt connections, as tightening has been determined to affect performance under cyclic loading. The chief concerns of the Texas Department of Transportation (TxDOT) were stated in a report published in 1997 by the Texas Transportation Institute (TTI): “How tight do the anchor bolt nuts need to be?

What techniques and procedures are best suited for tightening? How can tightness be inspected?" (James et al. 1997).

1.3 Fatigue

The effects of fatigue in any material are very difficult to predict. Unlike most material responses, small imperfections and geometry most often influence fatigue failure. Typically major factors such as material yield strength or bolt diameter have little, if any, impact on fatigue life of anchor bolts (Frank 1980). At the very least, such issues that are normally principal predictors of structural behavior play a much smaller role in determining fatigue performance.

Fatigue failure consists of two components. One is the formation or initiation of cracks. This typically occurs at areas of stress concentration. Such concentrations always exist to some degree, and the magnitude determines how influential it is in affecting fatigue performance. These are most often found at imperfections in the material or at geometric stress risers such as holes or edges. In the case of anchor bolts, the critical stress concentration is most likely to be found at the initial interface of the threads of the bolt and nut. However, in real situations, flaws such as thread damage, machined holes, or cuts in the bolt can control the formation of cracks that lead to failure.

The second issue, and that of greater importance, is that of stress range. It is the loading and unloading of the bolt in a cyclical fashion that mitigates and propagates the cracks that can lead to the eventual failure of the structure. In the case of the anchor bolts examined in this study, such loading is primarily caused by wind. Even in the absence of significant stress concentrations, fatigue can cause failure in materials (Ugural and Fenster 1995).

Wind-induced fatigue loading is typically seen in conjunction with other forms of loading. The primary elements in anchor bolt loading include the fatigue combined with a constant tensile axial load as a result of the dead load. The axial load affects the mean stress level seen during fatigue loading. These components are noted in Figure 1.3 below. Other loads may include compressive axial load, bending, torsion, or other types depending on the use and configuration of the structure.

It has been previously mentioned that stress range is the most critical aspect of fatigue. Even at a mean stress of zero, the application of stress range can cause failure. This illustrates the importance of stress range as greater than that of mean stress.

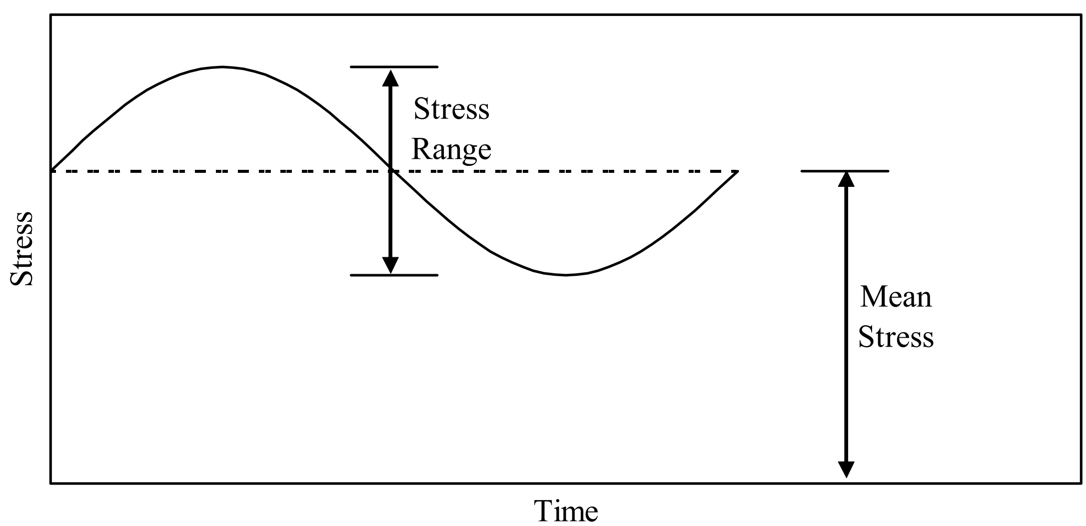

Figure 1.3 – Components of fatigue

1.4 Typical anchor bolt setup

The anchor bolts examined in the current study and in much of the research concerning double-nut fixtures are used for affixing the base of roadside illumination structures or cantilevered highway signs to concrete foundations. A typical assembly of anchor bolts used by TxDOT is shown in Figure 1.4.

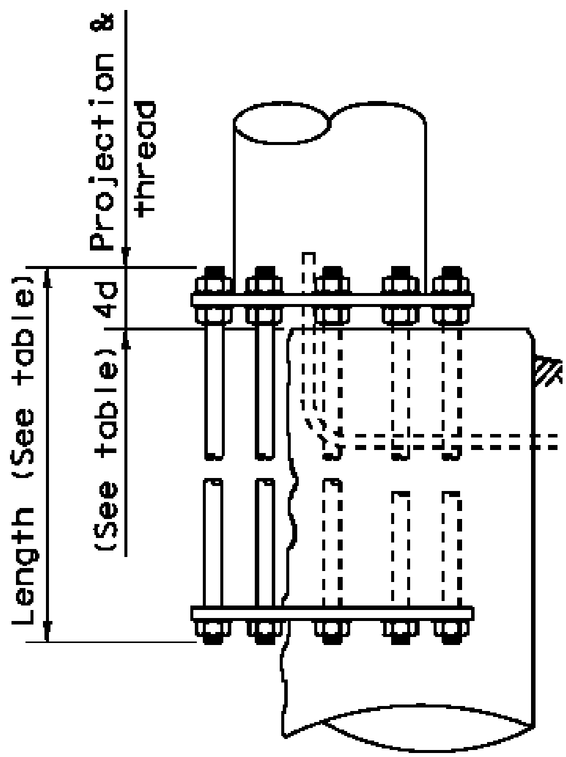

Figure 1.4 – Anchor bolt assembly

The double-nut assembly of an individual anchor bolt can be seem more clearly in a schematic illustration of a test specimen, Figure 1.5. The bottom nut is often called the leveling nut, since it is used to level the base-plate and thus ensure the vertical erection of the structure. The top nut is the nut that is tightened, securing the base-plate against the bottom nut and preventing uplift or overturning of the structure. Washers between the nuts and the base-plate better distribute the bearing forces and facilitate the turning of the nut.

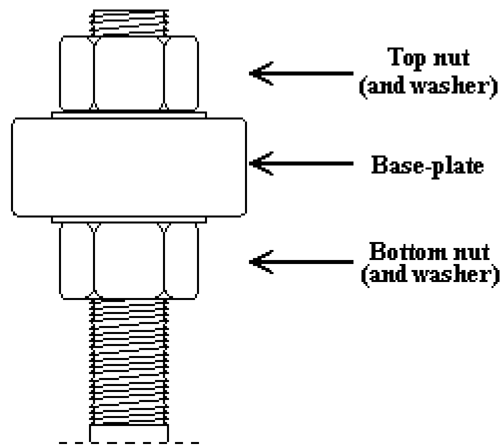

Figure 1.5 – Individual assembled anchor bolt

In actual installations, the base-plate is a large circular or rectangular plate with holes around it to allow for the anchor bolts. A typical plan view of a base-plate is given in Figure 1.6. For the current study, the base-plate was a simple ring used as a spacer, necessary to the development of preload over the length of the bolt between the nuts. Maximum allowable TxDOT details were used as a basis for the plates used in testing. Further details on base-plate dimensions can be found in Section 2.

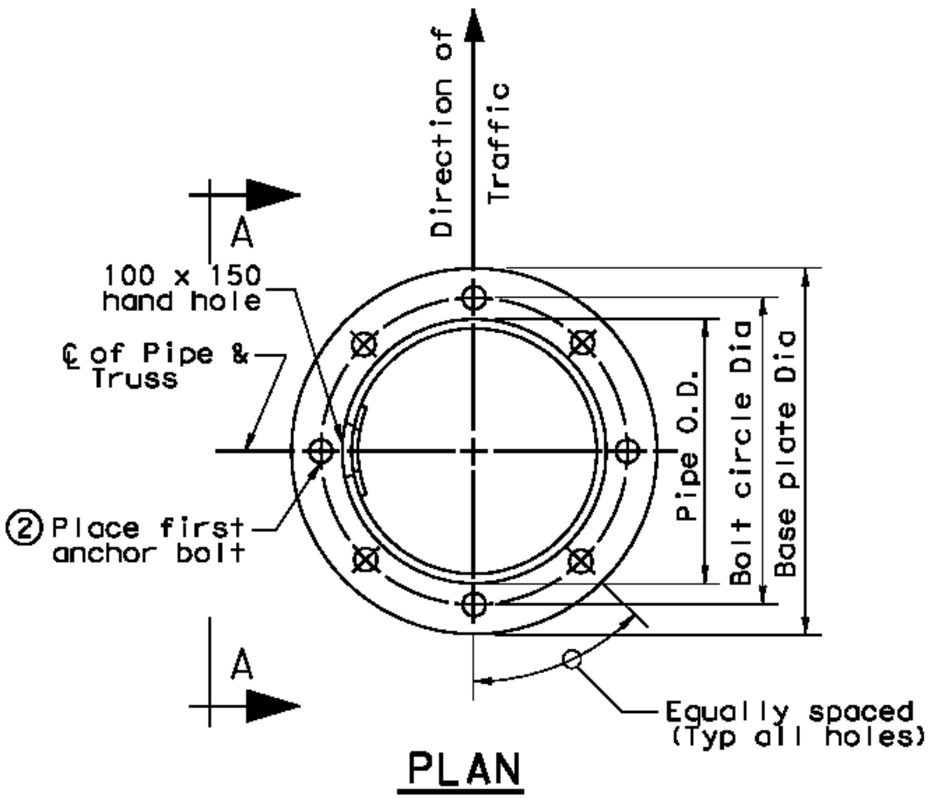

Figure 1.6 – Typical base-plate detail

1.5 Basic preload theory

Understanding the usefulness and advantage of preloading, and why such a condition affects fatigue performance, is critical to understanding the tests that were performed for the current study. When turning the top nut past a snug tight condition, the resulting preload shields the double-nut connection from external loading. Any applied force, such as that due to the uplift caused by wind load, must first overcome the initial tensile preload to have any affect on the bolt length within the double-nut connection. This reduces the stress range experienced within the double-nut connection and causes the likely location of failure to shift from below the top nut to below the bottom nut, as has been shown by Frank (1980), Van Dien et al. (1996), James et al. (1997), and Keating et al. (2004). The end effect is that the addition of static tensile load along a given length of the bolt reduces the effect of cyclic load along that same region, preventing premature failure.

1.6 Adequate fatigue performance

Bolts without sufficient preload are determined to perform inadequately under the cyclic loading. The high stress range located at the interface of the base-plate and the top nut, coupled with the high levels of stress seen there, leads to premature fatigue failure. The measurement of these stress ranges and stress concentrations are discussed in detail in Sections 3 and 4.

The fatigue behavior of a sufficiently-preloaded double-nut connection is considered to be adequate performance. That is, adequate performance of an anchor bolt can be measured by the reduction in stress range between the nuts to a level below that seen by the rest of the bolt. The best case for fatigue failure is to have it occur below the double-nut assembly. This case is the minimum that is expected from anchor bolts in the field. Again, the measurement of stress ranges and concentrations is discussed in Sections 3 and 4.

1.7 Past research, nationwide

Fundamentals for double-nut connection research include studies dating back to 1980. In that year, Frank (1980) performed a general study of anchor bolt fatigue in his article, “Fatigue Strength of Anchor Bolts”, and came to a number of important conclusions. He concluded that steel grade, thread size, bar diameter, thread forming method, and galvanizing do not significantly influence the fatigue strength of anchor bolts. Diameters tested ranged from 1-3/8 to 2 inches, and thread pitch ranged from 4-1/2 to 8 threads per inch. Steel ranged from 26.7 to 171 ksi yield strength. He also came to an important conclusion about preload. In his tests, Frank found that tightening the double-nut connection 1/3 of a turn-of-the-nut beyond snug tight significantly improved fatigue life and moved the failure location to the outside of the connection. Snug tight was defined by Frank as 200 ft-lbs of applied torque.

In the 1996 article, “Fatigue Testing of Anchor Bolts”, Van Dien et al. (1996) concurred with Frank’s findings, including the value of preloading the anchor bolts. The migration of failure location to the outside of the fixture was noted in specimens turned to 1/3 turn-of-the-nut beyond snug tight. The snug tight definition from Frank’s earlier study was used. This article raised concerns about inspection of the anchor bolts for cracks “[d]ue to the inherent configuration of bolted connections.” This statement highlights one of the great advantages of applying turn-of-the-nut preload to anchor bolts: the facilitation of inspection.

After the failures that prompted much of the research into anchor bolts since 1990, in-depth inspection and testing of anchor bolts took place in Michigan. In a report from the Michigan Department of Transportation (MDOT) in 1990, “Action Plan for Cantilever Sign Problem” and the final report in 1992, inspection again became a major issue. MDOT found that the methods then used for inspection in most states were antiquated, inadequate, or even non-existent. After inspection of Michigan's own highway structures, an anchor bolt in one cantilever sign pole was found completely broken inside the connection, at the first fully engaged thread of the top nut, a failure that would have gone unnoticed save for the intense and comprehensive inspections after the accidents. Again, inspection and failure location came to the forefront of the issues.

Publications such as these clearly demonstrate the benefits of turn-of-the-nut preload. This leads into the question of how to best introduce preload and how much preload to apply to achieve adequate fatigue performance.

1.8 Past research, TxDOT

The question of required preload for double-nut anchor bolt fixtures has been addressed in several publications associated with the Texas Transportation Institute and Texas A&M University, which are well-represented by Abraham (1997). Abraham cited optimum levels of preload with respect to desired fatigue characteristics as shown in Table 1.1.

Table 1.1 – Optimum preload as determined by Abraham

Nominal Preload Stress Factor (maximum axial stress multiplier)	Tensile Stress Range reduced by:	Thread Roots in:	Failure Location
0 to 33%	0 to 25%	Tension	Below top nut
33% to 200%	25 to 40%	Tension	Below bottom nut
200% +	40%	Compression	Below bottom nut

Methods by which preload was generated varied in the study. Abraham, that using a cheater pipe and wrench, was unable to consistently generate the turn-of-the-nut desired. Also, the torque required to turn a nut to a certain rotation varied significantly from bolt to bolt, making a torque wrench an unreliable method. The best method was found to be using a 16 lb sledgehammer and knocker wrench to achieve the specified turn-of-the-nut.

Hodge (1996) studied other aspects of tightening and fatigue as a part of his work on the fatigue of high-mast luminare anchor bolts. He studied the effects of snug tightening and preloading anchor bolts and misalignment. Preloading of anchor bolts was again found to cause a shift in probable failure location, this time through finite element modeling. When subjected to loading, stress concentrations and stress ranges were observed on the model, which determined the probable location of failure. This analytical confirmation of the behavior reaffirms experimental results found by Frank, Van Dian et al., and other research.

The work of Abraham and Hodge was summarized in TTI Report 1472-1F entitled “Tightening Procedures for Large-Diameter Anchor Bolts”, where results were reported and new procedures and standards for anchor bolts recommended. The connection between preload, stress range, and failure location was clearly stated in this report, “The location of the maximum axial stress range is [at the outside of the connection], which explains why the mode of failure shifts to this region. (James et al. 1997).”

A continuing study, "Supplemental Laboratory Testing to Extend Scope of Proposed Standard Method of Tightening of Anchor Bolts", was performed by TTI for TxDOT looking more closely at issues of tightening and fatigue for a much larger range of issues including lubricants, a wide range of bolt diameters, thread pitch, material grade, and reuse of bolts. Results confirmed the findings of Frank (1980) and Van Dien et al. (1996) regarding improved performance of anchor bolts in fatigue when preloaded by a turn-of-the-nut standard. The study also showed the previously observed migration in failure location when so preloaded. Electrically conducting lubricant was found to be the lubricant of choice, and 8UN bolts of 2-1/4 diameter and larger were recommended for exclusion from TxDOT specifications due to fatigue issues. Bolt reuse, even after yielding in preload, was shown to not be a significant factor in fatigue life.

In dealing with a large variety of bolt diameters, ranging from 1 inch to 3 inches, issues arose with previous research. Research by Frank (1980) and Van Dien et al. (1996) discussed migration of failure location in terms of 1/3 turn-of-the-nut beyond snug tight. It was found in Keating et al. (2004) that for some coarse thread pitches and large diameters, or a combination of the two, achieving 1/3 turn-of-the-nut was not possible through practiced methods. Most fatigue tests were performed in the new study with bolts tightened only 1/6 turn-of-the-nut, sometimes less, and still achieved the favorable behavior of preloading found in past research. This led to questions regarding the precise amount of preload required to achieve enhanced performance under fatigue. Eventually an extension of this research developed into the testing performed for the current study.

1.9 Continuing research

By piecing together the current body of knowledge on preload in anchor bolts, the current study was performed to directly relate the issues of interest: turn-of-the-nut tightening and adequate fatigue performance. Previous tests that demonstrated such improvement have most often been performed at 1/3 turn-of-the-nut beyond snug tight, or recently at only 1/6 turn-of-the-nut or less. The next logical question is, "At what turn-of-the-nut does one see the advantage of preload?" This is a valid question for each bolt diameter and thread pitch used by TxDOT.

1.10 Importance of failure location migration

The migration of failure location to a point outside the double-nut fixture is important for purposes of inspection, as frequently noted in past research and reports. In an anchor bolt without preload, failure typically occurs at Location 1 as noted in Figure 1.7. A sufficiently preloaded bolt sees the failure location migrate to Location 2. As can be seen in Figure 1.4 above, Location 2 is visible and inspectable as opposed to Location 1, which is within the double-nut assembly. If failure occurs within the fixture at Location 1, inspection requires the disassembly of the structure or the use of costly equipment, limiting the ability and frequency of the Department of Transportation inspections, and increasing their cost. While cracking at Location 2 may still be difficult to detect, at the very least failure outside of the double-nut assembly is easier to detect and allows for the possibility that anchor bolt failures may be observed and the bolt replaced instead of going unnoticed as was the case in Michigan as mentioned above.

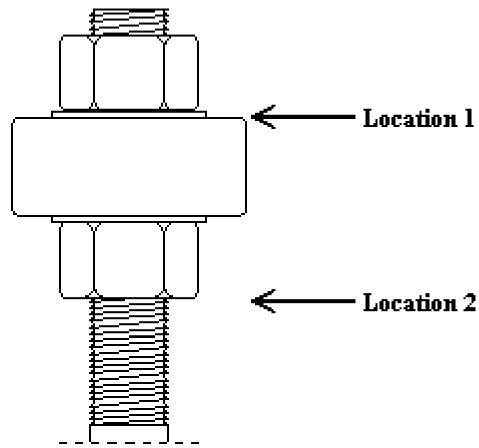

Figure 1.7 – Bolt failure locations

The most important aspect of the migration of failure location is as an indicator of desirable behavior. The shift in failure location to outside the double-nut connection implies the existence of significant preload in the tightened section of the anchor bolt and a reduction in stress range in the interior of the double-nut assembly, which have been shown to extend fatigue life within the tightened section. The relationship between preload, stress range, and failure location is discussed extensively in Sections 3 and 4.

1.11 Problem summary

The problem may be summarized by the following question: "What turn-of-the-nut generates enough preload to lower stress ranges to a level that moves the likely failure location move from Location 1 to Location 2 on the double-nut anchor bolt?" This question is valid for each diameter used by TxDOT in roadside applications. This study evaluates six different bolt diameters as shown in Table 1.2, and performs tests to determine the answer to the above question for each case.

Table 1.2 – Test matrix for progressive tightening tests

	Number of Specimens Tested	
Bolt Diameter (in.)	A193 Gr. B7	
	UNC	8UN
1	-	1
1-1/4	1	1
1-1/2	1	1
1-3/4	1	1
2	1	1
2-1/4	1	1

1.12 Objective and criteria

The final objective of the current study is a minimum turn-of-the-nut tightening standard that demonstrates the improved fatigue characteristics shown in previous research utilizing 1/3 or 1/6 turn-of-the-nut. The comprehensive testing of all diameters and thread pitches of the bolt samples to failure would be unrealistic. To substitute, two criteria for the effective improvement of fatigue characteristics have been established. The principal and theoretical criterion is the observed reduction in stress range experienced between the nuts of the double-nut connection. The practical criterion is the witnessed failure of an assembly at a point outside the connection when tightened to the specified minimum turn-of-the-nut. This location is labeled as Location 2 in Figure 1.7. The reduction in stress range and the shift in failure location are both seen as indicators of sufficient preload to ensure adequate fatigue performance.

2 METHODS

2.1 Progressive tightening test

Progressive tightening is the term used to describe the tests performed to gather data on stress ranges at incremental levels of turn-of-the-nut tightness. Bolt specimens were subjected to a series of short fatigue tests at increasing degrees of turn-of-the-nut as detailed below.

2.1.1 Bolt selection

Test specimens were chosen to fit a number of criteria and to represent the range of available options for anchor bolts in the field. The following variables were considered in the selection of specimens:

- Bolt diameter
- Thread pitch
- Material grade
- Bolt and thread length
- Galvanizing
- Availability

2.1.1.1 Bolt diameter

Bolt diameters were chosen based on those specified by the Texas Department of Transportation (TxDOT) for the installation of highway signage. From this list were taken all that were capable of testing by the fatigue fixture available in the lab. Diameters ranged from 1 inch to 2-1/4 inch by 1/4 inch increments.

2.1.1.2 Thread pitch

TxDOT utilizes two categories of thread pitch. The first of these is Unified National Coarse, more commonly known as UNC. UNC bolts vary in number of threads per inch with diameter of the bolt. Unified National 8 is more often known as 8UN and has 8 threads per inch for all bolt diameters. For the 1 inch bolt, UNC and 8UN have the same thread pitch. A comparison of both thread types is found in Table 2.1. For a full range of data, each thread pitch was tested for each diameter.

Table 2.1 – Bolt thread pitches by diameter

Bolt Diameter (inches)	UNC (threads per inch)	8UN (threads per inch)
1	8	8
1-1/4	7	8
1-1/2	6	8
1-3/4	5	8
2	4-1/2	8
2-1/4	4-1/2	8

2.1.1.3 Material grade

Only high-strength A193 Gr. B7 steel was used in the testing. Grade B7 steel has a nominal yield strength of 105 ksi. This level of yield strength allows elastic stress data to be collected to high degrees of turn-of-the-nut. Lower grades of steel are allowed by TxDOT, but were not deemed necessary to test because steel yield strength has been found to be unimportant to fatigue performance in a number of studies (Frank 1980, Van Dien et al. 1996, Keating et al. 2004).

2.1.1.4 Bolt and thread length

All bolts for testing were required to be sufficiently long to accommodate the necessary strain gaging and thread length, and of appropriate length to fit into the testing apparatus. Thread length was governed by the need to fit on both nuts, washers, and the base-plate on the double-nut side. While the opposite end only required space for one nut, washer, and the base-plate, it was cut with the same length of thread as the double-nut side. This was done to simplify assembly and offer redundancy should machining or other events damage the double-nut side and make it unsuitable for testing.

2.1.1.5 Galvanizing

In the field, all bolts, nuts, and washers are galvanized to protect them from corrosion due to the elements. In the test, all new specimens acquired were without any galvanizing on the bolts, nuts, or washers. The presence of galvanizing is unimportant for fatigue applications (Frank 1980).

2.1.1.6 Availability

As the progressive tightening test was a continuation study of research funded by TxDOT, efforts were made to save time and money by utilizing existing specimens left over from previous research. Substitutions were only made in cases known to be acceptable based on past research. Substitutions were made in the following cases:

- All tests carried out for development of the procedure were 1-3/4 inch specimens with galvanizing and non-standard bolt and threaded lengths. These were developmental specimens not included in the final results.
- The 2-1/4 inch 8UN specimen was galvanized and fully threaded.
- The 1-3/4 inch UNC specimen was reused after modification from developmental tests. It was galvanized and had non-standard bolt and thread lengths.
- The 1-3/4 inch 8UN specimen was previously used in a tightening test, without fatiguing. It was galvanized and had non-standard bolt length and was fully threaded.

Consequently, a group of specimens was put together as shown in Table 2.2. All bolt specimens were of steel grade A193 Gr. B7 with a nominal yield strength of 105 ksi.

Table 2.2 – Progressive tightening test specimens

Bolt Diameter (in.)	Thread type	Thread pitch	Galvanizing	Bolt Length (in.)	Thread Length 1 (in.)	Thread Length 2 (in.)
1	8UN	8	N	22	6	6
1-1/4	UNC	7	N	22	8	8
1-1/4	8UN	8	N	22	8	8
1-1/2	UNC	6	N	22	8	10
1-1/2	8UN	8	N	22	10	10
1-3/4	UNC	5	Y	28	11	3
1-3/4	8UN	8	Y	22	Full	Full
2	UNC	4-1/2	N	28	11	10.5
2	8UN	8	N	28	10	10
2-1/4	UNC	4-1/2	N	28	12	10
2-1/4	8UN	8	Y	28	Full	Full

2.1.1.7 Base-plate selection

The base-plates involved in the assembly of a test specimen were adapted from TxDOT specifications and Keating et al. (2004). Each base-plate was circular with a hole machined through the middle. From each of these references was gathered the following basis for the test plan:

Table 2.3 – Original specified base-plate dimensions

Bolt Diameter (in.)	Base-plate Inner Diameter (in.)	Base-plate Outer Diameter (in.)	Base-plate Thickness (in.)
1	1.25	4.50	1.25
1-1/4	1.50	4.50	1.50
1-1/2	1.75	4.50	1.88
1-3/4	2.06	4.50	2.25
2	2.31	8.19	2.38
2-1/4	2.56	8.19	2.75

The original base-plate dimensions above in Table 2.3 were designed for use in two separate test setups as detailed in Keating et al. (2004). Base-plate thicknesses were taken from the maximum allowable by TxDOT specifications. They were also used to collect tightening data as discussed later in this section. The above base-plates were readily available for use in testing and were adapted for use in the progressive tightening tests. The following changes were made:

- The large plate for the 2 inch bolt specimen was substituted with the plate from the 1-3/4 inch bolt specimen. The lower tolerance, while causing problems in field setup, is workable in a laboratory situation. This increase in thickness for the 1-3/4 inch bolt specimen plate is conservative since a greater plate thickness results in less preload for the equivalent turn-of-the-nut.

