To the Graduate Council:

I am submitting herewith a thesis written by Paxton Charles Lifsey entitled "Validating the Design Tensile Strength of Grade 5 and Metric Class 8.8 Bolts." I have examined the final paper copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Civil Engineering.

Mark D. Denavit, Major Professor

We have read this thesis
and recommend its acceptance:

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(Original signatures are on file with official student records.)

Validating the Design Tensile Strength of Grade 5 and Metric Class 8.8 Bolts

A Thesis Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Paxton Charles Lifsey

May 2024

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Abstract

Bolted connections offer many benefits including ease of installation and maintenance and replaceable components, but must have sufficient strength in order to take advantage of these benefits. One of the major ways a bolted connection is loaded is under tension. In this loading condition, bolted connections can fail via tension in the threaded area, shear in the bolt threads, or shear in the nut threads. Standards such as FED-STD-H28/2B and ISO TR/16224 include equations for the stress area and strength of these limit states as well as for the required length of engagement such that tensile failure controls. These equations are based on old, inaccessible data. Between standards, differences arise in equations for the same concept. Whether these are based on testing results, theory, or differing motivations for the equation is unclear. Additionally, there is the potential for changes or improvements to have been made in the manufacturing process since the generation of these equations, which would impact their accuracy. The objective of this thesis is to verify and compare existing equations for the tensile strength of bolts. Extensive physical testing was performed. Several testing configurations were developed to measure bolt mechanical properties and tensile strength for the relevant limit states. Testing was performed on two different material grades of bolts across different diameters and thread types. The material grades are of the same nominal material strength and composition, but differ in their use of imperial or metric thread forms. Analysis of the results quantified the tensile and thread shear stress areas for comparison to the equations. The length of engagement was studied through a probabilistic Monte Carlo simulation as well. The results of the work demonstrate that the current equations are generally accurate but can be refined. Potential improvements to the equations are suggested, along with established procedures for evaluating different kinds of bolts in future work. These results enable more accurate and confident connection design.

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Nomenclature

Symbol	Definition
A_s	Tensile stress area
$AS_{n,min}$	Internal thread shear area, minimum material condition
$AS_{s,min}$	External thread shear area, minimum material condition
$AS_{s,max}$	External thread shear area, basic condition
C_1	Nut dilation factor
C_2	Thread bending factor, external
C_3	Thread bending factor, internal
D_1	Minor diameter, internal thread
D_2	Pitch diameter, internal thread
D_c	Diameter of chamfer
D_m	Mean diameter of bell mouthed section of nut in the length of thread engagement
H	Addendum of external thread / height of sharp v-thread
LE	Length of engagement
LE_{cr}	Critical length of engagement
LE_r	Required length of engagement such that tensile rupture controls the strength of the bolt
P	Pitch
R_1	Shear stress area ratio
R_2	Material strength ratio

Td	Tolerance major diameter, external thread
Td_2	Tolerance on pitch diameter, external thread
TD_1	Tolerance on minor diameter, internal thread
TD_2	Tolerance on pitch diameter, internal thread
UTS_n	Ultimate Tensile Strength of the nut
UTS_s	Ultimate Tensile Strength of the bolt
d_{bsc}	Basic major diameter, external thread
d_{max}	Maximum major diameter, external thread
d_{min}	Minimum major diameter, external thread
$d_{1,bsc}$	Basic minor diameter, external thread
$d_{2,bsc}$	Basic pitch diameter, external thread
$d_{2,min}$	Minimum pitch diameter, external thread
es	Allowance, external thread
h_c	Height of chamfer
m	Nut height
m_{eff}	Effective nut height
n	Number of threads per unit length
s	Width across flat sides of hex nut

Chapter 1

Introduction

Many mechanisms for connecting materials exist across the varied engineering disciplines. In many steel structures for both civil and naval engineering projects, the major examples are welds, rivets, and bolts. While welds and rivets have their advantages, bolts are widely applicable, offering benefits like ease of installation and maintenance, replaceable components, and high strength.