- The large plate for the 2-1/4 inch specimen was machined to a diameter acceptable for use in the smaller test fixture. Calculation of rupture load limits confirmed the modification as acceptable.

The plate dimensions used for progressive tightening are listed in Table 2.4.

Table 2.4 – Final base-plate dimensions

Bolt Diameter (in.)	Base-plate Inner Diameter (in.)	Base-plate Outer Diameter (in.)	Base-plate Thickness (in.)
1	1.25	4.50	1.25
1-1/4	1.50	4.50	1.50
1-1/2	1.75	4.50	1.88
1-3/4	2.06	4.50	2.25
2	2.06	4.50	2.25
2-1/4	2.56	4.50	2.75

2.1.2 Specimen preparation

2.1.2.1 Machining

To allow for the application of strain gages to the surface of the bolts being tested, it was necessary to machine longitudinal grooves along the bolt to provide a workable surface. Groove dimensions varied according to bolt diameter to accommodate the various sizes of the strain gages and to allow sufficient room for wires to be run without interfering with the threading of nuts onto the bolts. Grooves were cut to a minimum of two inches beyond the length of thread to prevent stress concentrations near strain gage readings. The actual dimensions of the grooves are shown in Table 2.5.

Machining often left burrs or other imperfections on the threads that hindered the ability of a nut to be turned onto the bolt by hand. In such cases, threads were improved and burrs removed with a small metal file.

Table 2.5 – Groove dimensions

Bolt Dimensions		Groove Dimensions (in.)			
Diameter (in.)	Thread Pitch (threads/in.)	Left		Right	
		Width	Depth	Width	Depth
1	8	0.250	0.281	0.250	0.281
1-1/4	7	0.313	0.344	0.313	0.406
1-1/4	8	0.313	0.375	0.313	0.375
1-1/2	6	0.375	0.375	0.375	0.372
1-1/2	8	0.313	0.369	0.313	0.375
1-3/4	5	0.438	0.328	0.438	0.313
1-3/4	8	0.438	0.359	0.438	0.391
2	4-1/2	0.500	0.406	0.500	0.406
2	8	0.500	0.375	0.500	0.438
2-1/4	4-1/2	0.625	0.438	0.625	0.469
2-1/4	8	0.563	0.250	0.563	0.250

2.1.2.2 Strain gaging

Specimens were prepared for strain gaging after machining by cleaning grooves with methanol and a clean rag or cotton swabs until running a cotton swab over the groove produced no dirt or oil. A clean specimen was vital to achieving good readings with the strain gage and for good contact for the epoxy used to affix the gages to the bolt.

The gages used for data collection were Micro-Measurements CEA-06-250UW-120 with 0.25 inch gage length. Size varied with the diameter of the bolt and the allowance of the machined grooves in which the gages were placed. Strain gages were affixed centered at the locations and measurements shown in Figure 2.1 and Table 2.6, respectively.

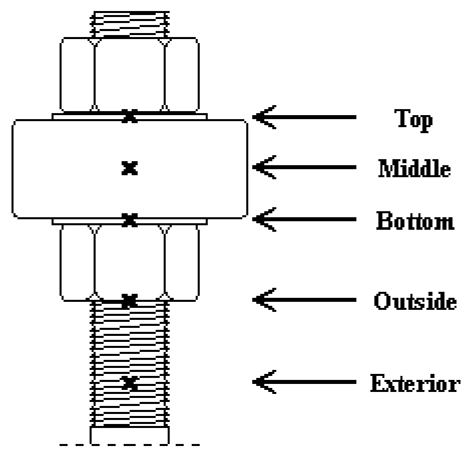

Figure 2.1 – Gage location

Table 2.6 – Gage placement

	Gage placement from end of bolt (in.)				
Bolt Diameter (in.)	Top	Middle	Bottom	Outside	Exterior
1	1.63	2.25	2.88	4.00	6.000
1-1/4	1.88	2.63	3.38	4.75	6.75
1-1/2	2.13	3.00	4.00	5.63	7.63
1-3/4	2.19	3.31	4.44	6.13	8.13
2	2.63	3.80	5.00	7.13	9.13
2-1/4	3.00	4.33	5.67	8.13	10.13

2.1.3 Experimental procedure

After a specimen was gaged and wired, it was assembled without a bottom nut so that a fatigue test could be performed in the absence of preload. This configuration is shown in Figure 2.2. The nut and base-plate on the double-nut side was carefully aligned with the appropriate strain gages. The specimen was then cyclically loaded to determine the stress ranges at each gage location.

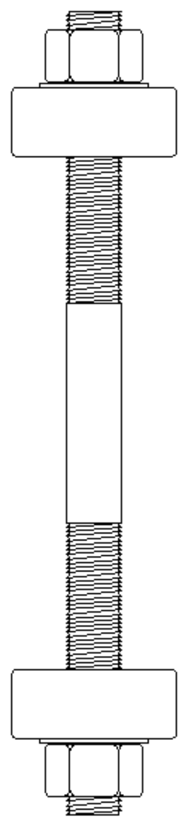

Figure 2.2 – Single nut configuration for progressive tightening

The complete specimen, with the bottom nut and washer, was then put through the assembly process that follows, as developed in Section 2 of Keating et al (2004). To begin, the nuts, washers, and base-plates were removed, leaving only the bolt itself. The bolt was clamped in a 500 kip MTS test machine to simulate the fixed condition experienced in the field due to the anchor bolt being implanted into the concrete base. A bolt can be seen clamped in the MTS in Figure 2.3. The clamping force was enough to keep the bolt from rotating during the tightening process, typically between 75 and 125 kips depending on the bolt diameter. A shim was placed under the bolt to allow the base-plate and nuts to be added, and a plate was placed on top of the non-threaded portion of the bolt to help distribute the clamping force and protect the threads from damage.

Figure 2.3 – Clamped bolt ready for assembly

The bottom nut, washer, and base-plate were installed on the specimen and aligned with the strain gages. A preliminary zero strain gage reading was taken. The top nut and washer were then installed and the overall placement of the strain gages was confirmed. The top nut was tightened to snug tight, as previously defined, and a strain gage reading taken.

The top nut was then removed and another zero reading taken. At this point, the threads of the bolt and nut, and the contact faces of the washer were lubricated with an electrically conducting lubricant and then reassembled and once again brought to a snug tight condition and a strain reading recorded. This can be seen in Figure 2.4, Figure 2.5, and Figure 2.6. The final bolt setup is seen in Figure 2.7.

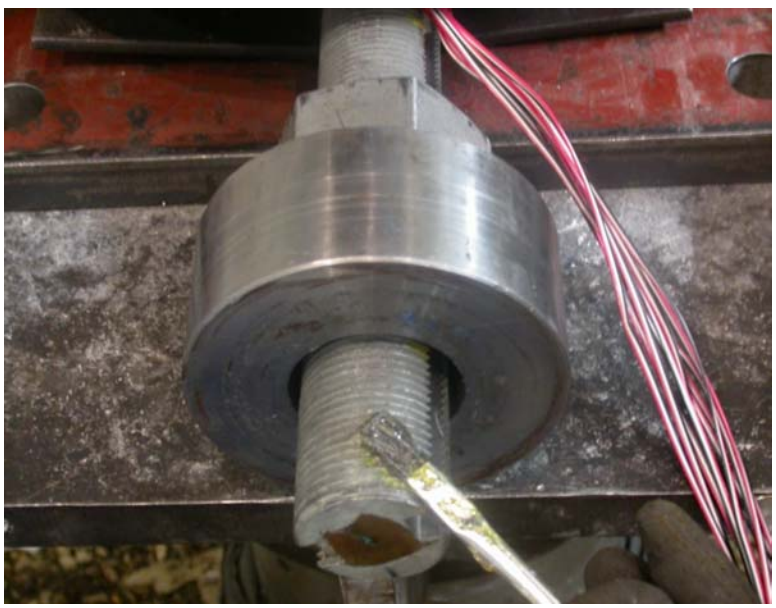

Figure 2.4 – Lubricating bolt threads

Figure 2.5 – Lubricating nut threads

Figure 2.6 – Lubricating washer faces

Figure 2.7 – Fully assembled bolt specimen

Once at that final snug tight condition, the specimen was installed in the 100 kip test machine as seen in Figure 2.8, cycled for a minimum of three seconds while recordings of strain were taken. Maximum and minimum fatigue loads were calibrated so that the minimum stress was 1 ksi and the maximum stress was 15 ksi.

Figure 2.8 – Assembled bolt fatiguing in 100 kip MTS machine

The specimen was then again fixed in the 500 kip MTS machine and tightened 1/24 of a turn using a 16-lb sledgehammer and box-end slug wrench as shown in Figure 2.9. The 1/24 increment was defined by marked intervals on the nut as seen in Figure 1.2. This technique was identified as the most reliable form of tightening by Abraham (1997) and further specified in Keating et al. (2004). Once tightened, a reading of preload was recorded, then the specimen was reinstalled in the 100 kip MTS test machine for fatiguing at the new turn-of-the-nut tightness.

Figure 2.9 – Tightening with a slug wrench and sledgehammer

This procedure of tightening and fatiguing was carried out until either a minimum of 1/6 of a turn-of-the-nut was achieved or the connection refused further tightening. Once the final condition was reached, the nuts were loosened utilizing the slug wrench and sledgehammer and a final zero reading taken.

2.1.4 Data collection and assembly

2.1.4.1 Preload data

Throughout the tests, preload data was taken by strain gages as described above in the experimental procedures. The gage noted as Middle in Figure 2.1 was used to record preload. Strain gage data was converted to stress using Hooke's law in a spreadsheet. Since data was taken at two different gages, on the same diameter of the bolt, two stress values were measured, which were averaged to get a mean preload value for that given turn-of-the-nut. This procedure was repeated for each turn-of-the-nut tested.

Due to machining of the bolts required for stain gaging, some adjustment of these values was required. Both the original and adjusted values are found in the tables in the Results section. An explanation of the methods used for adjusting the data is found in Section 4.

2.1.4.2 Fatigue data

Fatigue data was collected at a frequency of approximately 10 Hz. From the collected data, the first one-third of a second, or approximately three cycles, was excluded to allow for any settling of the specimen. The five maximum and five minimum values during a representative sample duration were recorded and averaged for each strain gage. The difference of these was found to gain a representative strain range for the test. The strain values were converted to stresses in the same manner as the preload data, and the pairs of gages on each side of the bolt were averaged to get a mean stress value for that location. This was done for each turn-of-the-nut.

The resulting data was plotted as stress range versus degree of tightening, with a separate data line for each gage location. The charts discussed in Section 3 are the result of this data analysis. All stress range data for every turn-of-the-nut of every specimen can be found summarized in Appendix A.

2.2 Tightening tests

In addition to the preload generation data gathered from the tightening portion of the progressive tests detailed above, additional bolts of various diameters and thread pitches were subjected to incremental turn-of-the-nut tightening and the preload measured at each degree of tightness.

2.2.1 Bolt selection

Diameters, material grades, and thread pitches conformed to the same values as were tested in the progressive tightening tests. There were no set standards for galvanizing, bolt length or thread length in the tightening tests beyond what was required to carry out the tests. These properties do not affect the generation of preload in the bolt and were therefore not a concern in the testing. Table 2.7 lists the number of specimens with tightening data used in the current study. This number includes the progressive tightening tests for completeness. Summaries of the preloads generated by each turn-of-the-nut for each bolt diameter-thread pitch combination are found in Section 3.

Table 2.7 – Tightening test matrix

	Number of Specimens Tested	
Bolt Diameter (in.)	UNC	8UN
1	-	3
1-1/4	2	2
1-1/2	1	2
1-3/4	5	3
2	2	2
2-1/4	3	2

2.2.2 Base-plate selection

Tests which were exclusively for the purpose of gathering preload data at degrees of turn-of-the-nut tightening were carried out with the base-plate specified in Table 2.3. Tightening data collected as a part of the progressive tightening tests use the base-plates specified in Table 2.4. The specific base-plate thickness used in each test are noted in Table 2.8.

2.2.3 Specimen preparation

As tightening tests were performed over an extended duration throughout the research, many of the values that were held constant during progressive tests varied during the tightening tests. Among these were groove dimensions and plate thickness. These values varied principally between the progressive and the tightening tests.

Also, the additional tightening specimens were only gaged in the middle location, centered between the nuts so to capture the average preload experienced by the section of the bolt spanned by the base-plate. The majority of the gaging was done as in the progressive tightening tests; there were two gages on opposite sides of the bolt, which were averaged to obtain the nominal preload.

Some tests were carried out using a center gage method either in addition to, or in place of the two-gage approach used throughout the rest of the research effort. This was done in order to gather data in a manner that had less impact on bolt geometry. A 0.078 inch diameter hole was drilled in the bolt and filled with epoxy, into which a strain gage was inserted at a depth corresponding to the midpoint of the base-plate where preload readings were observed. It was found to be accurate or slightly underestimate preload, which is conservative (Keating et al. 2004). Thus, the preload data gathered by the center gage approach is also included. Data on each tightening specimen is found in Table 2.8.

Table 2.8 – Tightening specimen information

Diameter (in.)	Thread Pitch	Gage Type (Two gage, center, or both)	Reduced area (in.^{2})	Base-plate thickness (in.)
1	8	Both	0.392	1.25
1	8	Two gage	0.544	1.25
1	8	Two gage	0.503	1.25
1-1/4	7	Center	0.964	1.50
1-1/4	7	Two gage	0.788	1.50
1-1/4	8	Center	0.995	1.50
1-1/4	8	Two gage	0.821	1.50
1-1/2	6	Two gage	1.199	1.88
1-1/2	8	Center	1.487	1.88
1-1/2	8	Two gage	1.315	1.88
1-3/4	5	Both	1.705	2.25
1-3/4	5	Two gage	1.740	2.25
1-3/4	5	Two gage	1.730	2.25
1-3/4	5	Two gage	1.730	2.25
1-3/4	5	Two gage	1.730	2.25
1-3/4	8	Center	2.075	2.25
1-3/4	8	Two gage	1.831	2.25
1-3/4	8	Center	2.075	2.25
2	4-1/2	Two gage	2.417	2.38
2	4-1/2	Two gage	2.226	2.38
2	8	Two gage	2.666	2.38
2	8	Two gage	2.438	2.38
2-1/4	4-1/2	Two gage	2.935	2.75
2-1/4	4-1/2	Two gage	2.972	2.75
2-1/4	4-1/2	Two gage	2.849	2.75
2-1/4	8	Two gage	3.458	2.75
2-1/4	8	Two gage	3.358	2.75

2.2.4 Experimental procedure

The procedure for the tightening tests was fundamentally the same as the turn-of-the-nut tightening portion of the progressive tests. Bolts were lubricated and tightened in the manner specified above, with all appropriate preload readings for the middle gage location taken at each turn-of-the-nut. However, no fatiguing was done at any point during the test.

2.2.5 Data collection and assembly

Strain readings were taken and transformed into readings of stress, the preload in the bolt at the given turn-of-the-nut. In the case of the two gage setup, the stresses taken at each location for the given degree of tightness were averaged. From the collected values of preload, bar charts were developed to observe the average preloads generated for any turn-of-the-nut at each diameter-thread pitch combination. The charts also displayed the range of preload data for each diameter-thread pitch combination.

As with the progressive tightening data, adjustment of recorded values was necessary due to the grooves or holes machined for strain gage application. The process for adjusting the preload values is found in Section 4.

2.3 Fatigue test

For the purposes of verification, a single fatigue test was carried out on a bolt 2 inches in diameter with an 8UN thread pitch. The selection of this bolt as the critical condition is discussed in Section 4. The method used to fatigue the bolt was similar to the fatigue portion of the progressive tightening test detailed above, except the tightening to the appropriate turn-of-the-nut was carried out in a single event instead of the given incremental procedure.

For the fatigue testing, loads were applied to generate a minimum of 1 ksi in the bolt and a maximum of 15 ksi for a stress range of 14 ksi. This loading is the same as was used in the progressive tightening test of that specimen.

3 LABORATORY RESULTS

Through the methods presented in Section 2, laboratory tests were carried out on all specimens. Data was collected and presented in a set of graphs and tables showing stress range versus turn-of-the-nut tightening.

3.1 1 inch, 8UN specimen

3.1.1 Presentation of graph

The graph for the 1 inch, 8UN specimen is found as Figure 3.1. The key for the strain gage labels is found in Figure 2.1. With these two figures as references, one may begin to understand the change in fatigue behavior observed in the bolt as a function of preload.

In the figure, degrees of turn-of-the-nut tightening make up the *x*-axis, and represent a field-measurable estimation of preload. The relationship between turn-of-the-nut and preload is discussed later in the section. The values plotted consist of the stress range experienced by the bolt at each location, at each turn-of-the-nut. There is no data plotted for the Avg Middle value at 1/6 turn-of-the-nut because the strain gages at that location both failed. Numerical strain gage readings are found in the data tables, discussed below.

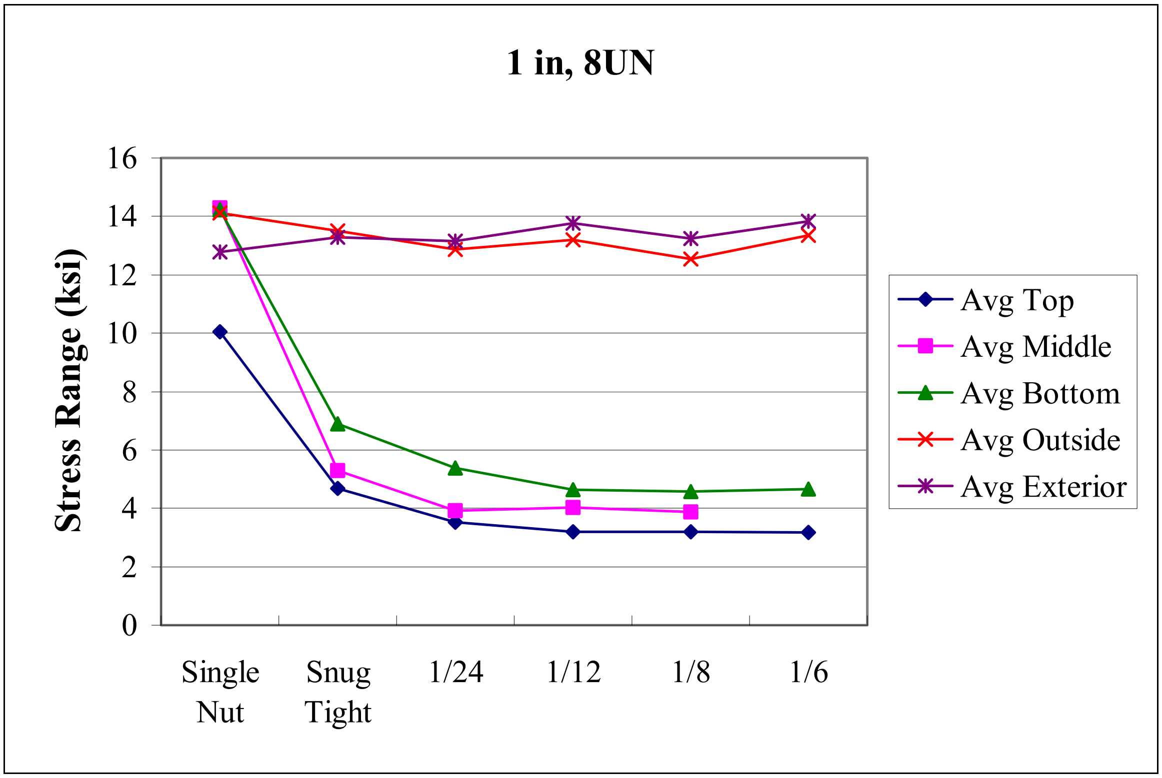

Figure 3.1 – 1 inch, 8UN stress range vs. turn-of-the-nut plot

Section 1 discussed two indicators of adequate fatigue performance. The first of these, and most important to the testing, is the reduction of stress range inside the double-nut connection. The figure above illustrates such reduction. It can be clearly seen that as the connection is tightened, the stress ranges experienced by the section of the bolt between the nuts decreases. By lowering the stress range experienced in the interior of the double-nut assembly, fatigue life of the bolt can be improved from the single nut case to one of adequate fatigue performance.

Over the course of tightening, the exterior location sees a relatively constant stress range. This stress range is equal to the applied loading and serves as a point of reference for the stress ranges experienced at the other gage locations. Slight variation exists in each of the tests due to the imperfect data collection method identified in Section 2. The variations are acceptable, however, and even with the imperfections the reading serves as a reliable basis of comparison.

Experimentally in Frank (1980) and Van Dien et al. (1996), and analytically in Hodge (1996) it has been shown that without significant preload, failure is likely to occur at the location of the top gages. The degree of tightness in the line graph labeled "Single Nut" is the fatigue response at zero preload. In the single nut configuration, as shown in Figure 3.2, it is observed that the top gage location is the only point at which the geometry is such that there is a stress concentration where cracks would be expected to initiate.

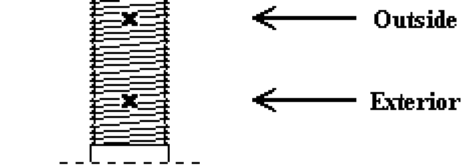

Figure 3.2 – Single nut gage locations

The inclusion of preload into the system rapidly and dramatically changes the stresses throughout the preloaded section of the bolt. Also, when the bottom nut is added to the geometry of the fixture, it introduces a stress concentration. Stress range also drops rapidly in the interior regions of the double-nut fixture, leaving the location of the outside gage the critical location.

3.1.2 Presentation of stress range table

The data found below in Table 3.1 corresponds to that plotted in Figure 3.1 above. These values illustrate numerically what the graphs show by inspection: stress ranges drop as preload is increased, with the outside stress dropping less rapidly than the other probable failure locations.

Table 3.1 – 1 inch, 8UN stress range vs. turn-of-the-nut table

Gage Location	Stress range (ksi) per turn-of-the-nut					
	Single Nut	Snug Tight	1/24	1/12	1/8	1/6
Top	10.0	4.7	3.5	3.2	3.2	3.2
Middle	14.3	5.3	3.9	4.0	3.9	Lost Gage
Bottom	14.2	6.9	5.4	4.6	4.6	4.7
Outside	14.1	13.5	12.9	13.2	12.5	13.3
Exterior	12.8	13.3	13.1	13.8	13.2	13.8

3.1.3 Interpretation of stress range data

3.1.3.1 Questions raised

There are several interesting behaviors to note from both the plot and the data table found above. The first of these is the separation of the data at gages measuring the same theoretical stress level. It could easily be expected for the top, middle, and bottom gages to read the same stress levels, as they all were intended to capture the measure of preload in the bolt between the nuts. Instead, the top gages read values lower than the middle gages at every point, and considerably lower at the single nut reading. This behavior is perplexing, since if there were to be variation, it seems far more likely that the top gages would read the largest stress ranges since it is the threads at that location that bear the immediate loading and unloading during testing. Similarly unusual is the fact that the bottom gage location reads values slightly higher than the middle location for all turn-of-the-nut. These behaviors were set to study via finite element analysis, which is discussed in Section 4.

Another interesting phenomenon in the gradual reduction in stress range experienced at the outside gage location. Since that gage was placed at the absolute exterior of the double-nut connection, it would be reasonable to expect the same stress levels there as at the exterior location, two inches down the bolt. This behavior was also examined in the finite element analysis discussed in Section 4.

3.1.3.2 Behavior inferences

Even with the unexpected stress range separation seen in the analysis of the laboratory results, several important observations can be taken from the graph and corresponding data table for the 1 inch, 8UN specimen. It is shown by inspection that a snug tight condition lowers stress ranges in the fixture interior to a point far below that of the exterior and outside locations. This indicates that a minimum level of tightening is required achieve an adequate level of fatigue performance of the bolt.

Also indicated by the progressive tightening test is that stress range decreases to a minimum value ranging between 3 and 5 ksi. For the 1 inch specimen, this minimum seems to have occurred at 1/12 turn-of-the-nut and tightening beyond that point yielded insubstantial changes in stress range seen between the nuts.

3.1.4 Presentation of preload table

Table 3.2 summarizes the static tightening data for each specimen, which states the nominal preload at each turn-of-the-nut and corresponds with Figure 3.1 and Table 3.1. Readings were taken at the middle location labeled in Figure 2.1 and represent an average preload for the specimen. Each of the two gages at that location is located directly between the two nuts responsible for the preloading. Those two gages, located opposite one another as detailed in Section 2, are averaged to find the mean stress found in the row marked "Recorded".

Table 3.2 – 1 inch, 8UN preload table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	0.3	23.2	1.6	24.3	24.8	63.9	Lost Gage		
Adjusted	---	---	---	---	20.2	61.5	Lost Gage		

The row labeled "Adjusted" is a modification of the directly recorded values. As discussed in Section 2, grooves were cut into each bolt to allow for the affixing of strain gages. This reduction of area non-conservatively affects the amount of stress seen in the bolt, increasing it due to the decrease in area. The adjusted preload is the stress that would exist in the bolt if the area had not been reduced due to machining. The equation for determining this value was developed in Appendix B of Keating et al. (2004) and is located below as Equation 3.1. Since the adjustments are dependant on a reference to a snug tight condition, adjustments were not made for snug or zero readings.

$$\sigma_b = \frac{T}{PL}\left(\frac{1}{\frac{1}{E_b} + A_b C}\right) + \sigma_{snug} \tag{3.1}$$

where:

σ_b = adjusted bolt tensile stress

σ_{snug} = bolt tensile stress at a snug tight condition

T = turn-of-the-nut past snug tight

P = thread pitch (number of threads per inch)

L = preloaded length (base-plate thickness)

E_b = Modulus of Elasticity of the bolt

A_b = nominal tensile cross-sectional area of the bolt

and:

$$C = \frac{T_{test}}{A_{testedbolt} PL\left(\sigma_{test} - \sigma_{snug}\right)} - \frac{1}{E_b A_{testedbolt}} \tag{3.2}$$

where:

T_{test} = turn-of-the-nut

$A_{testedbolt}$ = net area of the bolt tested (accounting for grooves)

σ_{test} = stress experienced at a given turn-of-the-nut, T_{test}, by a grooved specimen

3.2 Initial conclusions

Evaluation of the remaining ten specimens yields results that are very similar to those found above for the 1 inch case. Many of the same generalizations can be made for all remaining diameter-thread pitch cases. Examining exclusively the laboratory progressive tightening data, the following statements can be made. All final and complete conclusions are found in Section 6.

1. A snug tight condition was sufficient to lower stress ranges in the top, middle, and bottom locations to a point below that experienced at the outside and exterior locations. The reduction of stress ranges indicates that fatigue life at a snug tight condition would be extended beyond that seen at the single nut case and that, if fatigued to failure, the failure location would occur outside the double-nut connection. (This conclusion is modified through finite element analysis. See Section 4.)
2. By 1/6 turn-of-the-nut, all diameters and thread pitches see stress ranges reduced between the nuts of the connection to a value between 2.0 and 5.5 ksi. This takes into account the top, middle, and bottom gages.

3.3 Tightening test data

To further expand on the correlation of preload and turn-of-the-nut, data taken from tightening tests was considered. The methodology for these tests, which excluded any cyclic loading and only measured preload at the middle location, is found in Section 2.

3.3.1 Preload generation

The following charts show the average preload for 1/24 and 1/6 turn-of-the-nut tightness for both the UNC and 8UN thread pitches. This bounds the turn-of-the-nut tightening data for both thread pitch cases. A full accounting of such data, ranging from snug tight to 1/6 turn-of-the-nut is found in Appendix B. Error bars show the range of data collected. All preload values were adjusted using the methods described above.

Figure 3.3 shows the minimum turn-of-the-nut tightening data for the UNC thread case. A number of useful observations can be made from the above graph regarding tightening trends and preload variance. Generally, it is clear that equivalent turn-of-the-nut tends to generate less preload with increase in plate thickness according to the bolt diameter. This is true despite the fact that as bolt diameter and plate thickness increase, the thread pitch becomes coarser. Base-plates increase in thickness as bolts increase in diameter as shown in Table 2.4. This seems to indicate that the plate thickness is a more important indicator of preload development than the thread pitch.

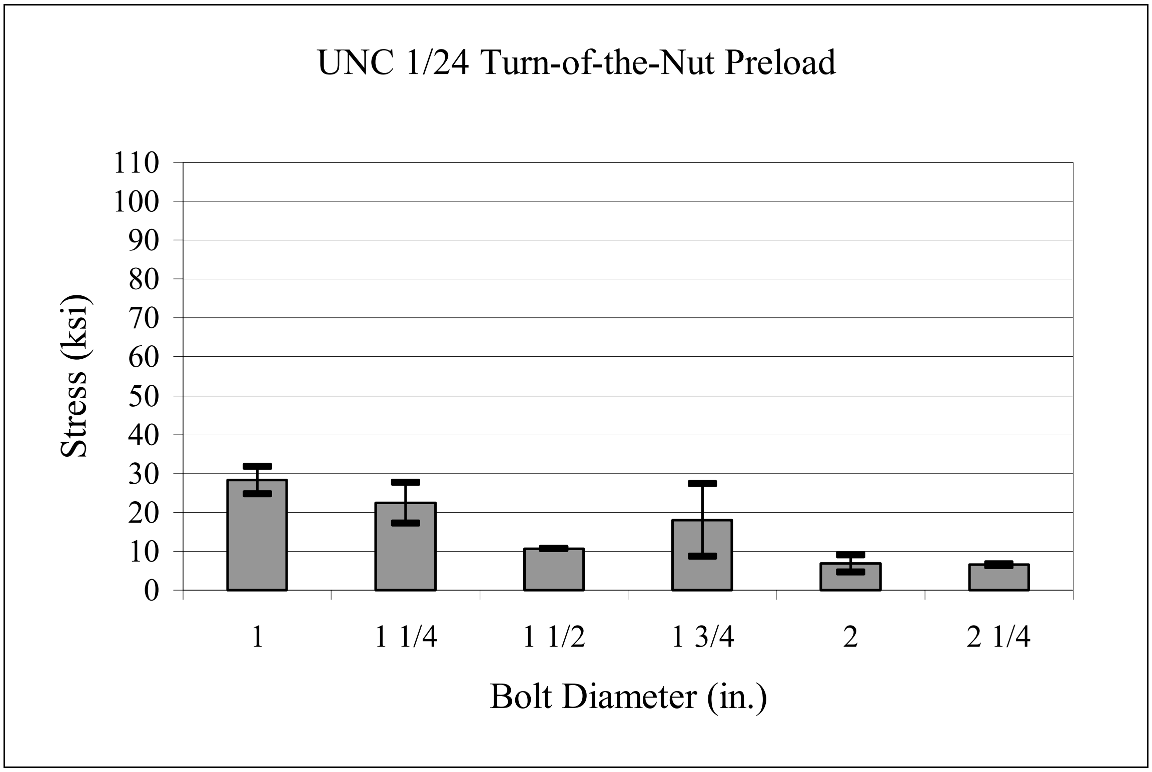

Figure 3.3 – UNC minimum turn-of-the-nut preload data

There is still some question that remains regarding the impact of thread pitch versus that of plate thickness on preload generation. The data for the 1-1/2 inch UNC case shows a significantly lower preload generated than seen in the next larger specimen, the 1-3/4 inch UNC. However, if one references Table 2.8, it can be seen that only one specimen was tested for that diameter-thread pitch combination. That particular specimen saw uncharacteristically low preloads throughout testing, as can be seen in Table 3.3. Circumstances seem to point toward an aberrant test, however this does highlight a highly significant fact about turn-of-the-nut tightening; while it may be the best and most consistent method available, preload generation using turn-of-the-nut tightening varies greatly and is difficult to predict. This must be considered in any form of tightening standard proposed.

Table 3.3 – 1-1/2 inch, UNC preload

Turn-of-the-nut	Recorded Preload (ksi)	Adjusted Preload (ksi)
Zero1	-1.3	-
Snug1	4.9	-
Zero2	-1.4	-
Snug2	5.4	-
1/24	11.4	10.6
1/12	14.1	12.9
1/8	17.7	15.9
1/6	19.0	17.1
Zero3	2.1	-

Due to the constant thread pitch between bolt diameters, the 8UN 1/24 turn-of-the-nut data found in Figure 3.4 complies much better with the expected reduction in preload with increase in diameter, and therefore plate thickness. The data set for the 8UN case was also a bit more complete, with at least two specimens tested for each diameter, lessening the influence of any seemingly abnormal test as in the UNC case. The data for the 1 inch column is the same as in Figure 3.3 because thread pitch is the same for both UNC and 8UN cases.

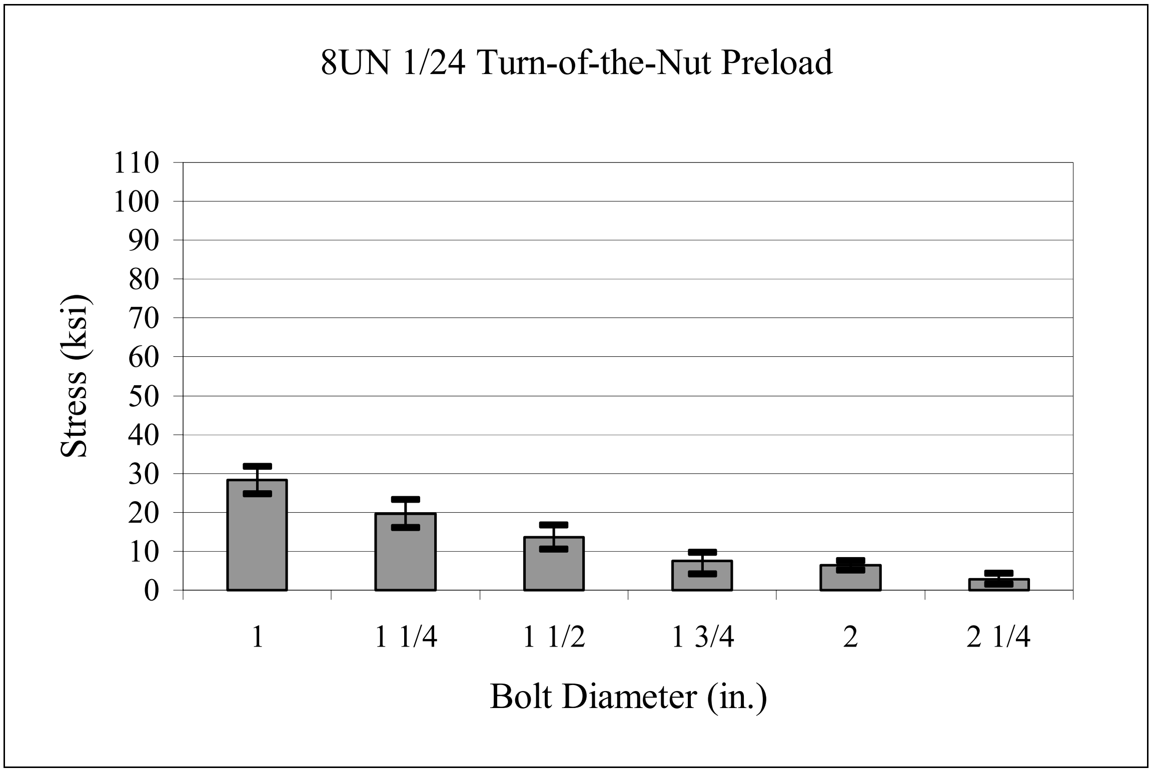

Figure 3.4 – 8UN minimum turn-of-the-nut preload data

Figure 3.5 shows the same basic representation of data as did the 1/24 case shown in Figure 3.3. The 1-1/2 inch bolt shows unusually low preload as discussed above. One of the 1 inch specimens reached the yield point of the steel at 1/6 turn-of-the-nut, but yielding did not occur for any other tightening tests. There is no data for the 2 inch case because neither of the tested specimens achieved 1/6 turn-of-the-nut tightening. This issue is discussed below.

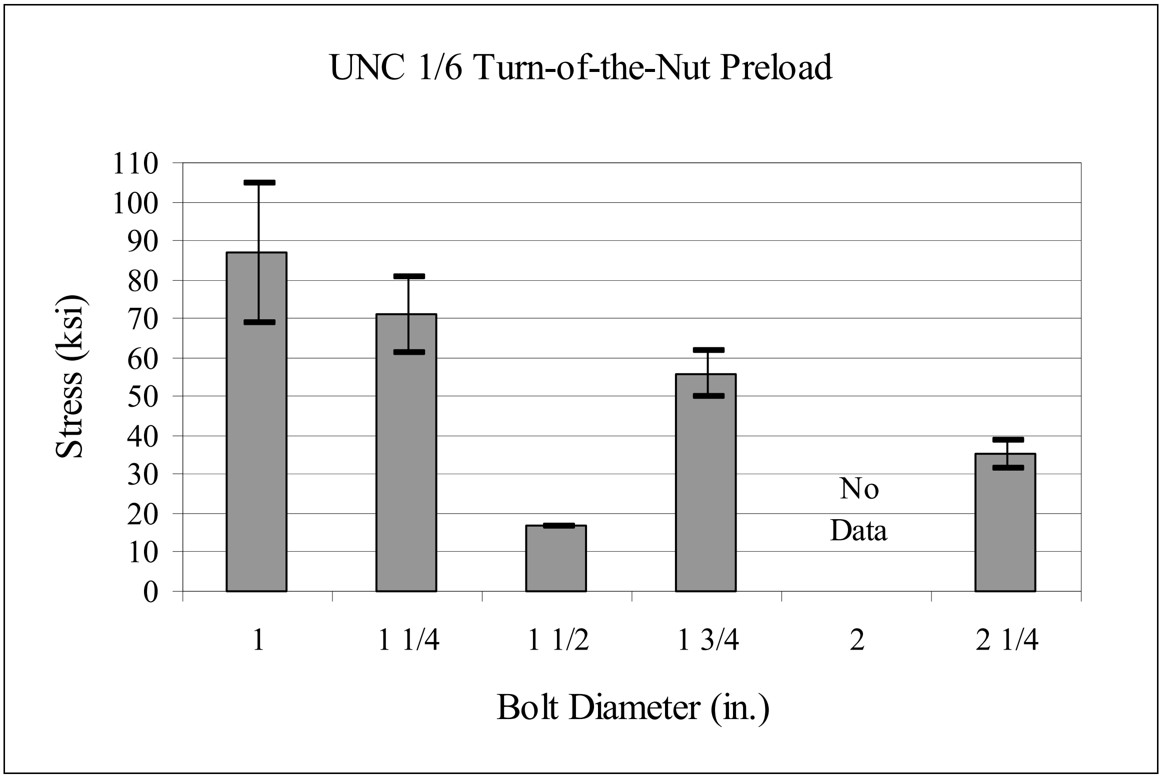

Figure 3.5 – UNC maximum turn-of-the-nut preload data

Again the 8UN case, shown for 1/6 turn-of-the-nut in Figure 3.6, follows the predicted trend of decreasing preload with increasing bolt diameter and plate thickness. As above, the 1 inch data is the same as found in Figure 3.5 because both UNC and 8UN categories have the same thread pitch for the 1 inch case. Aside from the aforementioned 1 inch case, no bolts were brought to yield at 1/6 turn-of-the-nut.

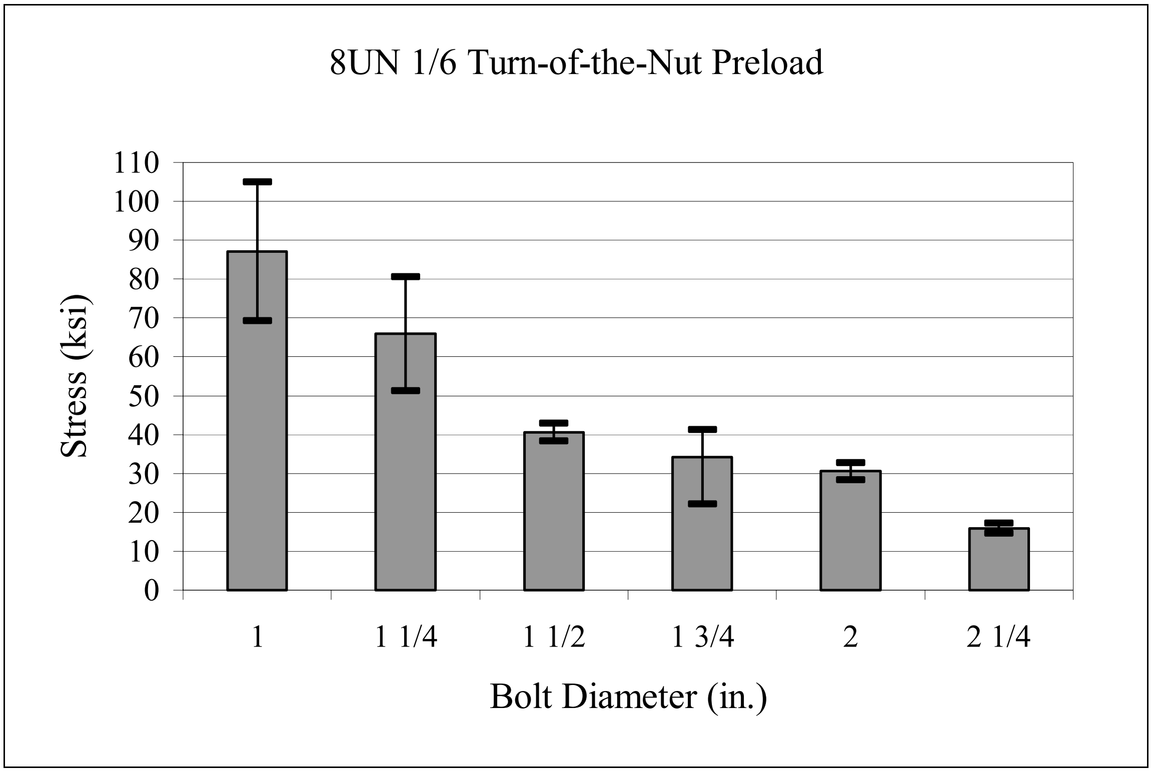

Figure 3.6 – 8UN maximum turn-of-the-nut preload data

3.3.2 Turn-of-the-nut feasibility

In addition to the generation of preload, the feasibility of attaining up to 1/6 turn-of-the-nut was examined. In all tests except for the 2 inch UNC case, the 1/6 turn-of-the-nut was achieved, as noted above. In the two tests carried out on this diameter, the feasible limit was slightly greater than 1/8 turn-of-the-nut. The 2 inch 8UN bolt and both the UNC and 8UN thread pitches for the 2-1/4 inch bolt achieved 1/6 turn-of-the-nut in every case.

No conclusions can be drawn from the data collected in the current study. The two specimens tested for the diameter-thread pitch combination in question are not sufficient to state with certainty the feasibility of turning the 2 inch UNC bolts to 1/6 turn-of-the-nut. Recommendations for further tests are discussed in Section 5.

3.3.3 Initial conclusions

Evaluation of the tightening data allows for the development of several initial conclusions about preload as it related to thread pitch and bolt diameter. All final and complete conclusions are found in Section 5.

1. UNC thread pitches generate more preload than 8UN thread pitches of the same diameter. This is directly related to the slope of the threads in each pitch, since a coarser pitch displaces more of the bolt with the same turn-of-the-nut.
2. Equivalent turn-of-the-nut generates greater preload in tests performed on smaller diameter bolts. This is the result of greater plate thickness with increasing diameter. This behavior is more easily seen in 8UN bolts, which holds thread pitch constant with increasing diameter. An equivalent turn-of-the-nut displacement of bolt generates less preload with greater plate thickness.
3. Tightening by 1/6 turn-of-the-nut will not develop yielding in the preloaded section of the bolt for tested diameters greater than 1 inch.
4. A tightening level of 1/6 turn-of-the-nut is attainable for all diameter-thread pitch combinations with the possible exception of 2 inch UNC,

which saw in testing a feasible limit of slightly greater than 1/8 turn-of-the-nut.

3.4 Verification

As discussed in Section 1, two criteria were laid out as indicators of adequate fatigue behavior. The first of these was the observation of decreased stress ranges in the preloaded section of the bolt. To this end, the above progressive tightening tests were carried out. The second was the migration of failure location to the exterior of the double-nut connection, below the bottom nut. Such a shift is a practical confirmation of reduced stress ranges within the preloaded section of the bolt.

Given these two criteria, verification of the results was desired. This was carried out through two processes: a fatigue test and finite element modeling. To maximize the efficiency of such testing, both processes were carried out on the critical bolt diameter-thread pitch combination.

3.4.1 Selection of the critical specimen

The critical specimen was identified as the diameter-thread pitch combination that generated the minimum preload for the minimum turn-of-the-nut. The snug tight condition was deemed to provide too wide a variance of results and too subjective a measurement of tightness to be specified by any form of turn-of-the-nut standard. Thus, 1/24 turn-of-the-nut was specified as the minimum tightness to be considered.

Several considerations were given to the selection of the test case that fit this criterion. To begin, all specimens evaluated in the progressive tightening were considered, except the 2-1/4 inch, 8UN bolt. This case was recommended for exclusion from TxDOT specification by Keating et al. (2004) due to unreliable fatigue performance at any turn-of-the-nut tightness. Of the remaining test cases, tightening data was considered for all tightening data as discussed above. Both the minimum levels of preload as well as the average preload generated for each test case were compared for 1/24 turn-of-the-nut. This comparison can be seen in Figure 3.7.

Data is mixed in the determination of the critical test case, but the 2 inch, 8UN specimen was taken as the most critical. Its average preload was the lowest of any considered specimen. While the 1-3/4 inch 8UN and 2 inch UNC cases did show at least one case of lower preload generated each, the maximum difference in preload was only 1 ksi. Further, the general trends in the generation of preload must also be taken into account: greater plate thickness that correlates with increasing diameter generates less preload than the thinner base-plates at smaller diameters, and the 8UN thread pitch generates less preload than UNC at equivalent turn-of-the-nut. When looking at all available information, the selection of the 2 inch, 8UN specimen is justified as the worst-case for the generation of preload.

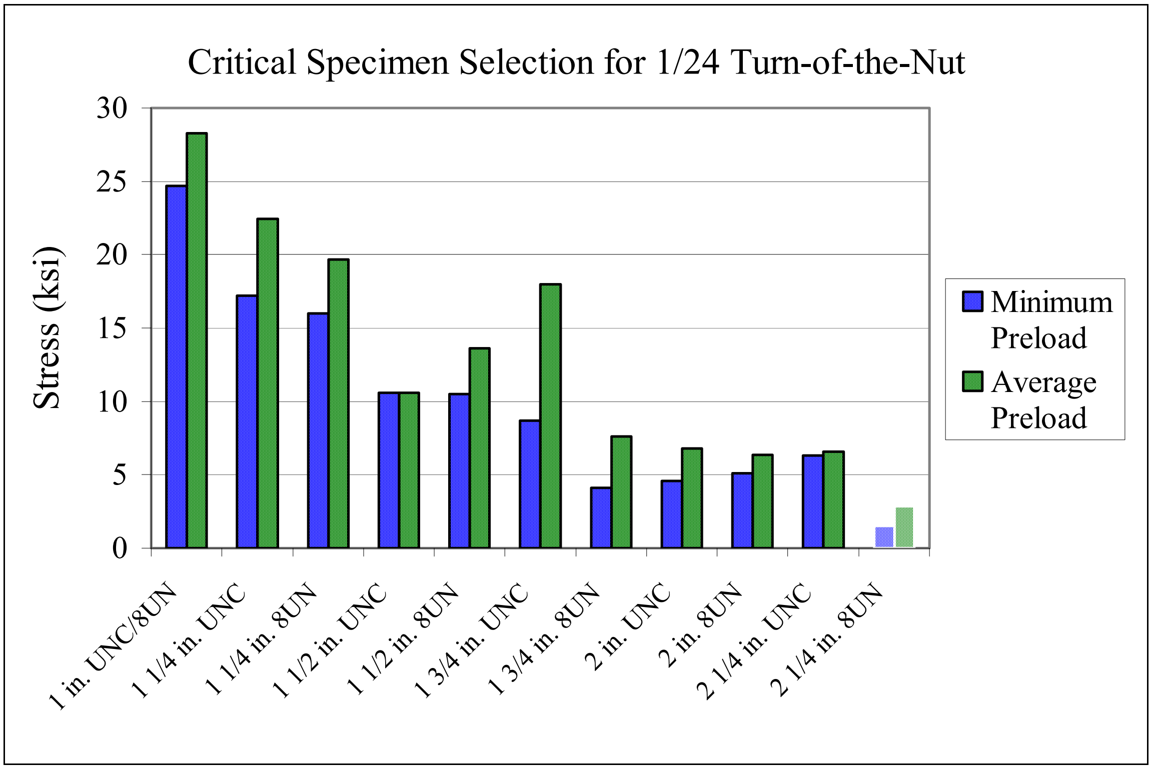

Figure 3.7 – Preload comparison for critical specimen selection

3.4.2 Fatigue verification

Figure 3.8 shows the stress range plot for the critical bolt specimen. The same bolt specimen was used in the verification fatigue test as was used in the progressive tightening test, with the same loads, although no strain readings were taken. A turn-of-the-nut of 1/24 was predicted, according to laboratory data, to migrate the failure location to the outside of the connection. A condition for satisfactory performance was set at 2 million cycles without failure at the specified 14 ksi stress range. This condition was specified in previous research sponsored by TxDOT, reported in Keating et al. (2004).

The first fatigue test that was run on the 2 inch 8UN bolt failed after 1,523,950 cycles at an applied stress range of approximately 14 ksi. Failure occurred at the first fully engaged thread of the top nut, not outside the connection as predicted.