Being able to take full advantage of these benefits requires selecting the correct fastener, the correct design approach, and the proper installation method for the given situation. Many factors influence this selection process, including environmental corrosion, hydrogen embrittlement, desired preload and torque, bolt material strength and reliability, and the mechanical strength of the connection.

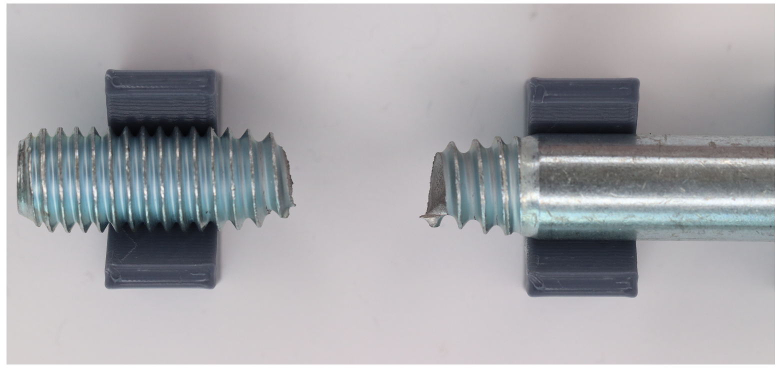
While all of these factors are important, ensuring the connection has the sufficient strength has direct implications on safety and is therefore a primary concern. Bolts can be subject to different actions, including shear, tension, torque, and bending; or combinations of these actions such as tension and torque during tightening. Bolts under tensile loads are the focus of this thesis, as the aging design equations for bolts in tension warrant investigation.

Three distinct failure modes for bolts in tension exist: tensile failure in the threaded section perpendicular to the load, shear failure in the bolt threads parallel to the load, and shear failure in the nut threads parallel to the load (Figure 1.1). The strength of the threaded parts in each of these failure modes depends on the geometry of the threads and the material properties of the bolt and the nut.

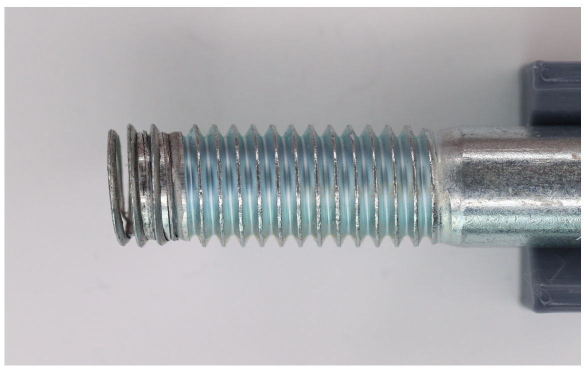
Equations for the tensile strength of bolts are presented in several standards including FED-STD H28/2B (GSA, 1991) and ISO TR/16224 (ISO, 2012). Hereafter, these standards will be referred to as FED-STD and ISO, respectively. Tensile strength for the limit state of tensile failure in these standards is equal to the circular tensile stress area times tensile strength of the bolt material. Equations are provided for the tensile stress area based on the geometry of the threads, but these equations vary between standards and have multiple variations given in FED-STD. Additionally, the data these equations are based on are not referenced by the standards and seem to be inaccessible for review.

Equations for the thread shear area of bolts are also presented in these standards. Thread shear area in these standards is an effective cylinder through the threads. Equations for thread shear strength are provided in ISO, equal to the thread shear area times the shear strength of the failing material, i.e., the shear strength of the bolt for external failure and the shear strength of the nut for internal failure. While the thread shear area equations are mechanically similar between standards, ISO incorporates different corrections to shear strength to account for thread bending and nut dilation, yielding results inconsistent with FED-STD.