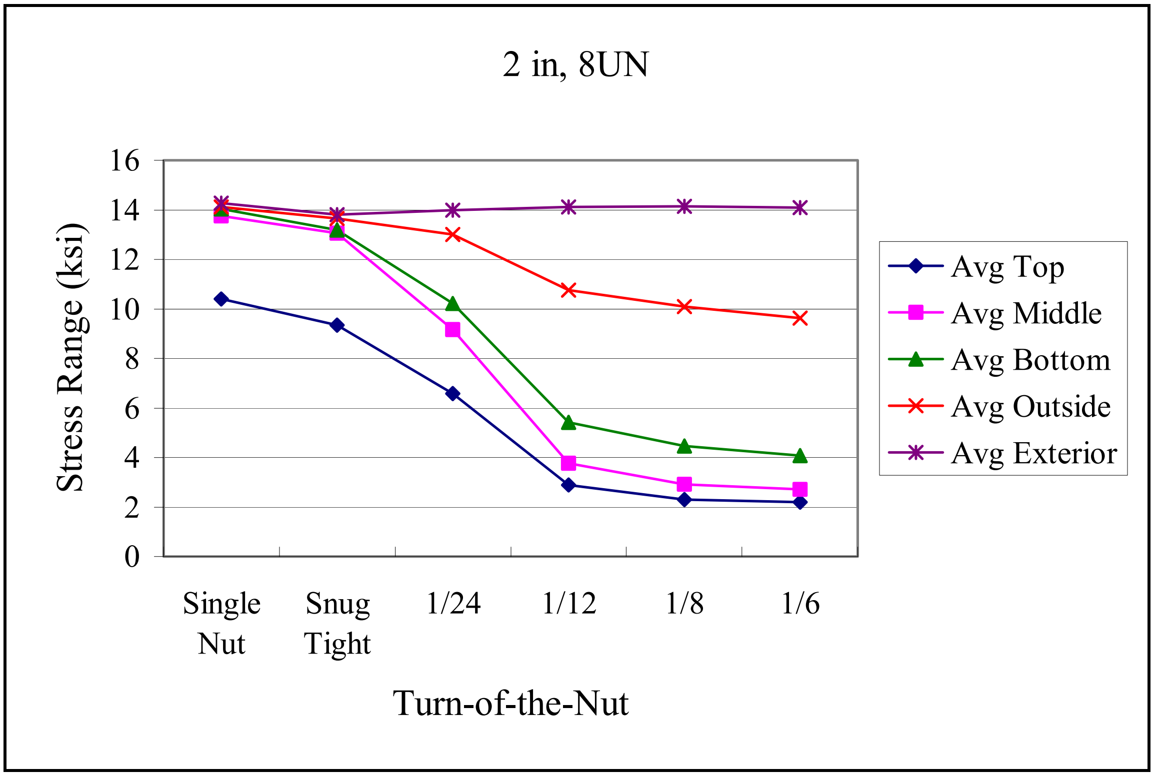

Figure 3.8 – Critical bolt stress range plot

The same specimen was used again with the double-nut fixture moved down the bolt to adjust for the broken length, at the same loads and the same degree of tightness, to attempt to duplicate results. After 6,902,870 additional cycles, no failure occurred. It is difficult to interpret this broad range of results for such limited testing, but it at the very least demonstrates the uncertainty in dealing with fatigue of anchor bolts. Results are illustrated in Figure 3.9, along with the categories of AASHTO fatigue classification.

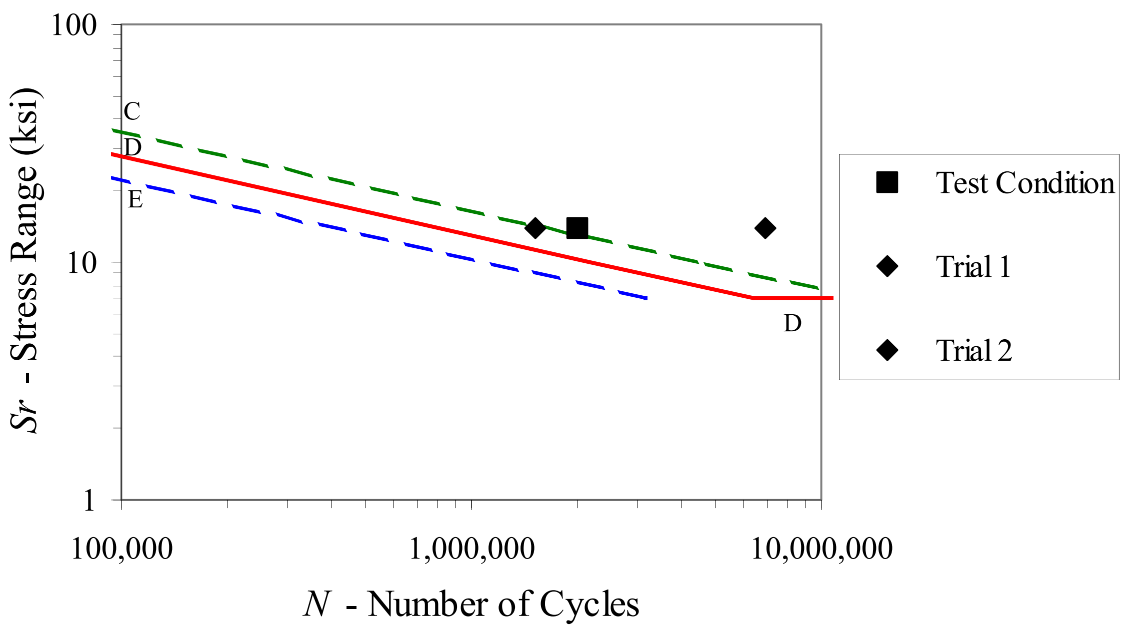

Figure 3.9 – Fatigue test results

The inconsistency between the two verification tests leads to further questions, but at least one conclusion. Even at 1/24 of a turn-of-the-nut, adequate performance can be achieved, as is shown by its tested life of nearly seven million cycles in the second fatigue test.

3.4.3 Verification by finite element analysis

Following the mixed data collected in the verification fatigue test, a detailed finite element analysis was carried out for the critical bolt specimen. This analysis can be found in Section 4.

4 FINITE ELEMENT ANALYSIS RESULTS

4.1 Model development

4.1.1 Model construction

A finite element model was developed to evaluate the theoretical behavior of a 2 inch 8UN anchor bolt for the purpose of verification of the laboratory results. As mentioned above, the 2 inch bolt diameter bolt with eight threads per inch was determined to be the critical bolt and was therefore chosen as the representative model to be analyzed using the finite element method. The model specifically evaluated the 10 inch section of bolt influenced by the double-nut connection described in Section 1. Bolt dimensions are given in Figure 4.1. Symmetry about the *y*-axis was used to simplify the model.

Areas were constructed to allow for easy collection of data at key locations within the model associated with locations of strain gages on the laboratory setup as shown in Figure 4.2. Results gathered from model were taken along the cross-section defined by the vertical line at the depth of the gages in the laboratory tests, as seen in the figure. The model had 4720 nodes and 1307 elements. All solid bodies were meshed with quadratic eight-noded PLANE82 elements with a preferred side dimension of 0.125 inches. This value was verified by a convergence test as discussed below. Meshing was carried out by ANSYS via the Free Mesh command. Preload was developed using two-noded PRETS179 elements on the bolt centered between the nuts. Contact was modeled using the ANSYS Contact Wizard and the pairing of contact and target elements, CONTA172 and TARGE169. The fully meshed model can be seen in Figure 4.3.

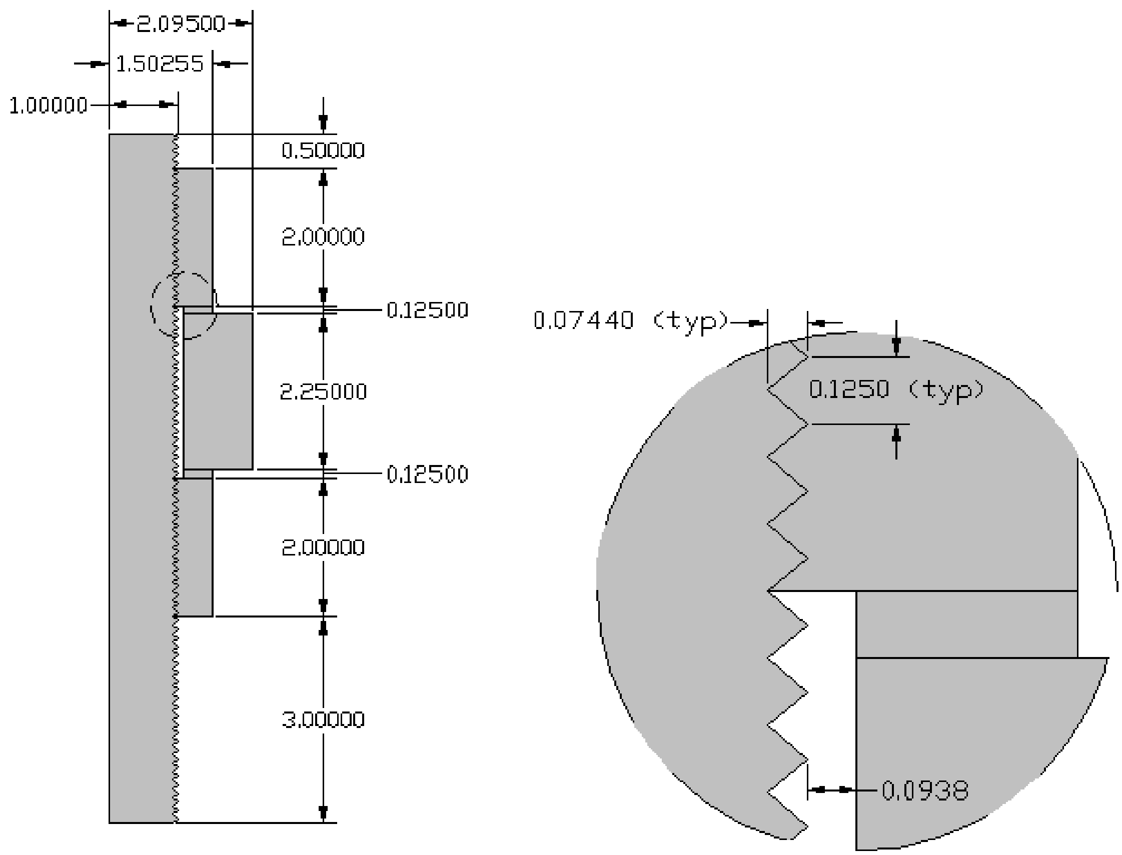

Figure 4.1 – Modeled bolt dimensions

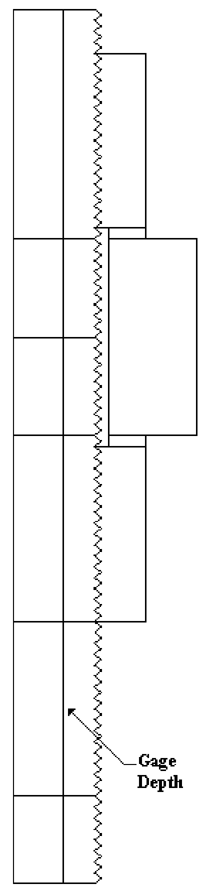

Figure 4.2 – FEM areas

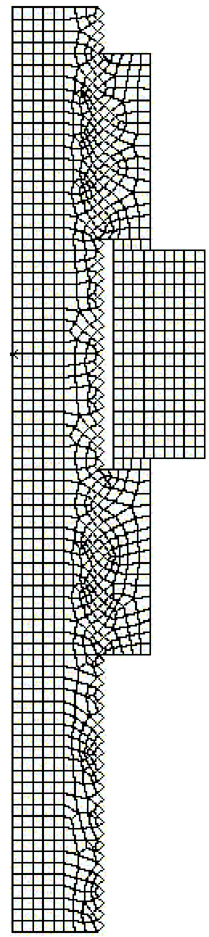

Figure 4.3 – FEM mesh

4.1.2 Boundary conditions

To simulate laboratory conditions and to allow for the *y*-axis symmetry to be utilized, the following boundary conditions were imposed. The base of the bolt was fully fixed against displacement in the *x*- and *y*-directions. The *y*-axis of the bolt about which symmetry was taken was fixed against displacement in the *x*-direction. The nuts, washers, and base-plate were also restrained from displacement in the *x*-direction. The end result was a model that neglected effects of bending and allowed only axial loading to play a significant role in the analysis.

4.1.3 Model convergence

A convergence study was performed to verify the validity of values generated by the model. Three different preferred mesh sizes were specified and ANSYS was allowed to plot the element mesh within those constraints. The mesh size of the modeled base-plate was not varied due to the need for a constant geometry for the application of the load. The finest mesh was used for all iterations of the convergence study.

Preferred mesh sizes supplied to ANSYS for consideration in its free meshing of the areas were 3/8 inch, 1/4 inch, and 1/8 inch. Under identical loading equal to the generation of 1 ksi of stress in the length of the bolt, the stress readings seen in Figure 4.4 were taken and compared. Differences between the readings were plotted in Figure 4.5.

Results from the convergence study show that data taken with the preferred element width of 1/8 inch generates reliable readings of stress in the bolt model. Even the improvement in the model from 3/8 inch to 1/4 inch preferred element width generated negligible changes in stress readings, with a maximum change of approximately 0.045 ksi. The final change of less than 0.030 ksi is more than adequate for the model.

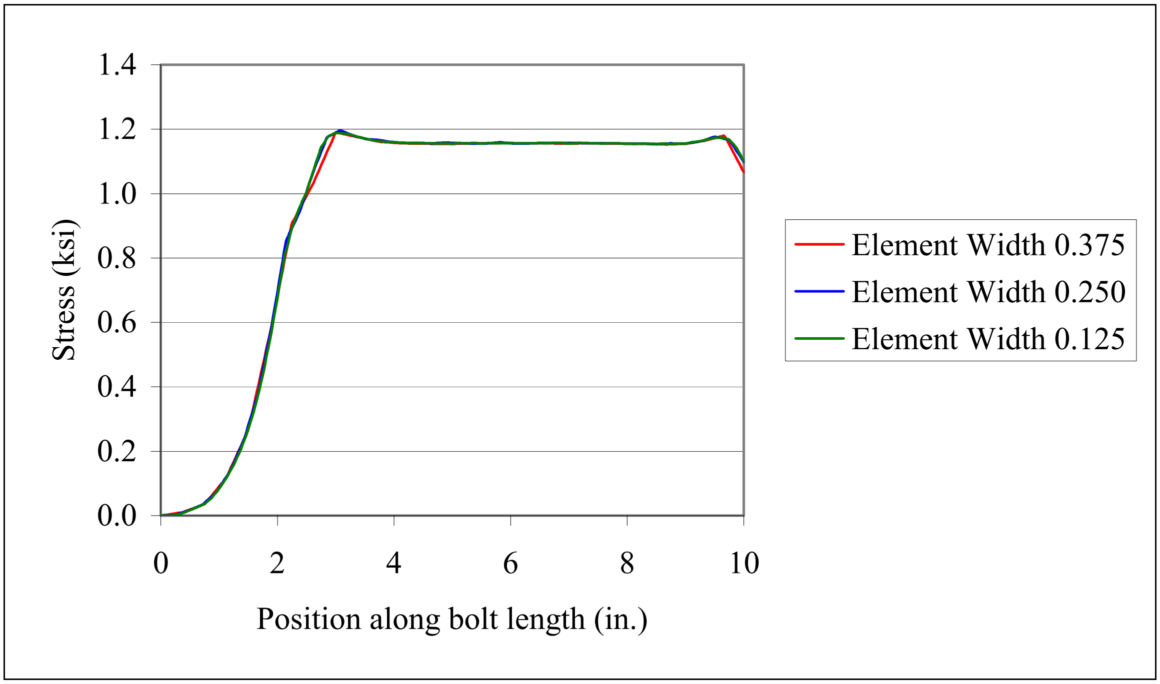

Figure 4.4 – FEM convergence study results

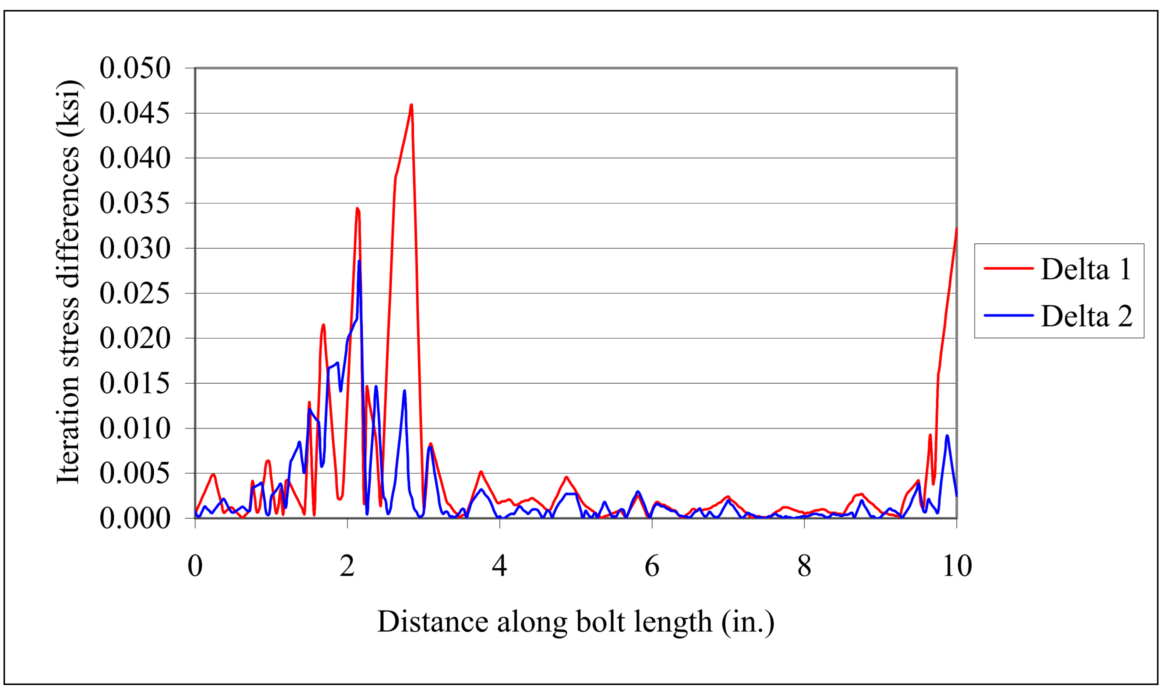

Figure 4.5 – Convergence study changes in stress between cases

4.1.4 Model loading

The two loading parameters of the model were preload and applied tensile load. Preload was generated using the aforementioned PRETS179 elements by applying an internal displacement between rows of elements at a location in the bolt corresponding to the middle gage location. This location is noted with an asterisk in Figure 4.3 and the element separation can be seen in Figure 4.6. This displacement was varied to generate different levels of preload in the bolt. Analysis of the bolt at these loads with no externally applied forces allows for the use of preload as the variable in plotting stress range. A contour plot of the maximum preload distribution, a nominal 56 ksi, is found in Figure 4.6. Stresses plotted below and throughout the analysis are those calculated at the nodes with averaging across the elements.

The second aspect of model loading was the application of degrees of tensile stress to obtain a stress range. Load was applied to the bolt in a manner similar to lab conditions, with a load distributed across the nodes of approximately 1/2 inch of the model measured from the outer diameter of the base-plate. The loading location can be seen in Figure 4.7.

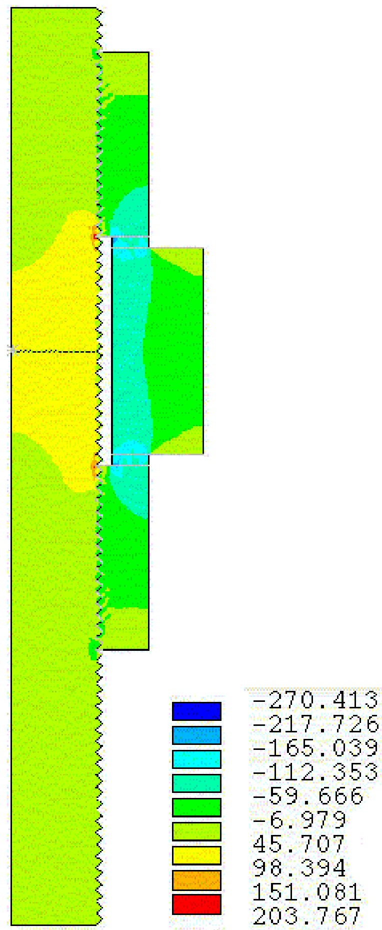

Figure 4.6 – Preload distribution

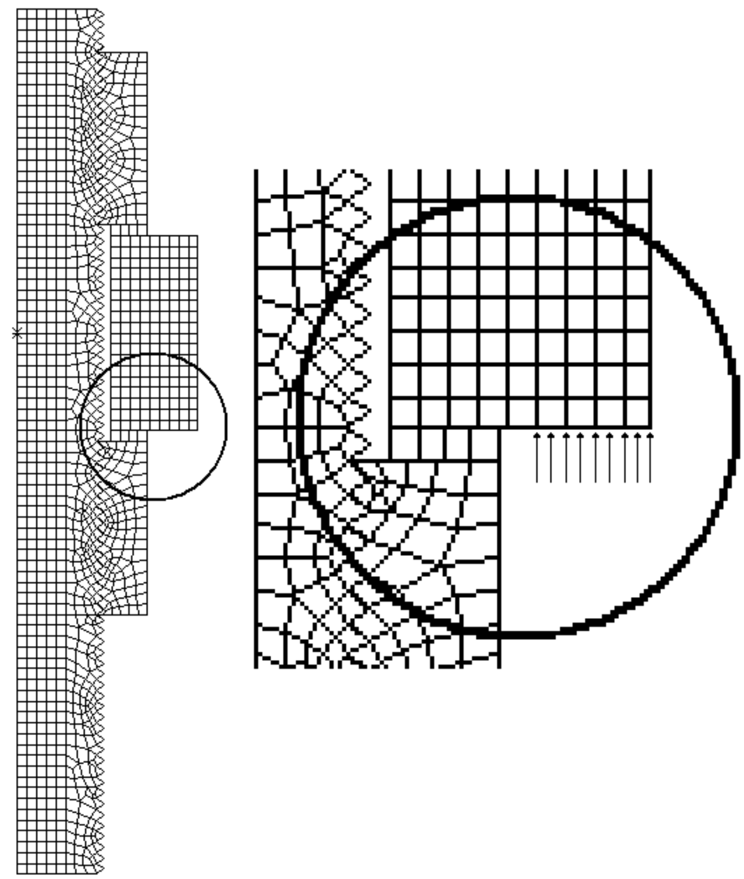

Figure 4.7 – FEM load location

This applied point loading was scaled to give the same stresses in the model as were witnessed in the laboratory tests for the single nut case. Adjustments were made by inspection of the graph shown in Figure 4.8. As discussed in Section 2, two gages took readings at each gage location. A sampling of the results collected at each gage was used to determine the appropriate levels of stress in the model. Loads were altered until the behavior of the model was fundamentally the same as that in the laboratory test. It was determined that 0.12 kips applied at each of the loaded nodes generated the appropriate lower load and 1.6 kips generated the appropriate upper load.

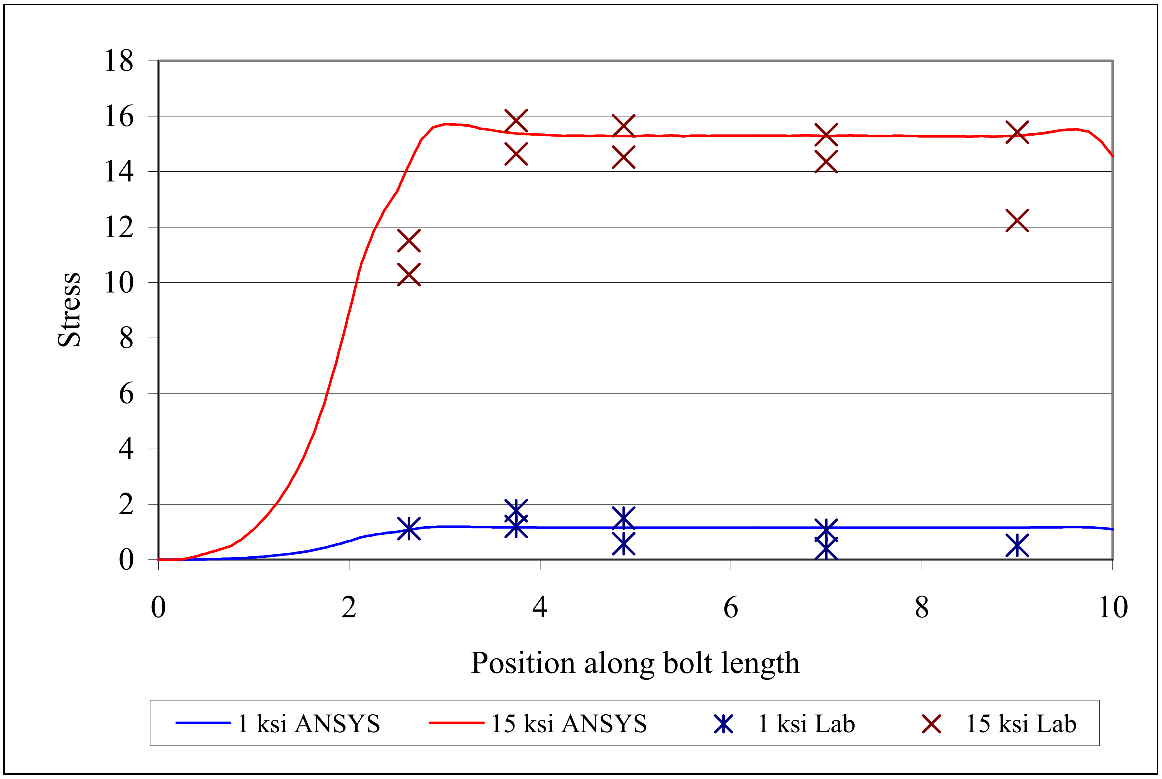

Figure 4.8 – FEM load calibration

4.2 Analysis

Once loads were determined, an analytical set of tests was carried out in the same manner as the laboratory testing. The bolt model, with zero preload representing the single nut test condition, was subjected to the lower and upper limit loads determined in the test calibration. The difference in response between a 1 ksi and 15 ksi load is the stress range. Results were collected for each case, and the stress range plotted versus preload in the same manner as shown in Section 3 dealing with laboratory testing.

Preload was then applied to the section of the bolt between the nuts to simulate the tightening of the bolt and a reading of preload taken. Applied preload levels can be seen in Figure 4.9. The response of the bolt at maximum preload is given below in Figure 4.10 as a sample of collected data. The low and high loads were then applied and the stress range recorded. This was done for a number of preload values to a point beyond that observed in the lab. Through this collection of data, Figure 4.11 was constructed. This can be directly compared with results plotted for the 2 inch, 8UN bolt results from lab testing, shown in Figure 4.12 plotted with respect to bolt preload.

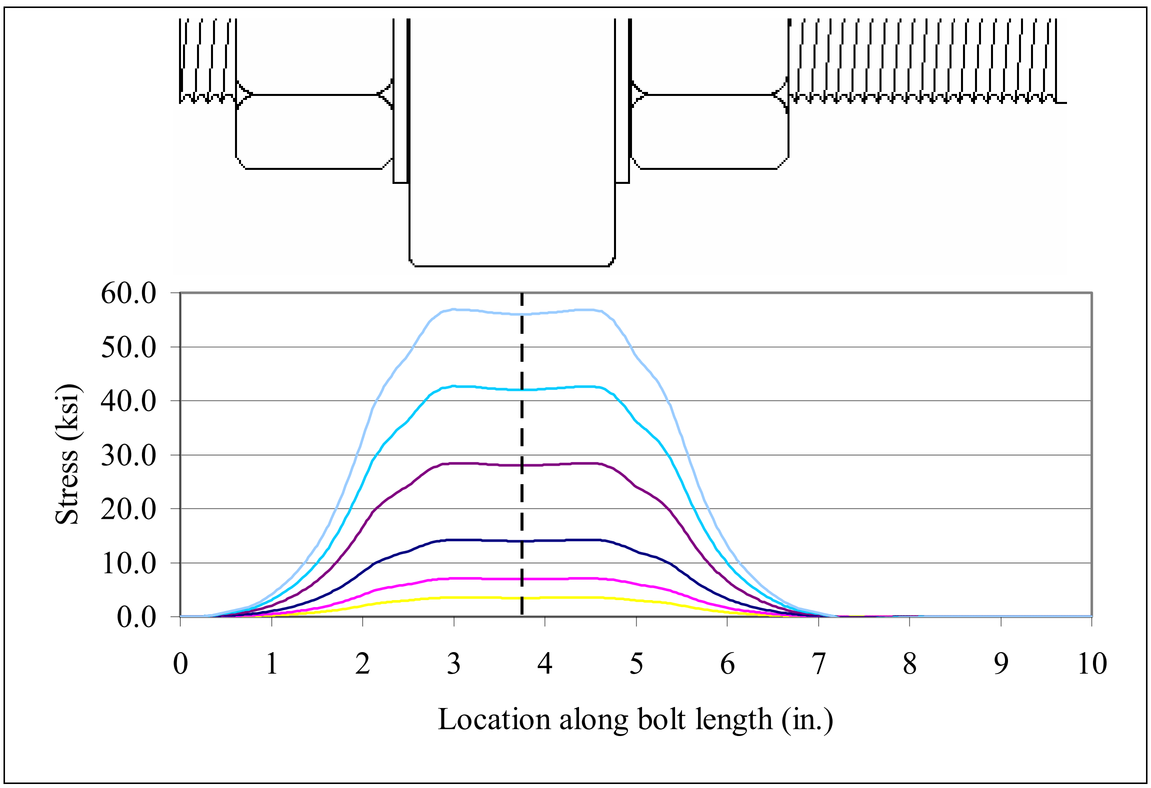

Figure 4.9 – FEM preload

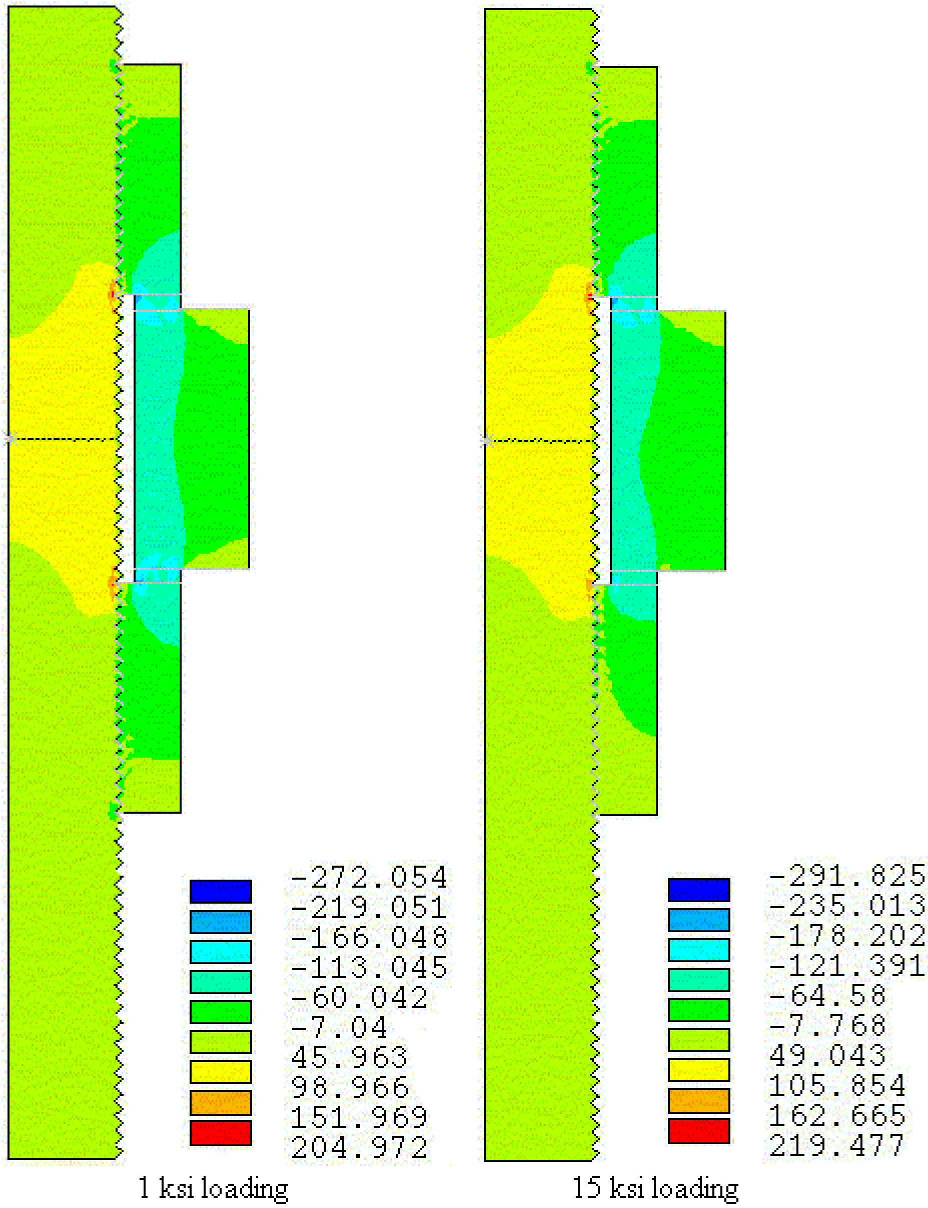

Figure 4.10 – FEM axial stress response to applied loads and maximum preload

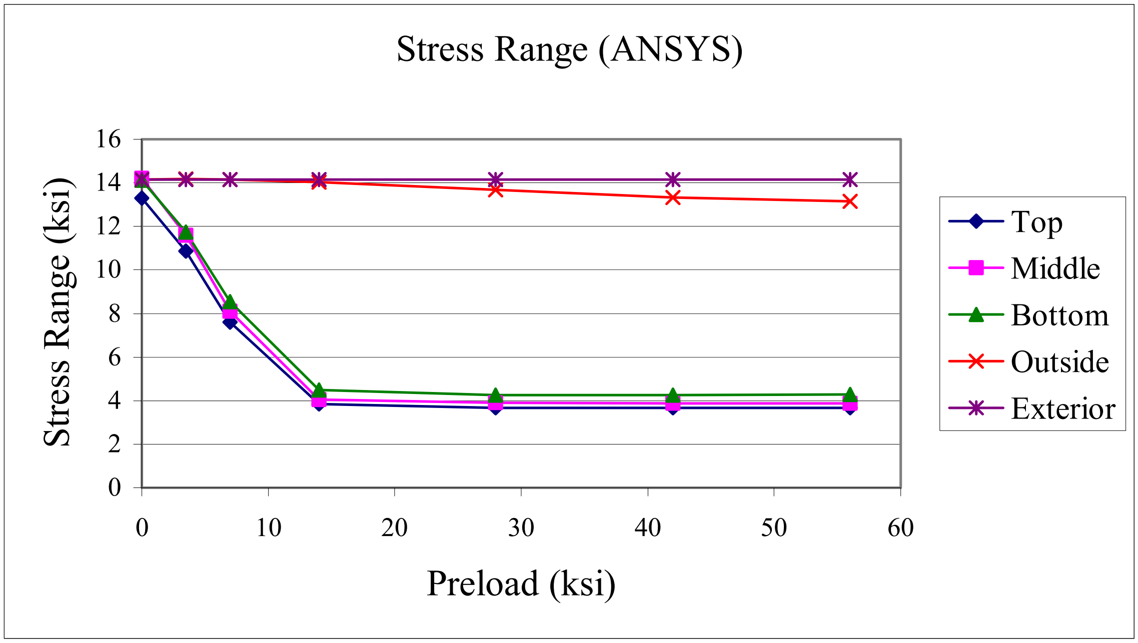

Figure 4.11 – FEM stress range vs. preload

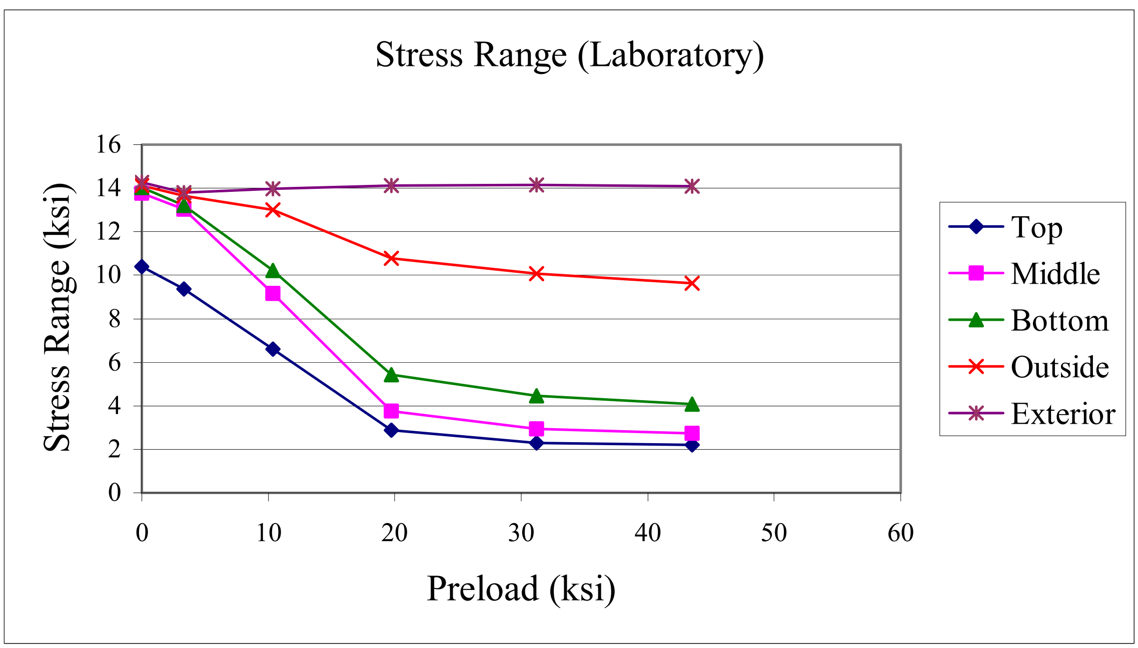

Figure 4.12 – Laboratory stress range vs. preload

4.3 Laboratory and FEM results comparison

It is shown by comparison that the separation of the stress ranges is present even in the idealized control of the finite element model. Examining plots of experienced stress levels in the finite element model can show the reason for the variance in data. Figure 4.13 shows a comparison of the single nut, zero preload condition with the maximum preload condition examined in the finite element analysis.

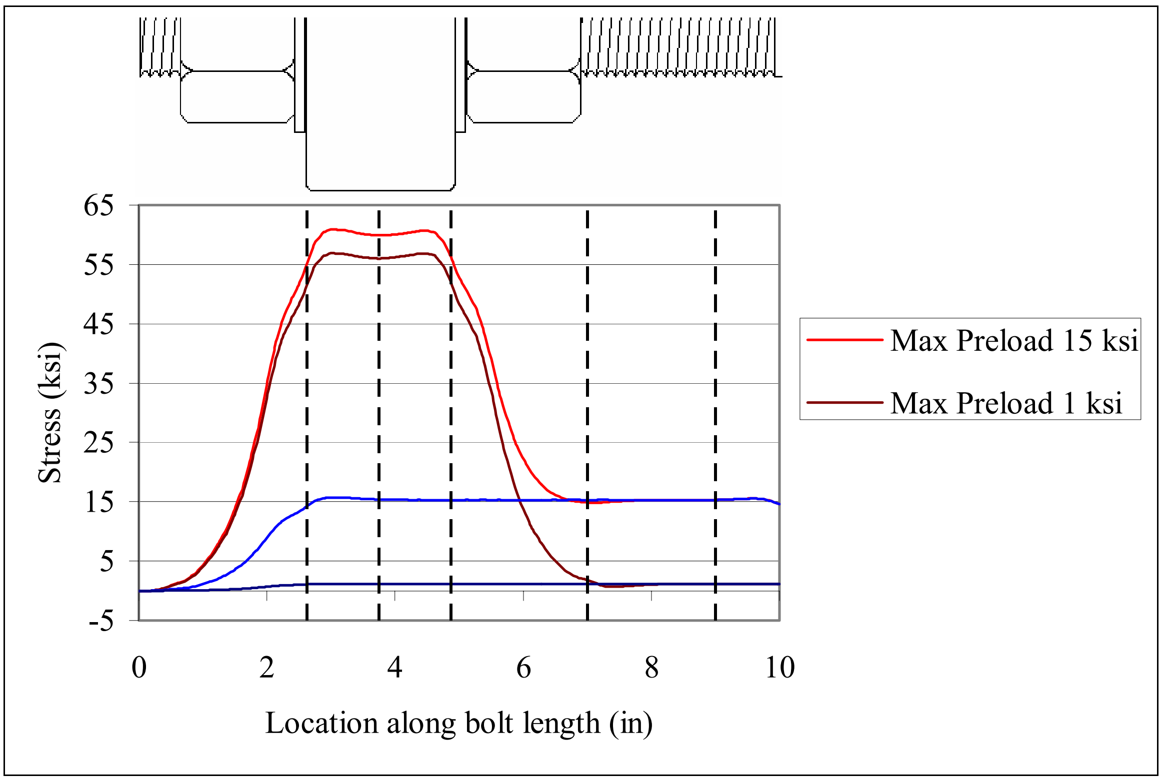

Figure 4.13 – Stress comparison

4.3.1 Outside stress range drop-off

The outside gages are at a location where one would expect to see the same stress levels as the exterior gage since at the end of the bottom nut the bolt should be free of preload and subject to the same stresses as the bolt shank. However, in both the FEM and laboratory data a distinct drop-off can be seen with increasing preload. As shown in Figure 4.14, the introduction of preload into the bolt sees an influence beyond the end of the bottom nut. This influence increases with the magnitude of the preload. The stress range fall-off is actually seen from both the 1 ksi stress side and the 15 ksi stress side as stress at the outside location stress increases and decreases, respectively. This causes a significant decrease in stress range immediately outside the bottom nut with increasing preload.

This behavior may help to explain why failure outside the connection does not always occur at the bottom nut interface, but often in the first several threads below the bottom nut as observed in Frank (1980) and Van Dien et al. (1996). It seems that the largest stress range may actually occur at a length within an inch of the end of the bottom nut, the exact location dependant on the amount of preload present and the degree of applied stress range.

The falloff seen in the finite element model is substantially less than in the laboratory tests. It can be assumed that the idealization of the bolt behavior for the finite element model is responsible for the more ordered results. The model does not capture many of the nuances of the connection including residual stresses from friction and tightening, and the tolerance between nut and bolt threads. The finite element model assumes perfect fit between nut and bolt.

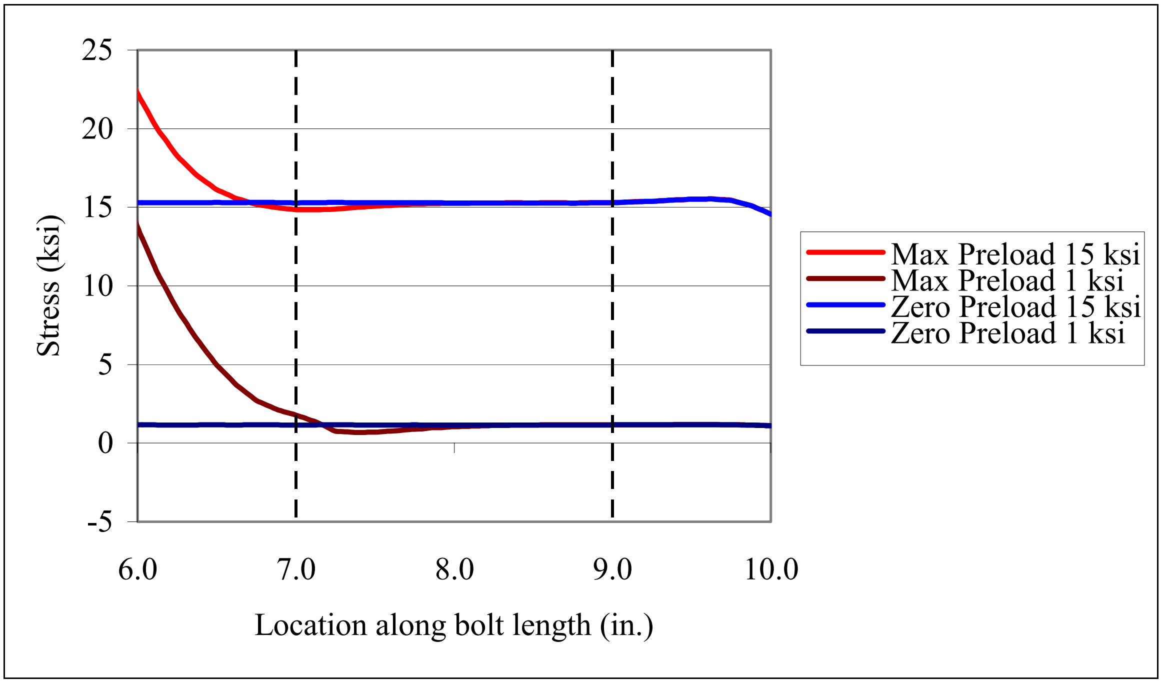

Figure 4.14 – Outside and exterior gage examination

4.3.2 Top and bottom stress range separation from middle

In both the laboratory and finite element tests, the stress range at the top gage location is always lower than at the middle location for all levels of preload. Similarly, stress range at the bottom location is higher than that at the middle. These variances were identified in Section 3 for the laboratory data. Figure 4.15 shows the response of the bolt to applied loads for both a preloaded and non-preloaded case.

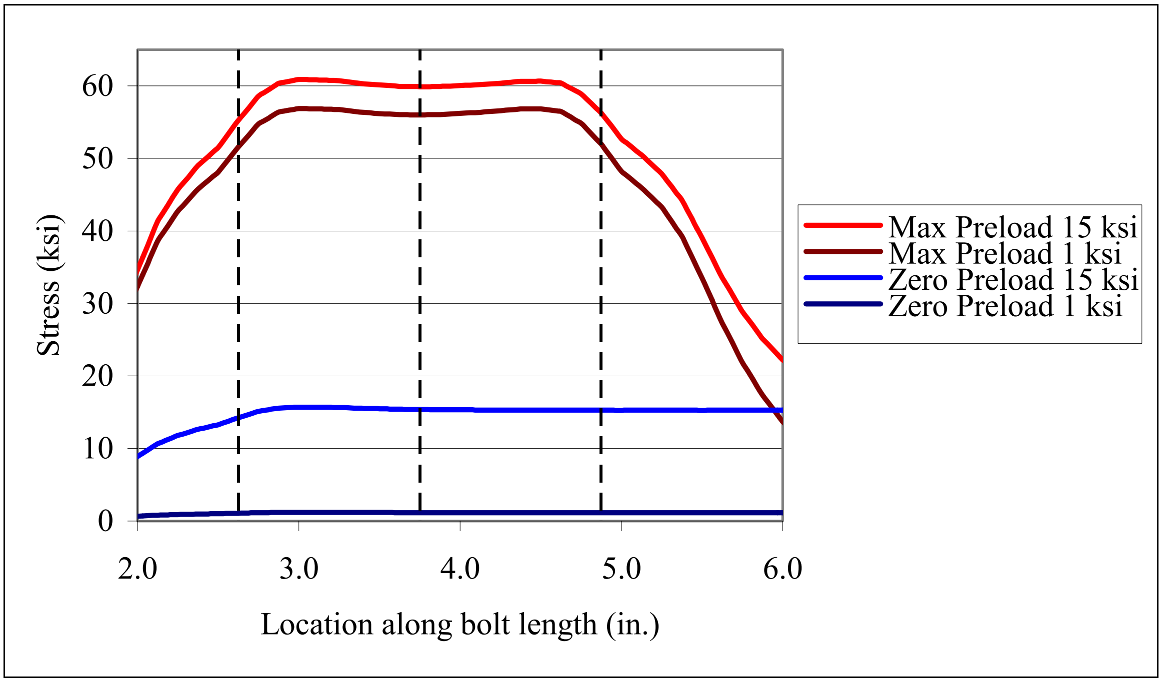

Figure 4.15 – Top and bottom gage examination

For the case of zero preload, a ramp-up of load can be seen at the location of the top gages at the depth that the gages were placed. This seems unusual, as the threads at that same location are seeing the highest tensile stresses in the entire bolt, even in a preloaded connection as can be seen in Figure 4.10 above. To evaluate the validity of all stress range values, further finite element analysis was carried out to inspect the possible connection of gage depth with measured stress range.

4.3.3 Effects of gage depth

Recalling the methods discussed in Section 2, gage depths were selected based on those used in previous testing and allowance for wiring required for the strain gages.

When finite element analysis was carried out to evaluate stresses closer to the surface of the bolt, much different stress ranges were found. Where at the gage depth the stress range at the top location was always significantly lower than at the middle location, when looking only 1/8 inch closer to the surface the opposite was true. Likewise, at a location 1/8 inch closer to the surface, the stress range at the bottom location went from being greater than at the middle location to being less. These dramatic changes raise a serious question: are predictions of the migration of failure location given by the lab data valid?

To answer this question, further results were collected from the model. Readings were taken at three depths: 0.44 inches (gage depth), 0.32 inches, and 0.20 inches below the outer diameter of the bolt. These values correspond to nodes and elements convenient to analyze on the model. At the minimum level of preload tested, roughly 3.5 ksi, analysis at shallower depths indicated that a migration in failure location was not likely to occur. Laboratory data suggested that such a shift would occur at that low a preload value. Stress in the top location was still much greater than in the outside location. This can be seen in Figure 4.16.