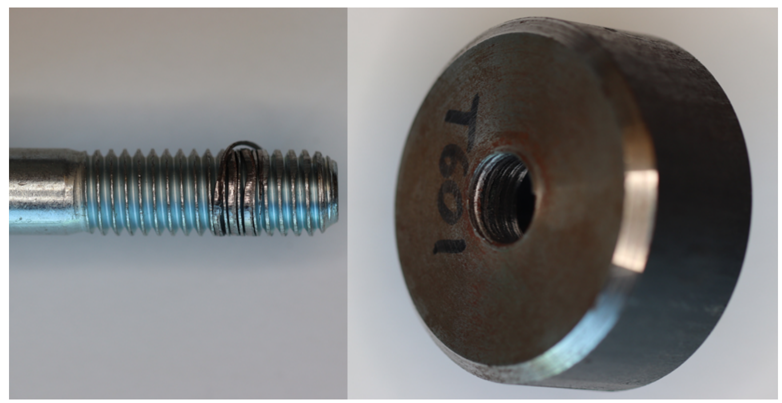
Often times, thread shear failure is to be avoided, as it is easier to replace a bolt failed in tension than it is one with stripped threads. In order to have tensile failure control, the thread shear strength must be sufficiently strong, and thus the length over which the threads are mated must be sufficiently long. This length is known as the length of engagement, LE. The thread shear area is proportional to this value, and so there exists a length of engagement referred to as the critical length of engagement, LE_{cr}, where the thread shear and tensile failure strengths are equal. Equations for the required length of engagement, LE_r, are provided in FED-STD, which aim to make tensile failure more likely. A probabilistic approach to determining the required



(a) Tensile Failure



(b) External Thread Shear Failure



(c) Internal Thread Shear Failure

Figure 1.1: Failure modes for bolts under tension.

length of engagement for nut specifications is presented in ISO. The equations in FED-STD have multiple variations, and it is not clear when each variation should be used. Further, these result in differing values in comparison to the probabilistic approach, weakening the validity of both methods.

These inconsistencies in the established equations lead to confusion in practice. These equations are, as noted, based on old, inaccessible data and experiments. Additionally, there were likely assumptions made regarding bolts that could be inaccurate, as bolt manufacturing processes could have evolved since the generation of the equations. This potentially leads to variability within groups of machined bolts, both in the material and geometrical sense. Regardless, the differences noted between equations produce noticeable changes in results, and whether these differences are based on testing results, theory, or differing motivations for the equations is unclear.

The objective of this thesis is to quantify the accuracy of existing design equations for high-strength steel bolts in tension and to develop modifications to the equations where appropriate. The objective is accomplished through extensive physical testing of bolts and analysis of the resulting data. Two grades of bolts will be examined: Grade 5 and Property Class 8.8. These two grades of bolts are of the same nominal material strength, but Grade 5 uses imperial diameters and threading while Property Class 8.8 uses metric. "Property Class" and "grade" will be used interchangeably throughout this thesis. Examining these grades separately will clearly demonstrate if the same conclusions can be drawn between the imperial and metric thread standards. Each of the three limit states will be evaluated individually and together in the context of the required length of engagement. A probabilistic assessment of the length of engagement was performed to account for variability in material properties, thread dimensions, and model error.

The structure of this thesis is as follows. Chapter 2 is a literature review, defining terms, concepts, and the current equations to be examined. Chapter 3 discusses materials and methods used throughout the experiments. Chapter 4 presents and discusses the findings of the experiments. Chapter 5 is a dedicated look at a proposed probabilistic evaluation of the length of engagement. Chapter 6 summarizes the work, identifies conclusions, and states recommendations.

Chapter 2

Literature Review

2.1 Behavior of Bolts Subject to Tension

The main failure modes for bolts subject to tension are tensile failure, where the bolt snaps in its threaded body; external thread shear failure, where the bolt threads are stripped; and internal thread shear failure, where the nut threads are stripped. Typically, designers prefer the tensile failure limit state to control, as it is usually the most economical to repair. Calculation of the controlling limit state depends on the length of thread engagement, geometry of the threads, and material strengths of the bolt and internally threaded part. Tensile rupture controls when the length over which threads are mated (i.e. the length of engagement) is relatively long, and shear rupture controls when the length of engagement is relatively short. Other factors, including nut dilation, coefficient of friction, applied torque, and number of threads in the grip also impact strength.

The amount of readily available, published research regarding fasteners in tension is surprisingly little. A large quantity of this available research that investigates the aforementioned factors do so outside the context of bolts in tension, limiting the selection further. Relevant papers, including some that are a larger focus of this thesis, are summarized to provide some insight into these effects.

One important piece of older research, Alexander (1977), forms the basis of the ISO equations. It identifies the equations further explored in later sections, as well as provides some insight into the effects of nut dilation and the coefficient of friction. Nut dilation is the phenomena where the 60° threads cause a wedging action when under load, increasing the minor diameter of the internally threaded part. This effectively reduces the shear areas of both threaded parts, decreasing the thread shear strength. This effect is included as a strength reduction factor in both thread shear strength equations in ISO.

Alexander (1977) also includes an explanation of the effect of applied torque on bolt tensile strength. It is well documented that the ultimate tensile strength of bolts in tension is reduced when the bolt is torqued. This is due to the shear stress transferring into the shank of the bolt via friction in the threads. What can also be seen is that the thread shear strength decreases due to nut dilation incurred by the applied torque. A reduced coefficient of friction further exacerbates this reduced shear strength by allowing nut dilation to occur more readily. However, a reduced coefficient of friction lessens the impact of torque on the axial strength by limiting the amount of shear stress transferred into the shank. Both axial and thread shear failure see reductions in strength due to torquing, but a well lubricated connection sees a larger reduction in thread shear strength than in axial strength. Thus, the more prominent effect is a shift in failure mode (Alexander, 1977).

As can be seen, the coefficient of friction has a significant effect in both nut dilation and the torque-tension relationship. Investigations into frictional effects are somewhat common, but many papers do not include their impact on tensile strength in their findings as their investigation is wholly focused on the friction of interacting surfaces. One paper, Dyson et al. (2023), revolves around properly characterizing the mechanisms of lubrication in fasteners. This research investigated the interaction between several different lubricants and fastener materials and through multivariate analysis made several interesting findings. First, it is concluded that the mechanism for lubrication changes between lubricants and includes plastic deformation of metals, burnishing and alignment, and adhering/embedding of non-metal particles. Secondly, the paper found that the friction coefficient for one material and one lubricant should not be assumed to be constant, and should be treated as a system property to be calculated in each system (Dyson et al., 2023).

Another factor affecting the failure mode of fasteners is the number of threads in the grip. Most fastener assemblies are expected to leave a number of threads between the bearing faces of the bolt head and nut face in order to prevent thread shear failure. Alexander (1977) posited that this shift in failure mode is due to necking of the threaded region near the engaged threads. This idea was confirmed by Grimsmo et al. (2016). Through a series of physical testing and finite element analysis, they demonstrated that the number of threads in the grip has several effects on failure behavior. A large number of threads in the grip led to tensile failure. A small number of threads in the grip led to thread failure. Intermediate amounts of gripped threads saw both failure modes. Grimsmo et al. (2016) observed that tests with small number of threads saw an overall increase in failure strength, and that necking of the bolt occurred close to the nut. This necking results in a loss of engaged threads and a weakened thread shear strength. The combination of these effects results in a shift in failure mode from tensile failure to thread shear failure at lower deformation levels for bolts with fewer threads in the grip. The effect of the number of threads in the grip is acknowledged in ASTM F606, which covers standard test methods for threaded fasteners. This standard requires at least 6 threads be within the grip, which is expected to be enough to avoid necking in the engaged threads.

These conclusions regarding behavior of bolts relevant to the number of threads in the grip was further supported by Yang et al. (2021). A fracture simulation of bolts in tension was performed with the goal of evaluating two models of fracture. A couple of interesting observations came out of this paper. First, it was observed that the thread rolling process introduces large plastic strain in the bolt threads, and reducing the thread shear fracture equivalent plastic strain yielded more accurate behavior of the fracture model. Second, the necking behavior for small numbers of threads in the grip was observed and confirmed to behave as described in Grimsmo et al. (2016).