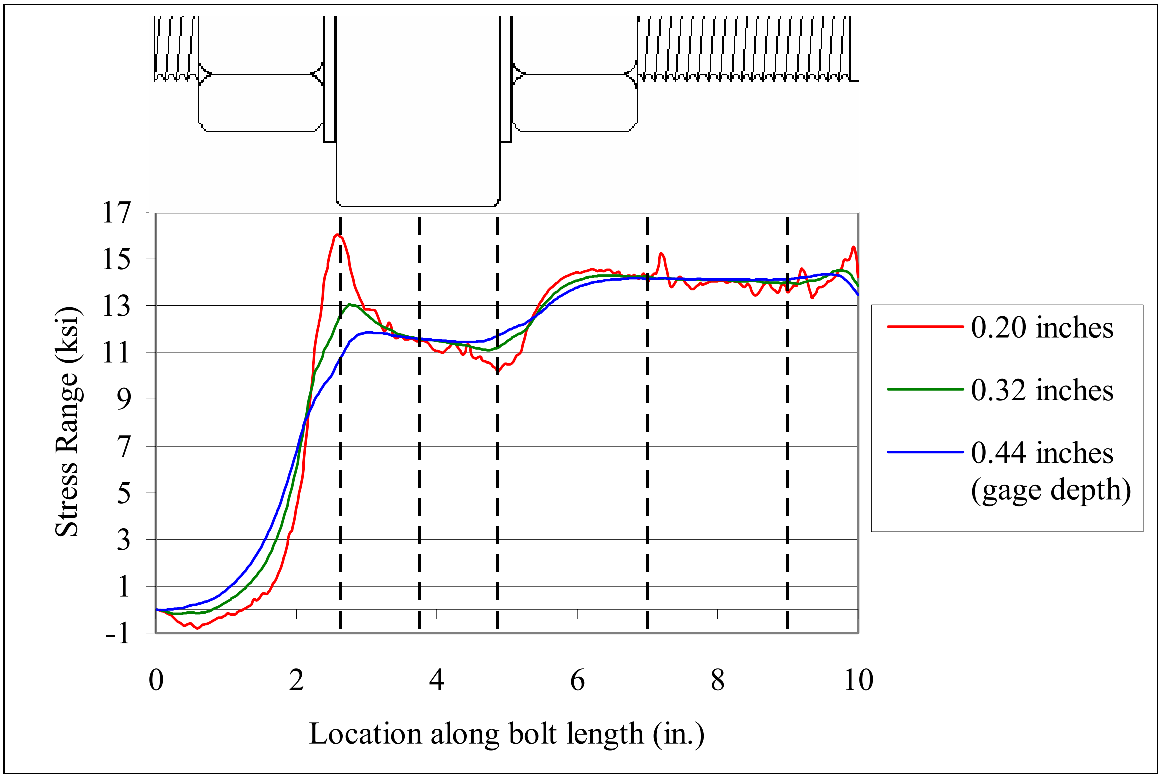

Figure 4.16 – Stress range for 3.5 ksi preload

At the next level of preload, roughly 7 ksi, the top location stress range dropped considerably below the level seen at the outside for all depths, a change which suggests the likely migration of failure location to outside the double-nut connection. A preload of 50 percent of the experienced stress range has previously been established as a benchmark for enhanced fatigue performance by Abraham (1997). Seven ksi is 50 percent of the experienced stress range in the testing for this particular specimen.

A turn-of-the-nut of 1/24 gave preload values between 4.6 and 7.6 ksi as shown in Figure 3.4. This shows, when combined with the conclusions drawn above, that 1/24 turn-of-the-nut is not a consistently valid standard for enhancement of fatigue performance. This is in line with the limited fatigue verification of the 1/24 condition seen in the lab. A turn-of-the-nut of 1/12 generated between 9.7 and 17.3 ksi preload, suggesting that it would make a more reliable standard.

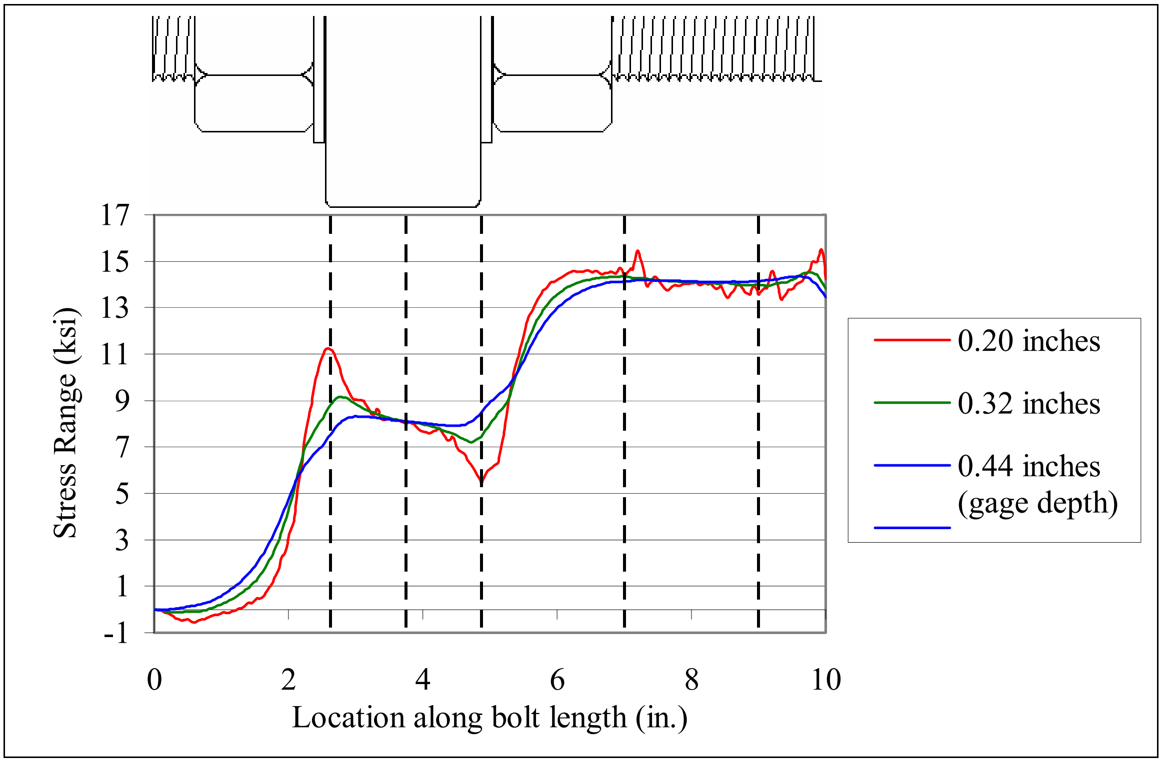

Figure 4.17 – Stress range for 7 ksi preload

Figure 4.16 and Figure 4.17 also illustrate a number of other important facts. Preload measured in the lab by the middle gage is constant, despite the gage depth issue that affects the top and bottom locations. The same is true for the outside and exterior locations. So, while readings at the top and bottom gage locations may vary with the gage depth, the preload reading at the middle location is valid for all measured values, as are the values of stress collected at the outside and exterior gages.

5 CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

In both the laboratory testing and finite element evaluation of double-nut anchor bolt assemblies, significant data was gathered over the course of the study. A number of conclusions were drawn regarding the turn-of-the-nut generation of preload, the accuracy of laboratory and finite element testing, and the generation of a tightening standard for all bolt tested bolt diameters.

5.1.1 Regarding the turn-of-the-nut generation of preload

- Impact tightening of double-nut connections using a sledgehammer and slug wrench is capable of generating significant preload in anchor bolts, to the point of achieving adequate fatigue performance of the assembly.
- Specified tightness of 1/6 turn-of-the-nut is a reasonable standard through the hammer tightening method for all diameter-thread pitch combinations. The 2 inch UNC test case reaches its feasible limit slightly beyond 1/8 turn-of-the-nut.
- UNC thread pitches generate more preload for a given turn-of-the-nut than an 8UN bolt of the same diameter.
- Less preload is generally generated with increasing base-plate thickness (as specified for increasing diameter) within a thread pitch category (UNC or

8UN) for a given turn-of-the-nut. This is especially true of bolts with the constant thread pitch of 8UN.

- Preload induced by turn-of-the-nut tightening reduces the stress range experienced by the portion of the bolt between the nuts during fatigue.
- Stress range reduction approaches a minimum level between 2.5 and 5.5 ksi with increasing tightness at a maximum of 1/6 turn-of-the-nut. This minimum does not appear to be a function of diameter, but holds for all tested specimens and thread pitches.

5.1.2 Regarding the accuracy of laboratory and finite element testing

- Finite element modeling can be used to capture important behaviors in bolt fatigue with relative accuracy, allowing for the design of laboratory tests to confirm results.
- When strain gaging threaded specimens, the surface provided for gage attachment should be as close to the outer diameter as possible, as the depth in the bolt from which readings are taken significantly impacts results.

5.1.3 Regarding the proposed uniform tightening standard

- Preloading a double-nut anchor bolt 1/12 turn-of-the-nut will improve the performance of the anchor bolt by the reduction of stress ranges between the nuts. This conclusion holds for the diameters specified in Section 5.

- Tightening to 1/6 turn-of-the-nut will not cause yielding across bolt cross-sections for any tested bolt greater than one inch in diameter. Stresses experienced in the initial threads involved in the bolt-to-nut connection are likely to exceed yield stress.

5.2 Standard recommendation

The data collected for the current study suggests a minimum tightening standard of 1/12 turn-of-the-nut to achieve sufficient fatigue performance. This value is supported by both laboratory and finite element results. An increase in minimum degree of tightness to 1/6 turn-of-the-nut past a snug tight condition, or refusal of the connection to further tightening is recommended. This increase provides for an easier reference point on the nut as well as a factor of safety. Refusal should only be accepted as a limit if the degree of tightness is at least 1/12 turn-of-the-nut. This standard is recommended for UNC bolts from 1 to 2-1/4 inches, and 8UN bolts from 1 to 2 inches.

5.3 Recommendations for continued research

The research conducted for the current study raises a number of additional questions. Further testing should be carried out for both verification and continuation of this research.

5.3.1 Standard verification

First and foremost, it is recommended that lab verification of fatigue performance at 1/12 turn-of-the-nut be conducted. Fatigue testing of a number of specimens of critical diameter-thread pitch combination (2 inch 8UN) that results in failure outside the double-nut connection would reinforce results found in the current study.

For best verification, tests should be carried out on bolts without grooves or other machining for the affixing of strain gages. This most closely simulates field conditions and mirrors the data gathered from finite element modeling, which did not include grooving. As in the progressive tightening tests, fatigue loads should be applied so to generate a minimum of 1 ksi stress in the bolt and a maximum of 15 ksi. Failure of the bolt below the bottom nut at such loading would indicate improved fatigue performance. Standards for acceptable fatigue performance and details on fatigue analysis of this and other diameters of anchor bolt may be found in Keating et al. (2004).

5.3.2 Standard refinement

If it is desirable to the Texas Department of Transportation to lower the turn-of-the-nut standard of tightening for some or all bolt diameters, or if the standard is to be applied to bolts beyond the specified diameters, finite element analysis and verification fatigue tests should be carried out. Finite element analysis should be performed for any diameter for which the standard is to be altered or extended. A model such as the one discussed in Section 4 has been shown to be an accurate predictor of stress range response to preload. For the calibration of such models, stress data from the current study may be used in the manner shown in Figure 4.6. From that point stress ranges may be evaluated as in Section 4, and a new standard developed.

Any altered or extended standard should also include a fatigue verification as described above. This practical laboratory exercise is vital for the establishment of confidence in recommendations.

This same procedure and test plan may be used to examine the effects of improper tightening of bolts. Cases where insufficient preload is developed can be studied through finite element analysis with practical confirmation tests performed in the laboratory.

5.3.3 2 inch UNC turn-of-the-nut feasibility study

Given the behavior of the 2 inch UNC test case in regards to the feasible limit of turn-of-the-nut tightening, further investigation of that diameter may be warranted. Further tightening tests may be carried out as described in Section 2 to evaluate the levels of preload found in the bolt for various degrees of turn-of-the-nut. Further tests should verify or refute the observed difficulty in generating 1/6 turn-of-the-nut preload for 2 inch UNC bolts. If the problem is consistent, the standard for this test case may need revision to be practically implemented in the field. As a part of any 2 inch UNC tightening verification, further tests should also be carried out on the 1-3/4 inch and 2-1/4 inch bolts to bound the data and further examine the feasibility of attaining the given standard with those bolts as well.

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APPENDIX A

PROGRESSIVE TIGHTENING DATA

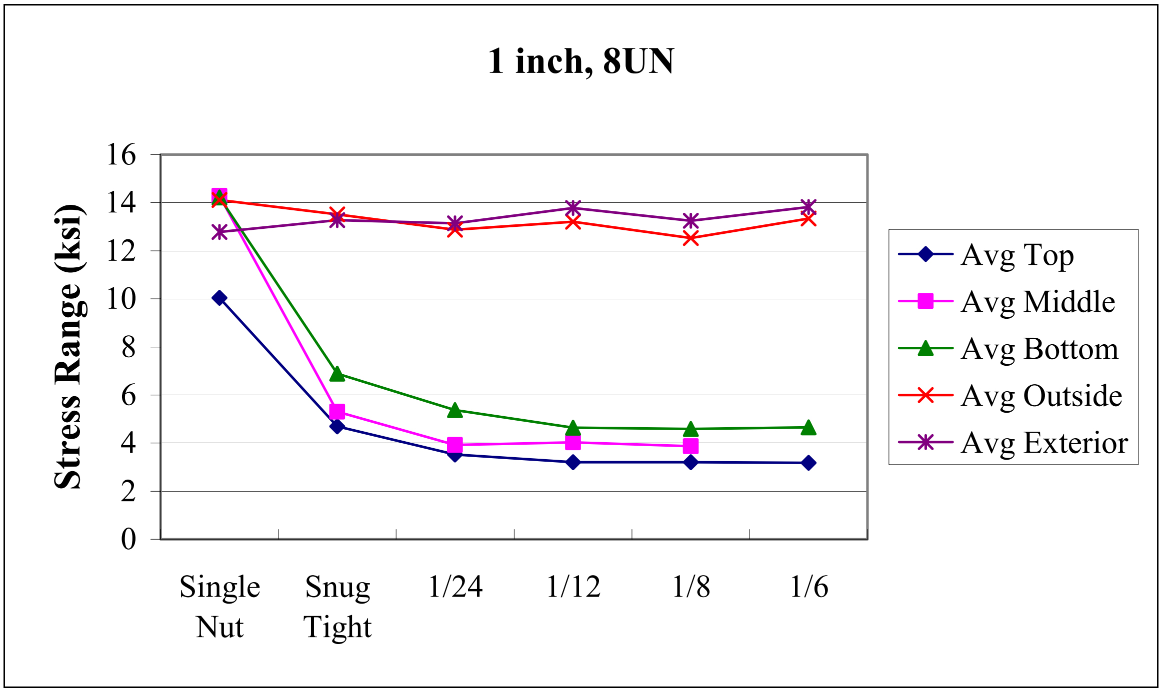

Figure A.1 – 1 inch, 8UN progressive tightening plot

Table A.1 – 1 inch, 8UN progressive tightening table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	0.3	23.2	1.6	24.3	24.8	63.9	Lost Gage		
Adjusted	---	---	---	---	20.2	61.5	Lost Gage		

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.0	---	---	4.7	3.5	3.2	3.2	3.2	---
Middle	14.3	---	---	5.3	3.9	4.0	3.9	Lost Gage	
Bottom	14.2	---	---	6.9	5.4	4.6	4.6	4.7	---
Outside	14.1	---	---	13.5	12.9	13.2	12.5	13.3	---
Exterior	12.8	---	---	13.3	13.1	13.8	13.2	13.8	---

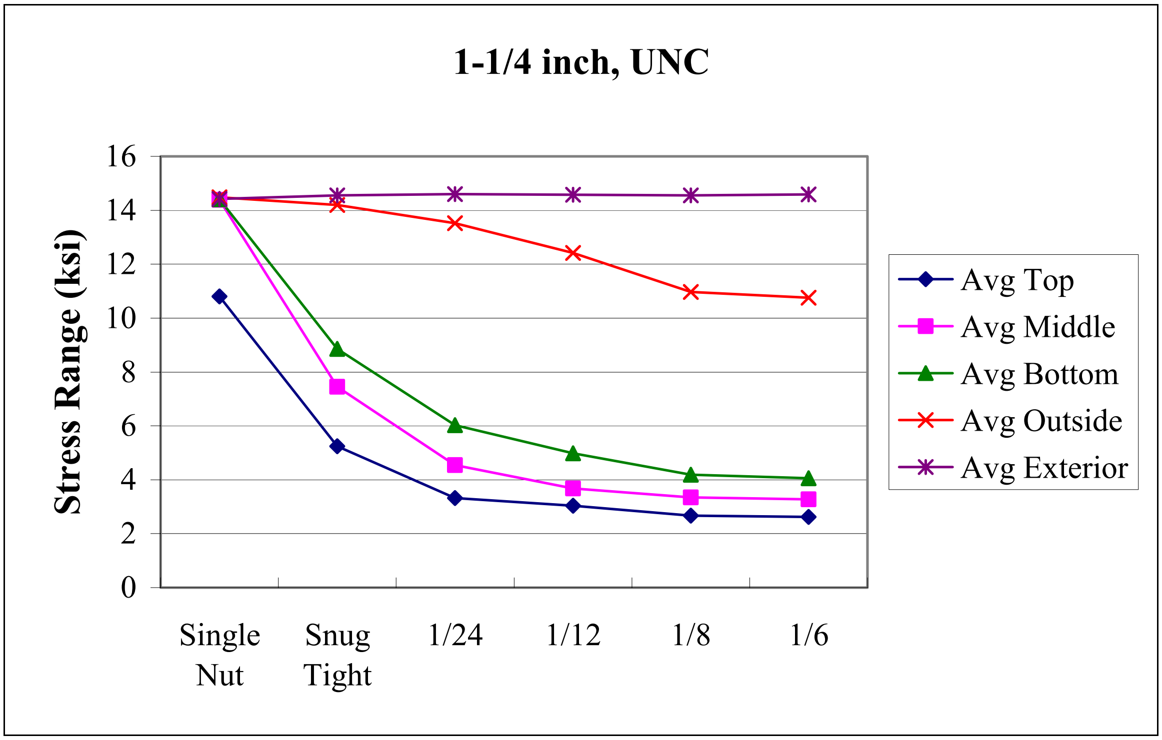

Figure A.2 – 1-1/4 inch, UNC progressive tightening plot

Table A.2 – 1-1/4 inch, UNC progressive tightening table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	-1.4	13.9	-2.1	11.1	27.3	50.6	73.2	82.2	22.8
Adjusted	---	---	---	---	24.7	44.3	63.4	70.6	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.8	---	---	5.2	3.3	3.0	2.7	2.6	---
Middle	14.4	---	---	7.4	4.5	3.7	3.3	3.3	---
Bottom	14.4	---	---	8.9	6.0	5.0	4.2	4.1	---
Outside	14.5	---	---	14.2	13.5	12.4	11.0	10.8	---
Exterior	14.4	---	---	14.5	14.6	14.6	14.5	14.6	---

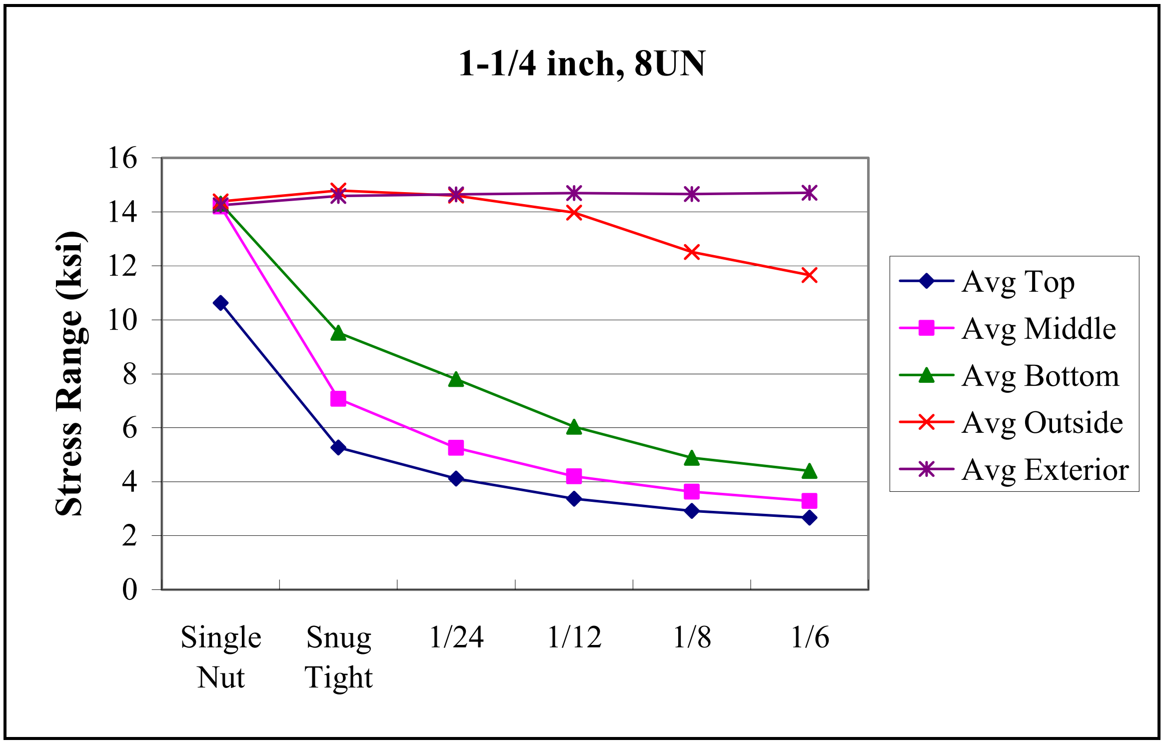

Figure A.3 – 1-1/4 inch, 8UN progressive tightening plot

Table A.3 – 1-1/4 inch, 8UN progressive tightening table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	-0.2	12.0	-1.0	12.0	25.4	44.9	71.6	92.4	6.1
Adjusted	---	---	---	---	23.3	39.9	62.8	80.6	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.6	---	---	5.3	4.1	3.4	2.9	2.7	---
Middle	14.2	---	---	7.1	5.3	4.2	3.6	3.3	---
Bottom	14.3	---	---	9.5	7.8	6.0	4.9	4.4	---
Outside	14.4	---	---	14.8	14.6	14	12.5	11.7	---
Exterior	14.2	---	---	14.6	14.6	14.7	14.7	14.7	---

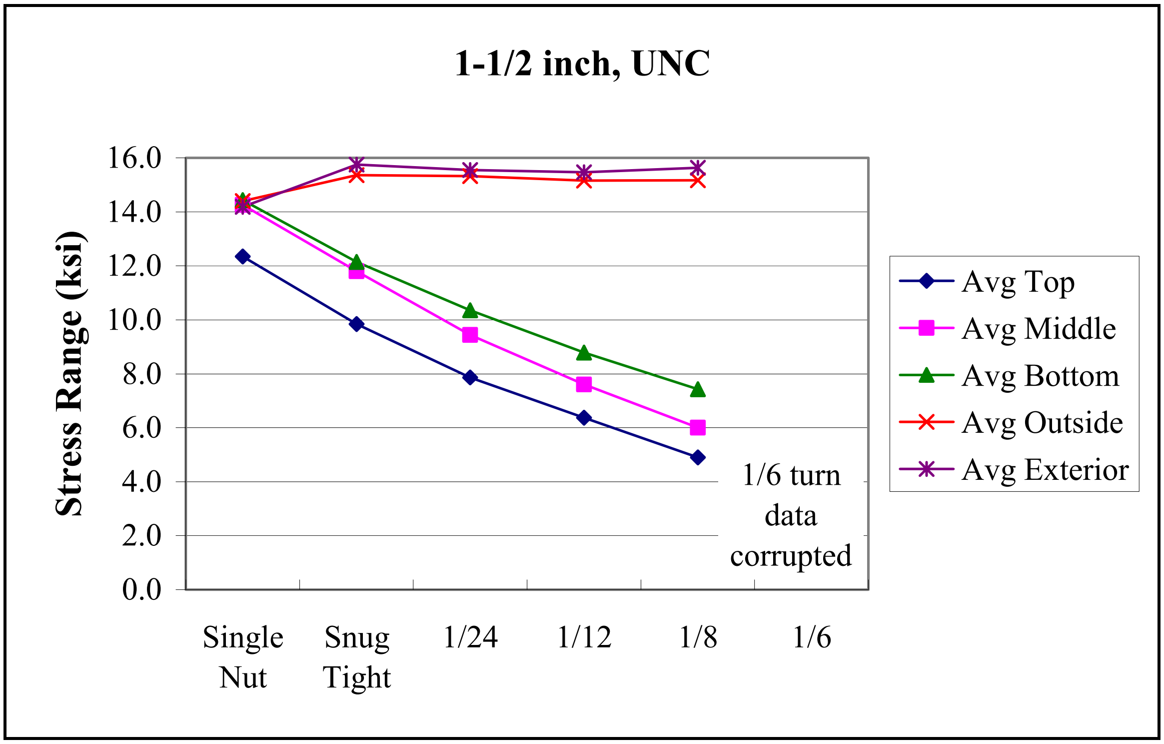

Figure A.4 – 1-1/2 inch, UNC progressive tightening plot

Table A.4 – 1-1/2 inch, UNC progressive tightening table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	-1.3	4.9	-1.4	5.4	11.4	14.1	17.7	19.0	2.1
Adjusted	---	---	---	---	10.6	12.9	15.9	17.1	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	12.3	---	---	9.8	7.9	6.4	4.9	File Corrupt	---
Middle	14.3	---	---	11.8	9.4	7.6	6		---
Bottom	14.4	---	---	12.1	10.4	8.8	7.4		---
Outside	14.4	---	---	15.4	15.3	15.2	15.2	No Data	---
Exterior	14.2	---	---	15.7	15.6	15.5	15.6		---