Several investigations regarding bolts have been performed by Hagiwara. In one of his papers, he sought out to verify the calculated critical length of engagement as given in the International Standard ISO 898-6 (ISO, 1994). The critical length of engagement is the threaded length at which the failure mode transitions from thread shear failure to tensile failure. Hagiwara used the staircase method with small samples (Hagiwara et al., 1995). The staircase method is an experimental test method that enacts changes to the independent variable in response to the results of a previous test. Hagiwara employed the length of engagement as the independent variable and the type of failure (tensile or thread failure) as the event deciding the next value of the length of engagement. The initial test would use a length of engagement based on that calculated from the ISO method, and a set increment was chosen. Should tensile failure occur, the length of engagement would be decreased by the increment making thread failure more likely, and vice versa. Eventually the two types of failure alternate between one increment, and the critical length of engagement would be taken between these values. Hagiwara concluded that the method described in the ISO standard was reliable, as the calculations from the standard were in agreement with results from the staircase method.

2.2 Strength Equations

Established equations used in the design of fasteners are described in this section. This thesis will focus on the two documents noted previously, FED-STD (GSA, 1991) and ISO (ISO, 2012).

The formulas, as they appear in this document, differ from their original sources. These changes are to improve consistency and readability. For example, decimal equivalents of mathematical constant have been replaced with the constants themselves (e.g., 3.1416 is replaced with π). Additionally, a consistent set of symbols, based primarily on FED-STD, is used.

2.2.1 Tensile Stress Area

FED-STD notes that "tests have shown that externally threaded parts fail in tension at loads corresponding to those of unthreaded parts with diameters midway between their pitch diameter and their minor diameters". The standard includes two formulas for tensile stress area that "have been applied successfully to steel and other materials with ultimate strengths up to 180,000 psi". Equations 2.1 and 2.2 are listed in FED-STD Table II.B.1 as Formulas (1a) and (1b), respectively.

$$A_s = \pi \left(\frac{d_{2,bsc}}{2} - \frac{3H}{16} \right)^2 \tag{2.1}$$

$$A_s = \frac{\pi}{4} \left(d_{bsc} - \frac{9\sqrt{3}}{16n} \right)^2 \tag{2.2}$$

The term $3H/16$ in Equation 2.1, is noted in FED-STD to equal "half external thread addendum". The external thread addendum typically refers to the radial distance between pitch line and the major line, thus this definition is inconsistent with the experimental observation stated in FED-STD that underlays the equations.

Half the radial distance between the pitch line ($D_{2,bsc}, d_{2,bsc}$) and the minor line ($D_{1,bsc}, d_{1,bsc}$) for the basic thread profile is $H/8$. The additional $H/16$ that is subtracted from $d_{2,bsc}/2$ in Equation 2.1 is intended to account for the difference between basic and minimum thread dimensions.

Figure 2.1 shows the basic thread profile for UN bolts. At basic dimensions, the internal and external threads are of the exact same size. Tolerances are applied to the basic thread profile in such a way that the largest allowable bolt will still fit in the smallest allowable nut. As a general rule, symbols with a capital "D" refers to