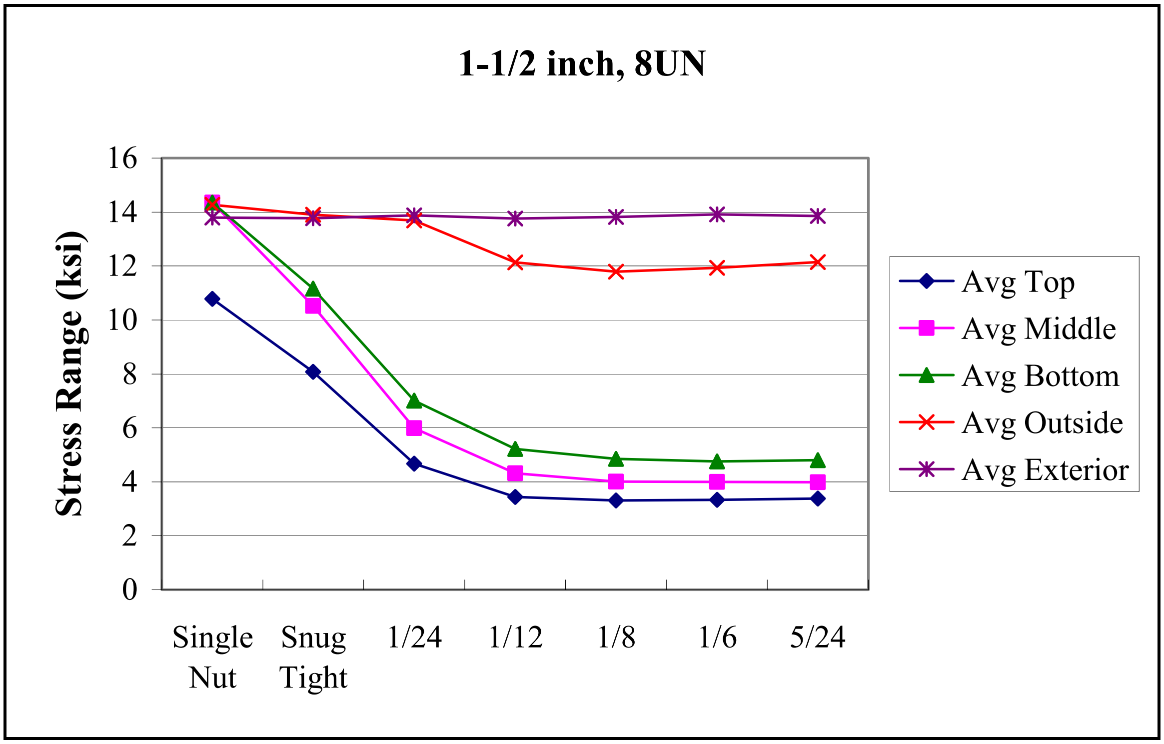

Figure A.5 – 1-1/2 inch, 8UN progressive tightening plot

Table A.5 – 1-1/2 inch, 8UN progressive tightening table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	0.1	6.4	0.1	6.3	17.8	37.3	46.6	47.1	1.7
Adjusted	---	---	---	---	16.7	34.3	42.6	42.9	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.8	---	---	8.1	4.7	3.4	3.3	3.3	---
Middle	14.4	---	---	10.5	6	4.3	4	4	---
Bottom	14.4	---	---	11.2	7	5.2	4.8	4.8	---
Outside	14.3	---	---	13.9	13.7	12.1	11.8	11.9	---
Exterior	13.8	---	---	13.8	13.9	13.8	13.8	13.9	---

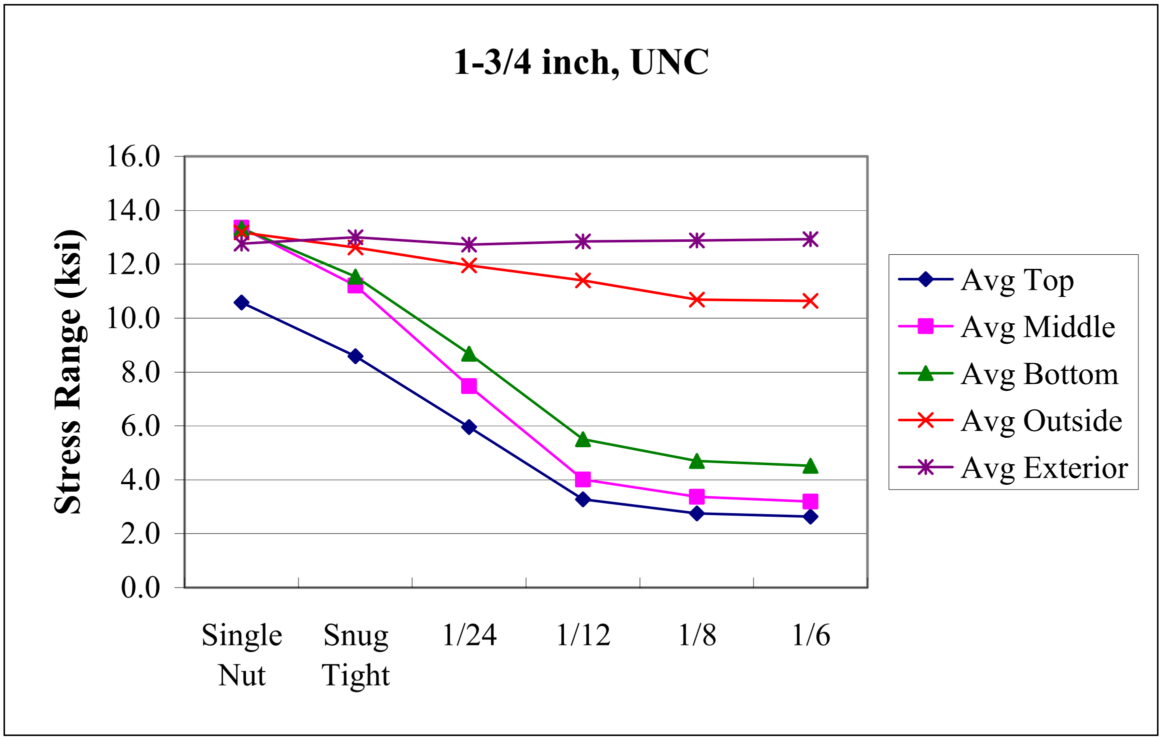

Figure A.6 – 1-3/4 inch, UNC progressive tightening plot

Table A.6 – 1-3/4 inch, UNC progressive tightening table

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	0.3	7.5	4.0	9.4	10.1	26.2	45.5	56.2	2.7
Adjusted	---	---	---	---	9.2	24.1	41.9	51.7	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.6	---	---	8.6	6.0	3.3	2.8	2.6	---
Middle	13.4	---	---	11.2	7.5	4.0	3.4	3.2	---
Bottom	13.3	---	---	11.5	8.7	5.5	4.7	4.5	---
Outside	13.2	---	---	12.6	12.0	11.4	10.7	10.6	---
Exterior	12.8	---	---	13.0	12.7	12.8	12.9	12.9	---

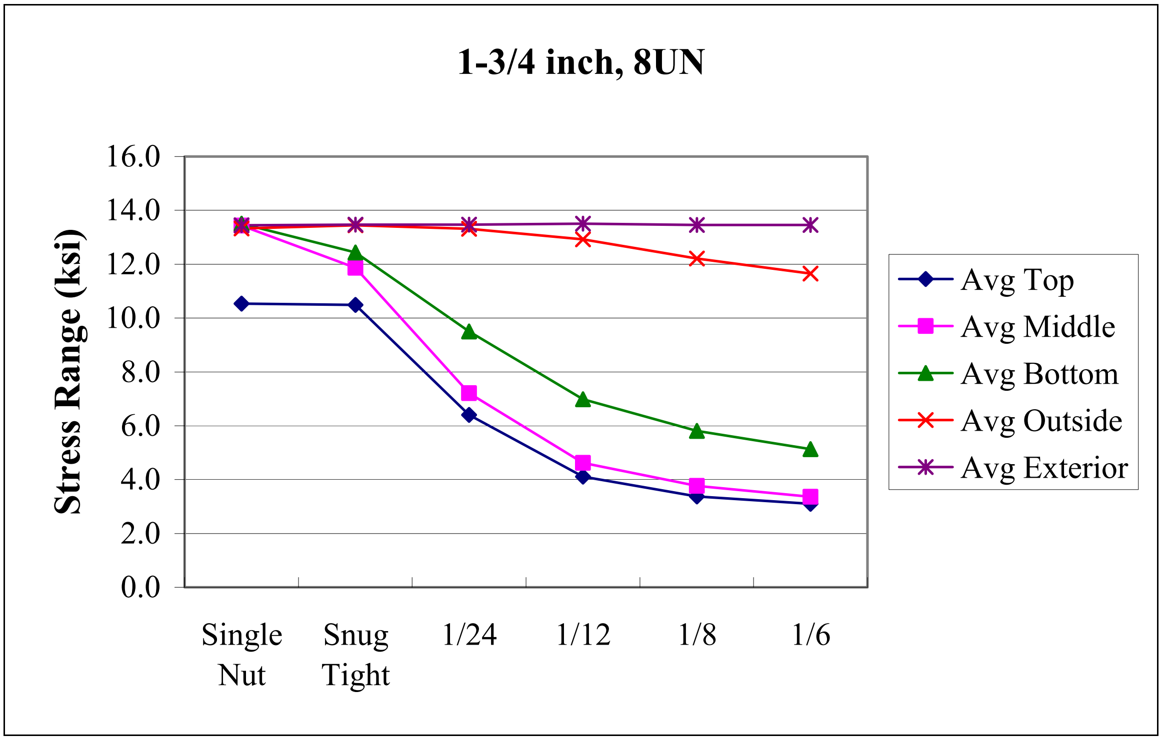

Figure A.7 – 1-3/4 inch, 8UN progressive tightening plot

Table A.7 – 1-3/4 inch, 8UN progressive tightening data

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	-0.4	4.1	-0.3	3.4	10.4	19.7	31.2	43.5	-1.5
Adjusted	---	---	---	---	9.6	17.9	28.2	39.2	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.5	---	---	10.5	6.4	4.1	3.4	3.1	---
Middle	13.4	---	---	11.9	7.2	4.6	3.8	3.4	---
Bottom	13.5	---	---	12.4	9.5	7.0	5.8	5.1	---
Outside	13.3	---	---	13.5	13.3	12.9	12.2	11.6	---
Exterior	13.4	---	---	13.5	13.5	13.5	13.5	13.5	---

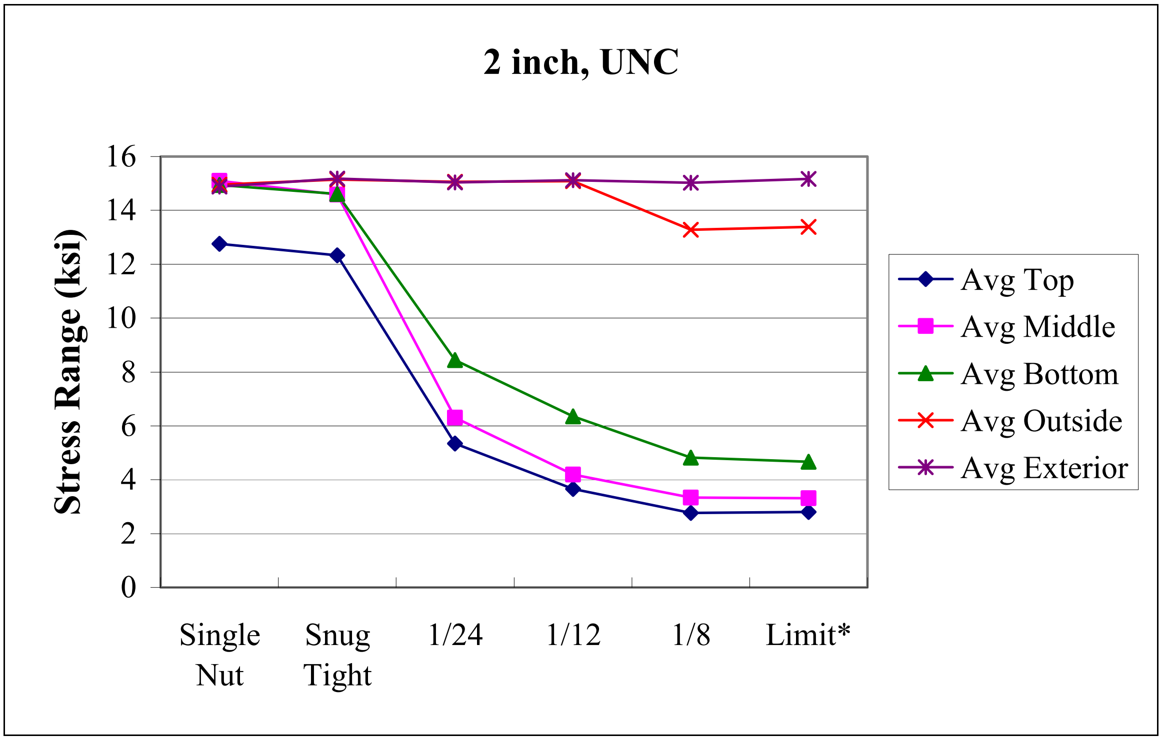

Figure A.8 – 2 inch, UNC progressive tightening plot

Table A.8 – 2 inch, UNC progressive tightening data

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	Limit*	Zero 3
Recorded	0.9	4.3	1.0	4.6	12.8	27.5	49.7	46.4	-9.5
Adjusted	---	---	---	---	12.0	25.2	45.2	42.2	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	Limit*	Zero 3
Top	12.8	---	---	12.3	5.3	3.7	2.8	2.8	---
Middle	15.1	---	---	14.6	6.3	4.2	3.3	3.3	---
Bottom	14.9	---	---	14.6	8.4	6.4	4.8	4.7	---
Outside	15.0	---	---	15.1	15.1	15.1	13.3	13.4	---
Exterior	14.9	---	---	15.2	15	15.1	15	15.2	---

*The feasible limit to tighten the given specimen was just slightly greater than 1/8 turn-of-the-nut

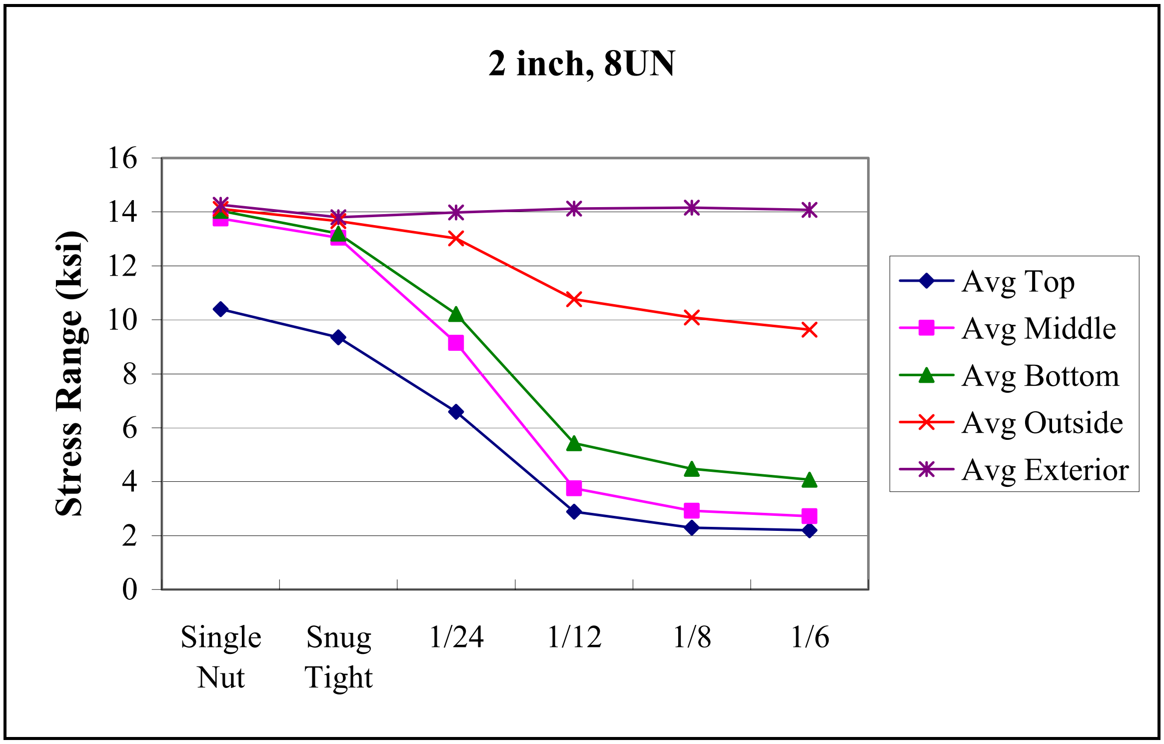

Figure A.9 – 2 inch, 8UN progressive tightening plot

Table A.9 – 2 inch, 8UN progressive tightening data

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	0.0	4.2	2.1	6.0	7.8	18.7	26.9	36.2	4.6
Adjusted	---	---	---	---	7.6	17.3	24.6	32.8	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.4	---	---	9.4	6.6	2.9	2.3	2.2	---
Middle	13.8	---	---	13	9.2	3.8	2.9	2.7	---
Bottom	14	---	---	13.2	10.2	5.4	4.5	4.1	---
Outside	14.1	---	---	13.7	13	10.8	10.1	9.6	---
Exterior	14.3	---	---	13.8	14	14.1	14.2	14.1	---

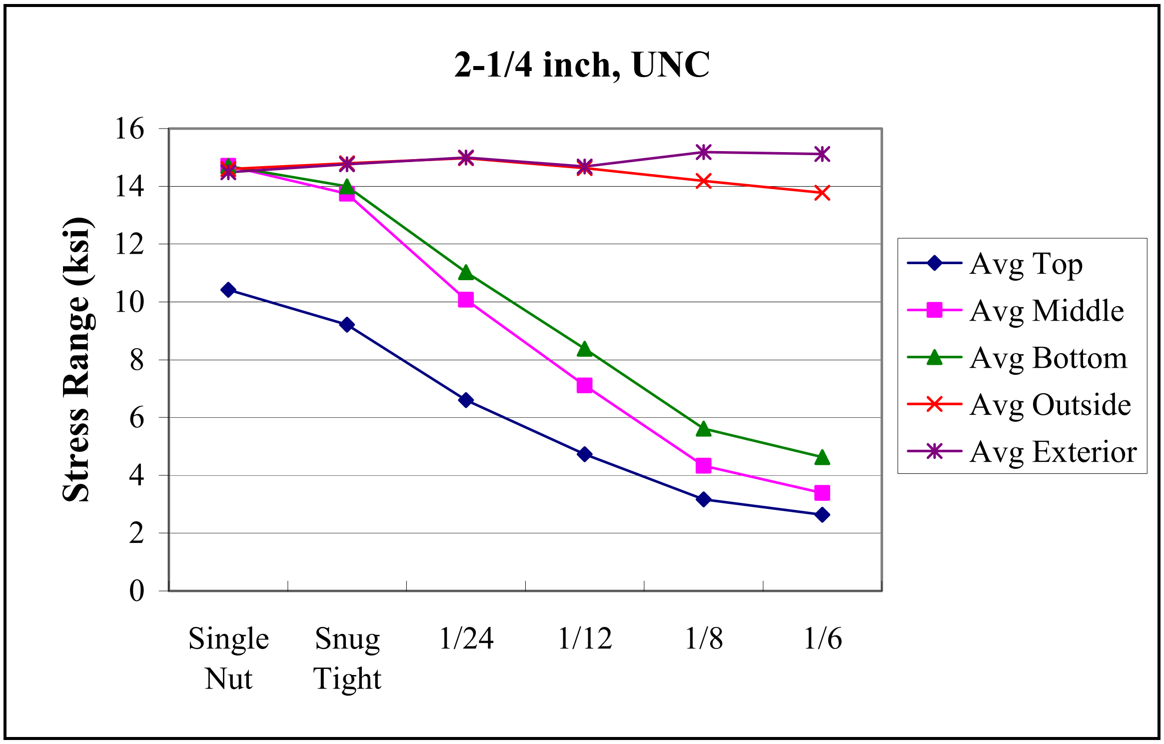

Figure A.10 – 2-1/4 inch, UNC progressive tightening plot

Table A.10 – 2-1/4 inch, UNC progressive tightening data

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	0.9	4.2	-0.6	2.2	7.1	12.4	24.3	35.0	-1.1
Adjusted	---	---	---	---	6.7	11.5	21.9	31.5	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	10.4	---	---	9.2	6.6	4.7	3.2	2.6	---
Middle	14.7	---	---	13.7	10.1	7.1	4.3	3.4	---
Bottom	14.7	---	---	14	11	8.4	5.6	4.6	---
Outside	14.6	---	---	14.8	15	14.6	14.2	13.8	---
Exterior	14.5	---	---	14.8	15	14.7	15.2	15.1	---

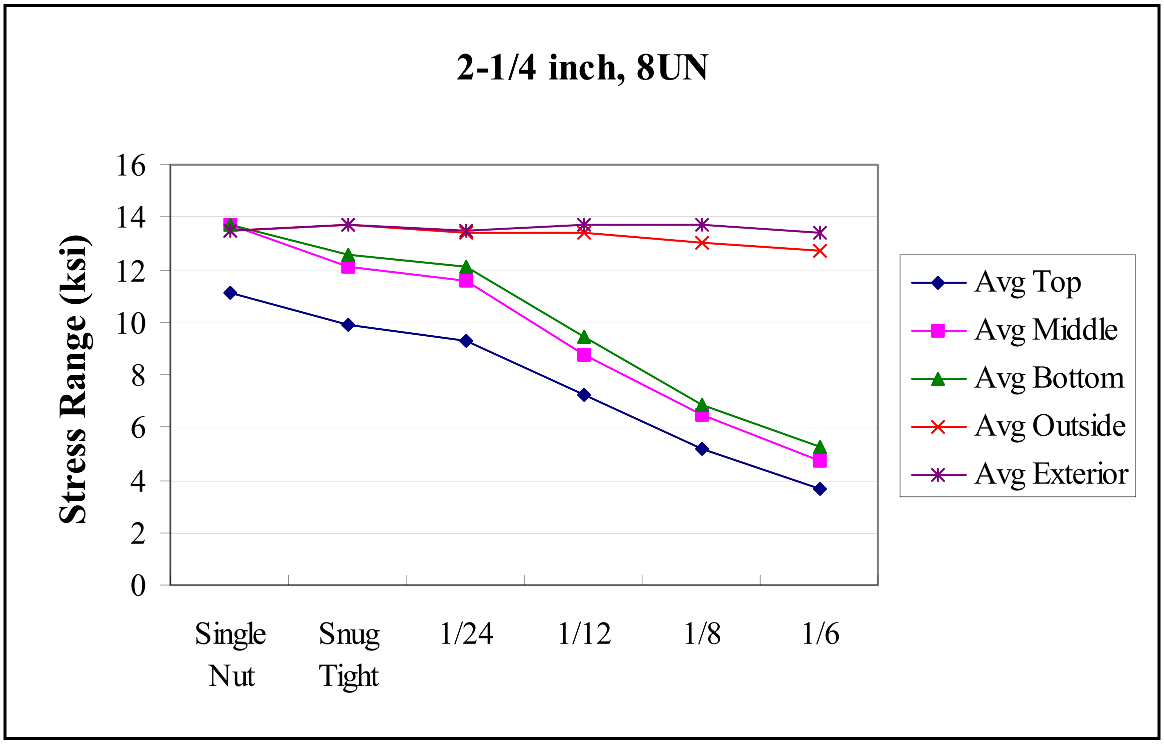

Figure A.11 – 2-1/4 inch, 8UN progressive tightening plot

Table A.11 – 2-1/4 inch, 8UN progressive tightening data

	Preload (ksi) per turn-of-the-nut								
	Zero 1	Snug 1	Zero 2	Snug 2	1/24	1/12	1/8	1/6	Zero 3
Recorded	-0.7	0.6	-0.8	0.8	1.5	5.2	8.5	15.3	-1.4
Adjusted	---	---	---	---	1.5	5.0	8.1	14.5	---

Gage Location	Stress range (ksi) per turn-of-the-nut								
	Single Nut	Snug 1	Zero 2	Snug Tight	1/24	1/12	1/8	1/6	Zero 3
Top	11.1	---	---	9.9	9.3	7.2	5.2	3.6	---
Middle	13.7	---	---	12.1	11.6	8.8	6.5	4.7	---
Bottom	13.7	---	---	12.5	12.1	9.4	6.9	5.3	---
Outside	13.5	---	---	13.7	13.4	13.4	13	12.7	---
Exterior	13.5	---	---	13.7	13.5	13.7	13.7	13.4	---

APPENDIX B

TIGHTENING DATA

B.1 UNC tightening data summaries

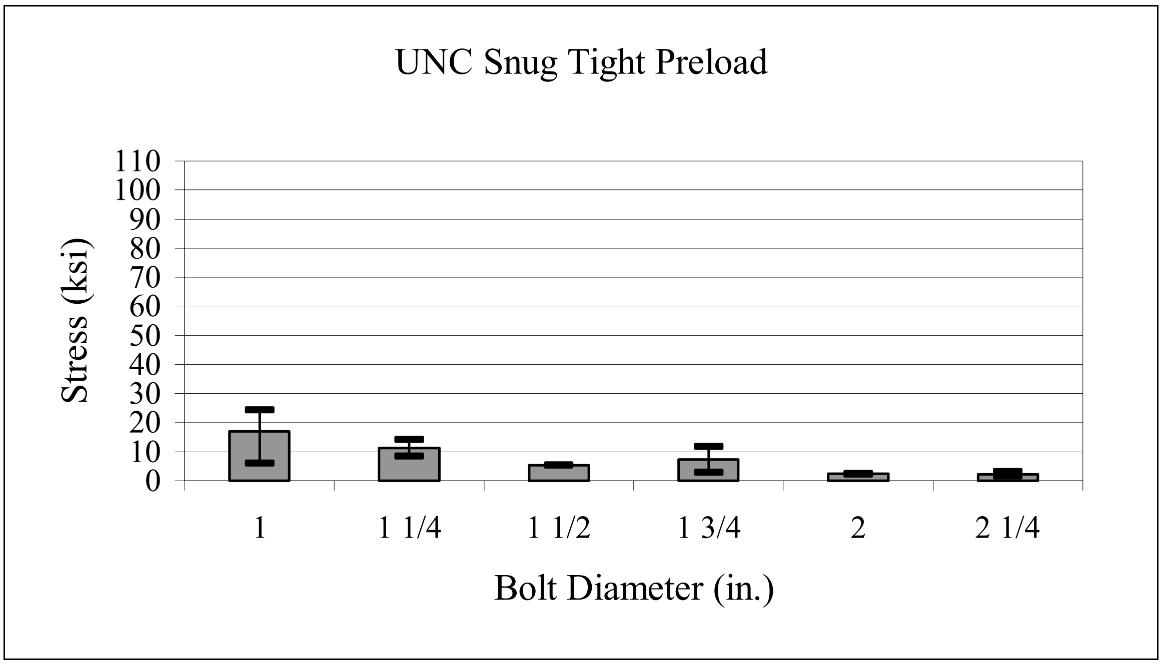

Figure B.1 – UNC snug tight preload

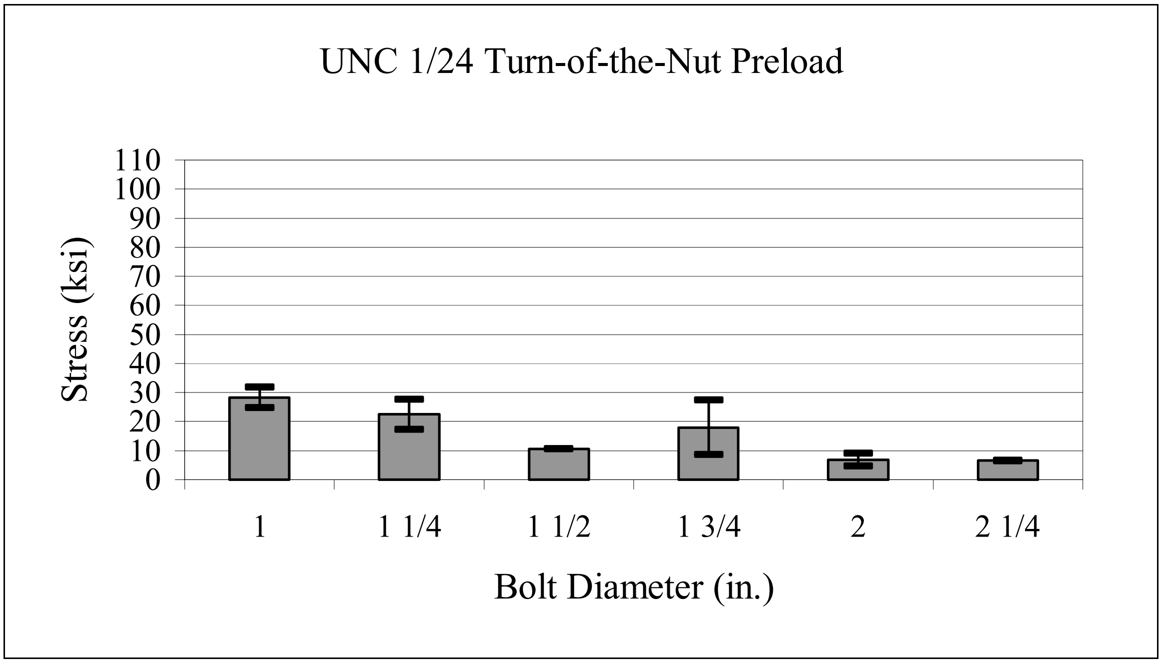

Figure B.2 – UNC 1/24 turn-of-the-nut preload

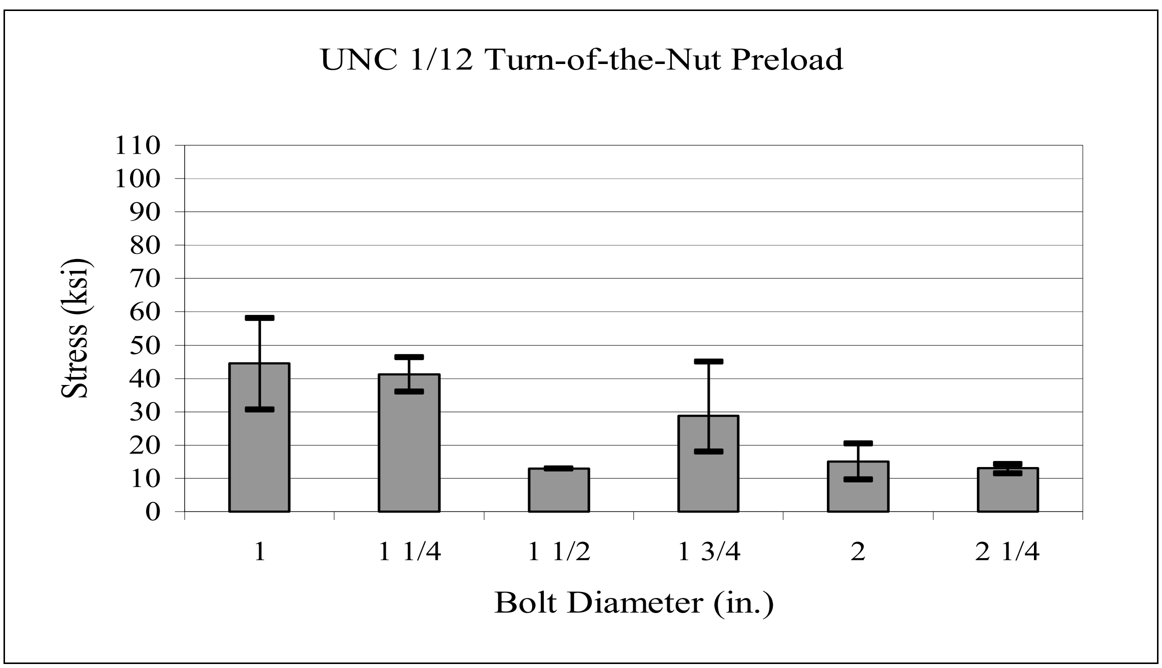

Figure B.3 – UNC 1/12 turn-of-the-nut preload

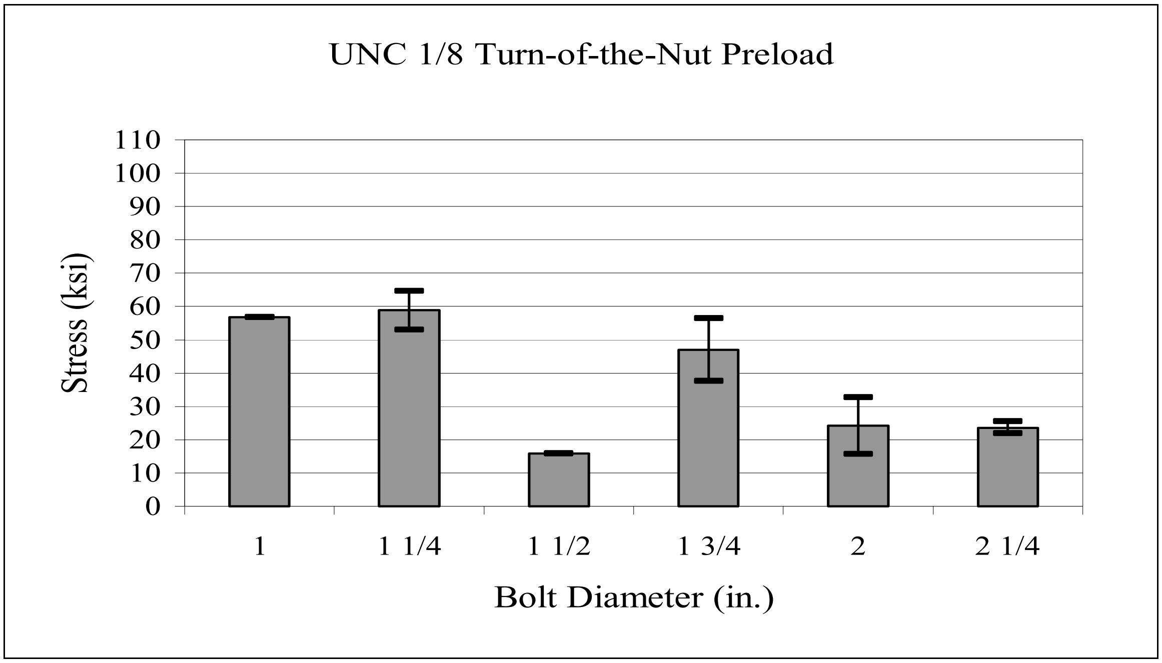

Figure B.4 – UNC 1/8 turn-of-the-nut preload

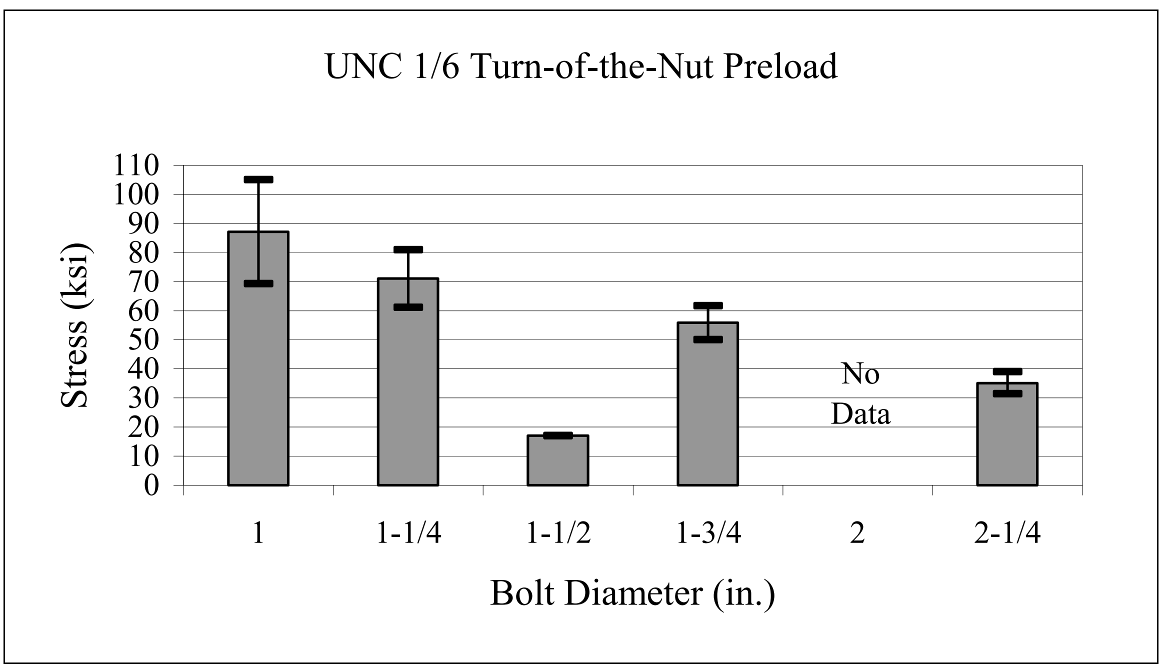

Figure B.5 – UNC 1/6 turn-of-the-nut preload

B.2 8UN tightening data summaries

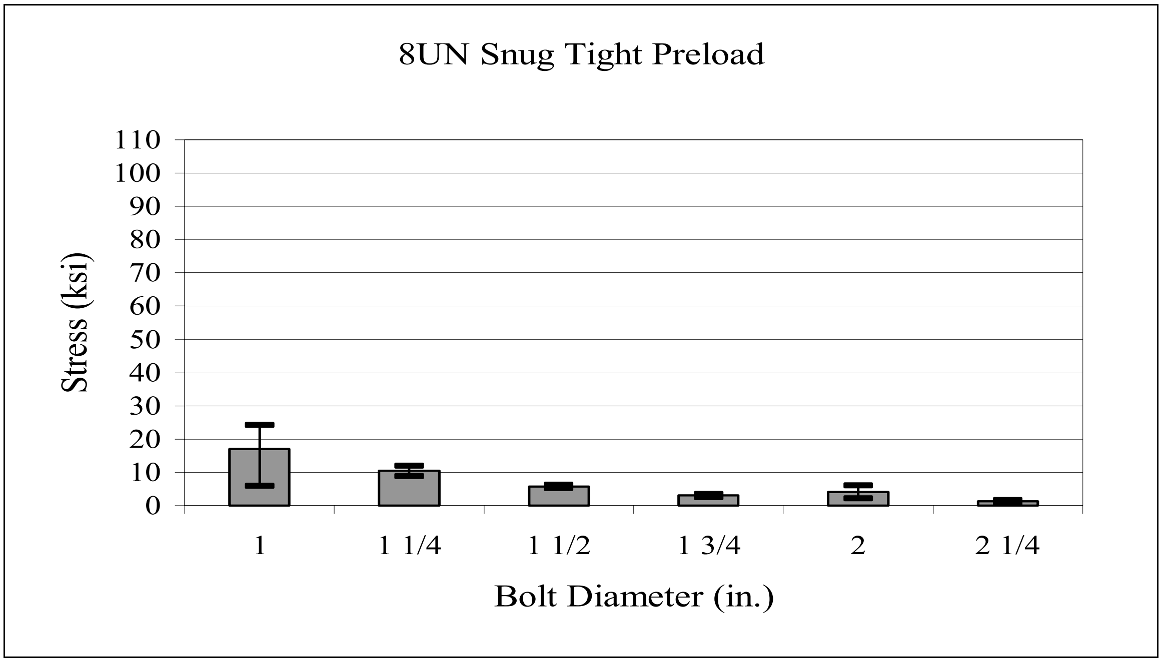

Figure B.6 – 8UN snug tight preload

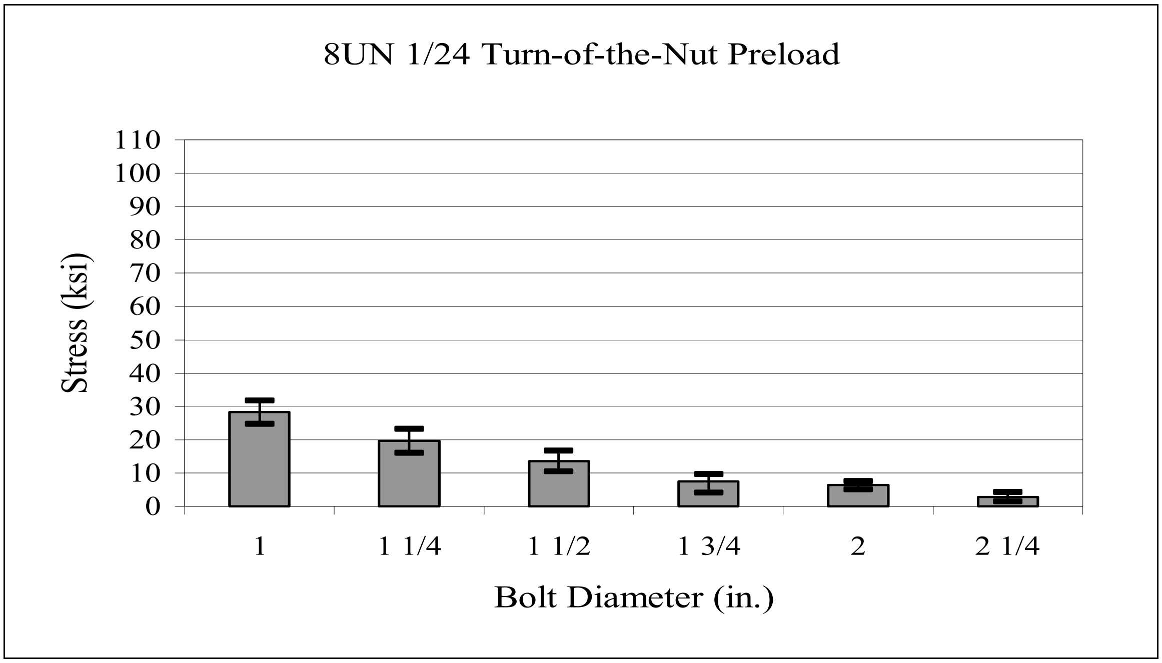

Figure B.7 – 8UN 1/24 turn-of-the-nut preload

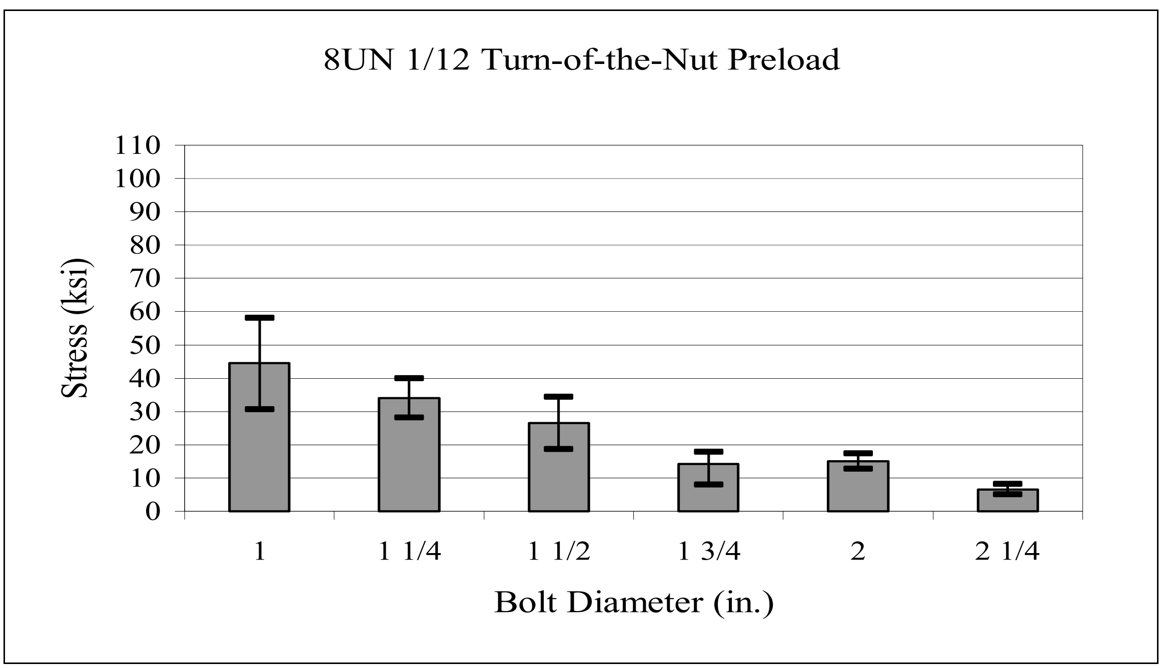

Figure B.8 – 8UN 1/12 turn-of-the-nut preload

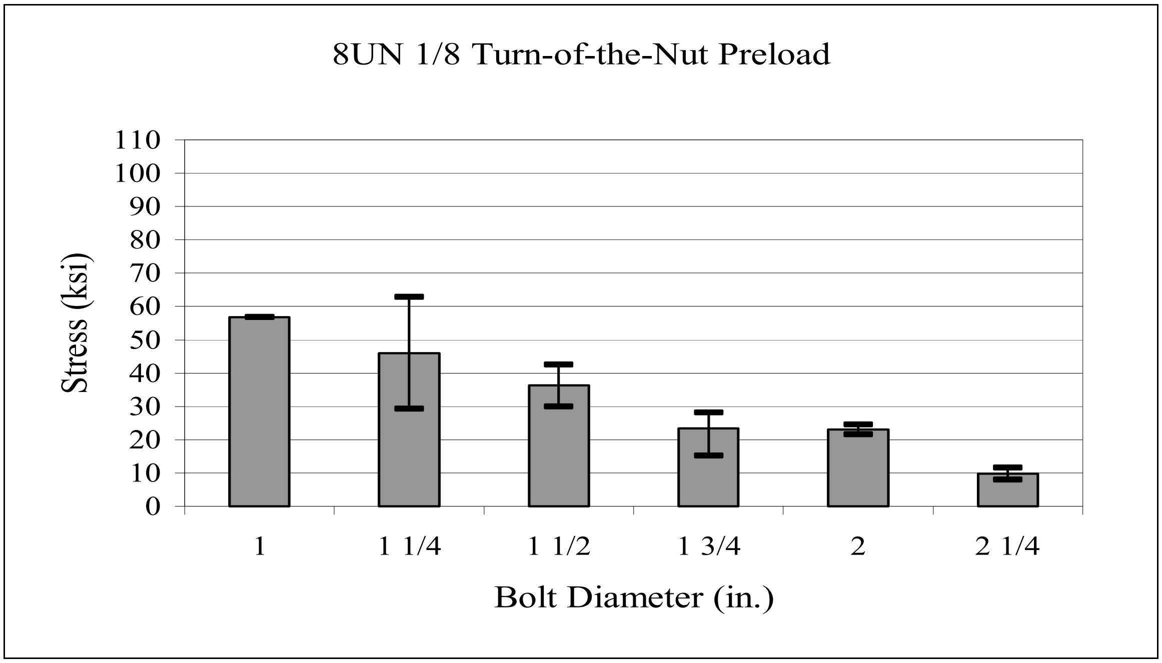

Figure B.9 – 8UN 1/8 turn-of-the-nut preload

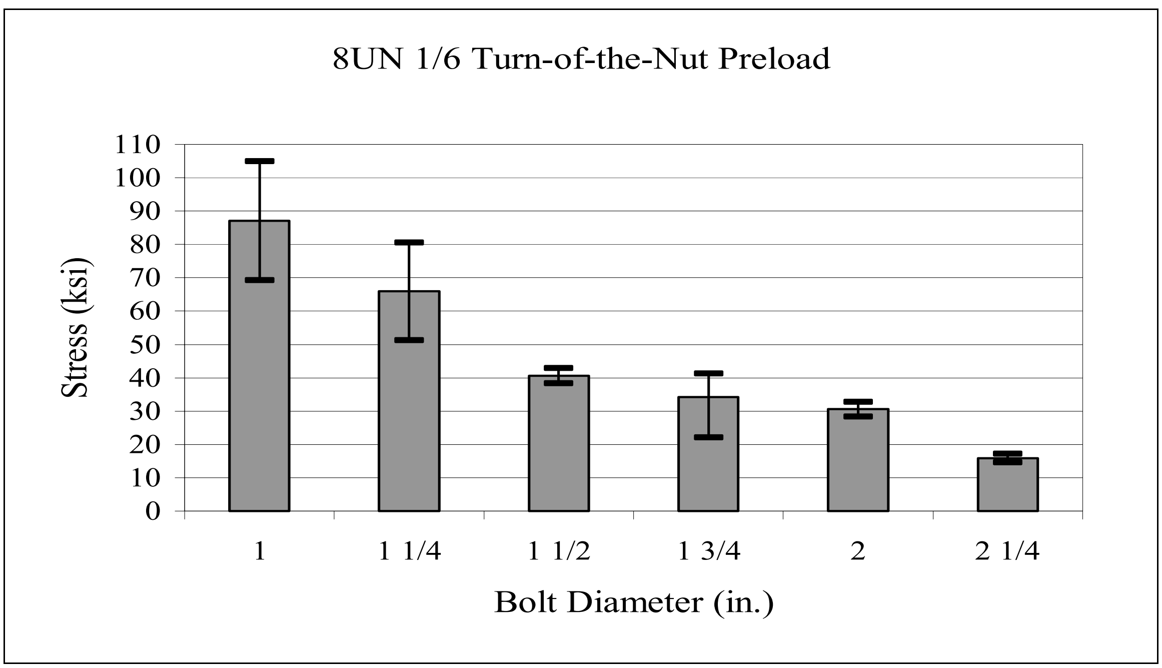

Figure B.10 – 8UN 1/6 turn-of-the-nut preload

VITA

Jason Halbert Richards was born in Beaumont, Texas, on October 6, 1979. He grew up in Terrell, Texas and graduated with honors from Terrell High School in May of 1998. Jason attended Texas A&M University and pursued the Structural Engineering track of Civil Engineering while a member of the Texas A&M Corps of Cadets and the Fightin' Texas Aggie Band, A-Battery. He graduated with his Bachelor of Science in Civil Engineering in December of 2002. Jason stayed at Texas A&M to pursue a Master of Science degree in Structural Engineering under the supervision of Dr. Peter Keating. He co-authored the research report, *Supplemental Laboratory Testing to Extend Scope of Proposed Standard Method of Tightening of Anchor Bolts*, during the term of his research assistantship with the Texas Transportation Institute. He graduated from Texas A&M University with his M.S. in May of 2004. Jason can be reached through his parents, Mr. and Mrs. W. H. Richards at:

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