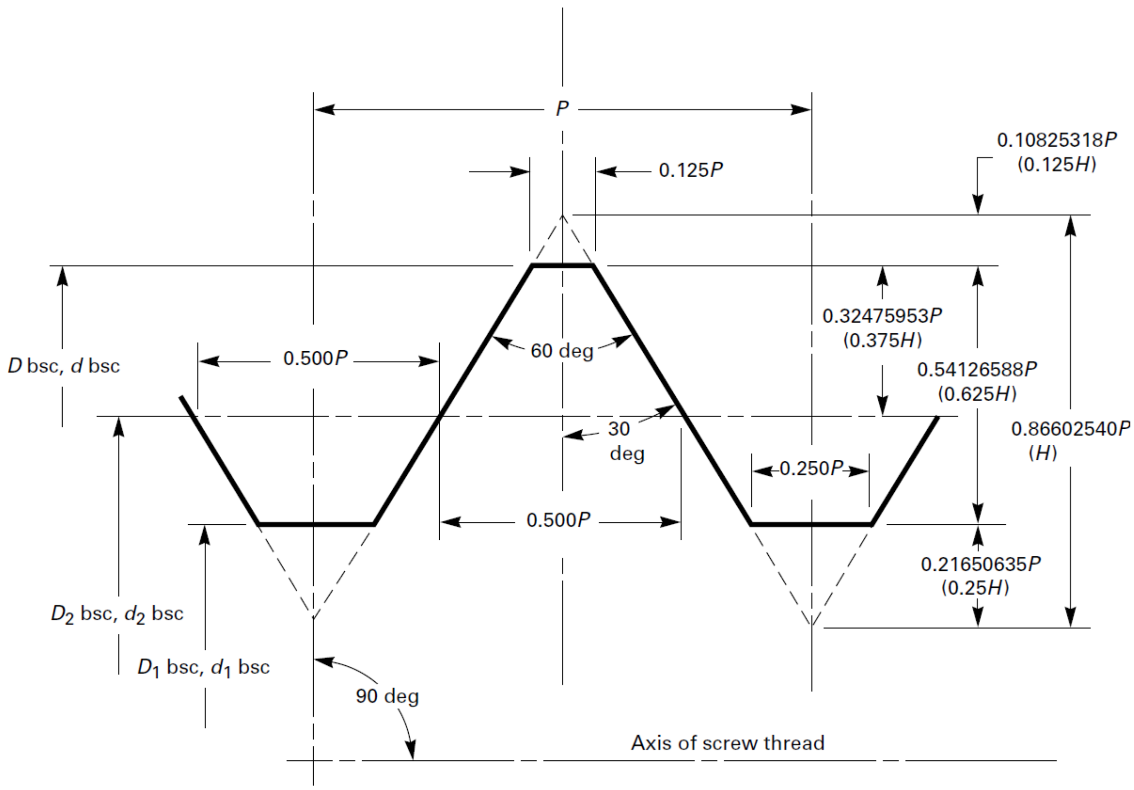


Figure 2.1: Basic Profile of UN and UNR Screw Threads (ASME B1.1)

internal thread dimensions and symbols with a lowercase "d" refers to external thread dimensions.

Figure 2.2 shows the range of pitch diameter and minor diameter for several thread types as tabulated in ASME B1.1 for class 2A threads. As noted in ASME B1.1, the minimum minor diameter values are only for reference and not typically specified. The diameters in Figure 2.2 are normalized according to Equation 2.3. The y-axis in this plot is in fractions of H. As expected, $-3H/16$ is between the ranges of the pitch and minor diameters.

$$d_{s,norm} = \frac{d_s - d_2}{2H} \tag{2.3}$$

where d_s is the tensile stress diameter, d_2 is the average measured pitch diameter, and H is the sharp V height of the thread based on the nominal thread pitch.

Equations 2.1 and 2.2 are identical when using precise definitions of $d_{2,bsc}$ and d_{bsc} from the basic thread profile (see Figure 2.1). Small differences can arise between the results of the two equations when using rounded values for the mathematical constants and thread dimensions (e.g., tabulated values).

ISO defines the tensile stress area with the following equation. The diameter term is not squared in the standard. The diameter term is squared in Equation 2.4 under the assumption that the square was omitted in the standard due to a typesetting error.

$$A_s = \frac{\pi}{4} \left(\frac{d_{2,bsc} + d_{1,bsc} - \frac{H}{6}}{2} \right)^2 \tag{2.4}$$

The tensile stress area in Equation 2.4 is equal to the average of the basic pitch diameter and the basic minor diameter minus $H/12$ (or $H/24$ from either side of the threads). This fraction of H does not have a direct physical equivalent on the thread form (i.e., no two locations of note are separated by $H/24$, Figure 2.1). The fraction of H is thus likely to be meant to account for actual bolt dimensions that are less

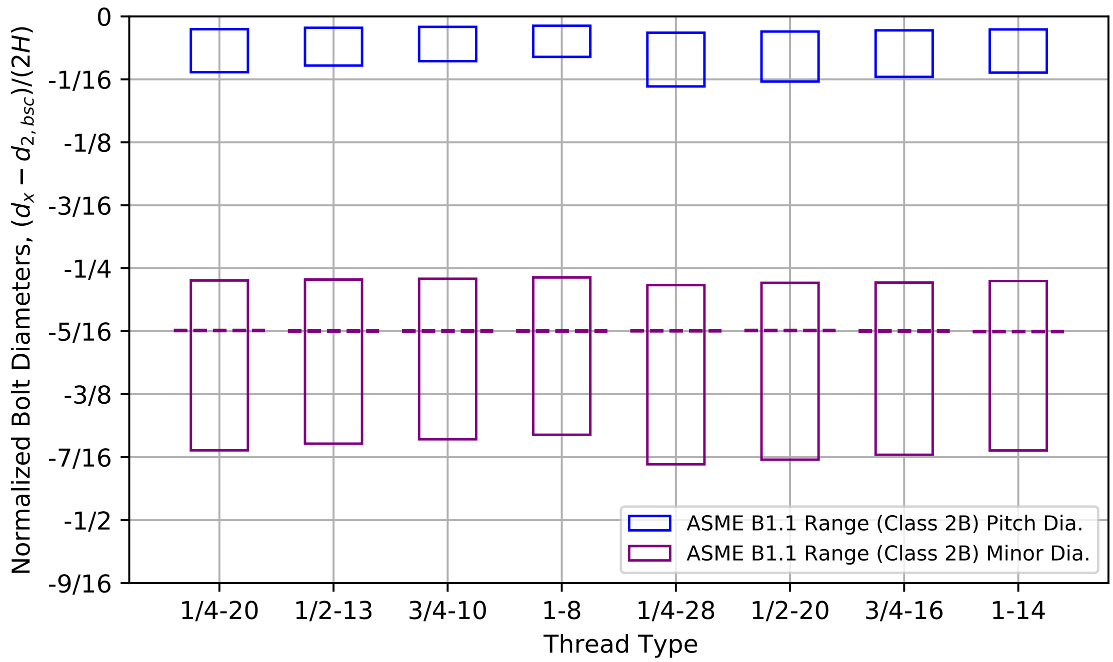


Figure 2.2: Normalized values of pitch and minor diameters. Box shows range from maximum to minimum. Dashed line shows basic. Data from ASME B1.1 for class 2A threads.

than the basic dimensions (Figure 2.2). Overall, it appears that the ISO equation has largely the same basis of formulation as the FED-STD equations, in that it places the diameter of the tensile stress area midway between the pitch and minor diameters and then accounts for differences between basic and actual dimensions.

A comparison between the FED-STD equations and the ISO equation is seen in Figure 2.3. The tensile stress areas are normalized by the nominal unthreaded body area, A_{bsc}.

$$A_{bsc} = \frac{\pi}{4} d_{bsc}^2. \tag{2.5}$$

Figure 2.3 demonstrates that the two FED-STD equations are the same. It can also be seen that the tensile stress area from the ISO equation is consistently smaller than that from the FED-STD equations. However, the percent difference between these tensile stress areas is small. For the thread types in Figure 2.3, the difference was less than 1%, with the largest difference computed for the 1/4 in. bolts.

2.2.2 Shear Area, External Threads

FED-STD notes that "the geometric shear area of an external thread at minimum material is equal to the area of that thread which is intersected by a cylinder with a diameter equal to the maximum minor diameter of the mating internal thread" over the length of engagement. The standard includes two formulas for this area.

$$AS_{s,min} = \pi n D_{1,max} \left[\frac{1}{2n} + \frac{1}{\sqrt{3}} \left(d_{2,min} - D_{1,max} \right) \right] LE \tag{2.6}$$

$$AS_{s,min} = \pi D_{1,max} \left[\frac{3}{4} - \frac{n}{\sqrt{3}} \left(TD_1 + Td_2 + es \right) \right] LE \tag{2.7}$$

A simplified formula, Equation 2.8, based on uncited empirical data is also included.

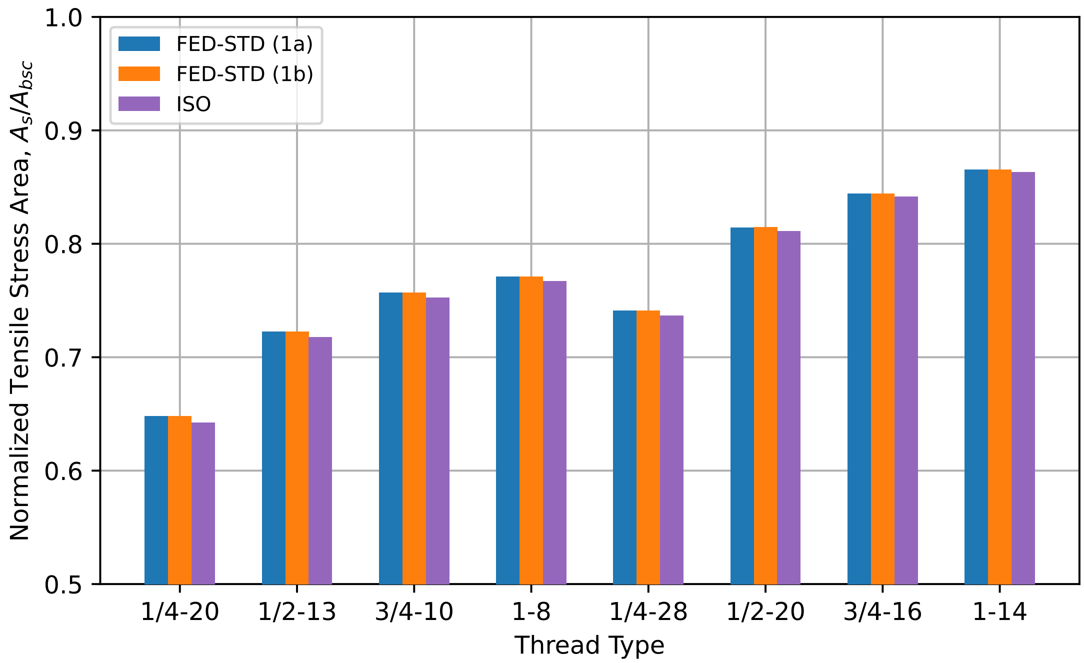


Figure 2.3: Comparison of Tensile Stress Area Equations

$$AS_{s,min} = \pi d_{2,bsc} \frac{5}{8} LE \tag{2.8}$$

Equations 2.6, 2.7, and 2.8 are listed in FED-STD Table II.B.1 as Formulas (4a), (4b), and (5), respectively.

FED-STD includes additional equations for the shear area of external threads at basic size, $AS_{s,max}$. Two formulas are given in FED-STD Table II.B.1 and labeled (6a) and (6b). Formula (6a) references a chart of nominal minimum length of engagement. Formula (6b) is shown here as Equation 2.9.

$$AS_{s,max} = \pi D_{1,bsc} \frac{3}{4} LE \tag{2.9}$$

Equation 2.9 is used when calculating the required length of engagement based on shear of internal thread.

The equation for shear area of external threads in ISO is given as Equation 2.10.

$$\begin{aligned} AS_s = 0.6 m_{eff} \pi n D_1 &\left[\frac{1}{2n} + \frac{1}{\sqrt{3}} (d_2 - D_1) \right] \\ +0.4 m_{eff} \pi n D_m &\left[\frac{1}{2n} + \frac{1}{\sqrt{3}} (d_2 - D_m) \right] \end{aligned} \tag{2.10}$$

where $D_m = 1.026 D_1$ and m_{eff} is the effective nut height as shown in Figure 2.4. $m_{eff} = m - 0.6 h_c$ for nuts with one chamfered end and $m_{eff} = m - 1.2 h_c$ for nuts with two chamfered ends. h_c is the height of chamfer on each end.

Equation 2.10 is similar in form to Equation 2.6 with the exception of an adjustment to the minor diameter of the nut over a portion of the nut height to account for the bell-mouthing. Bell-mouthing is the gradual taper of the internal threads that can occur during machining. Equation 2.10 also explicitly accounts for chamfers.

While the FED-STD equations do not explicitly include adjustment for chamfer, the standard states that "the effectiveness of partial threads, in the countersinks (or chamfers) on the ends of the internal thread, is not always the same" and recommends

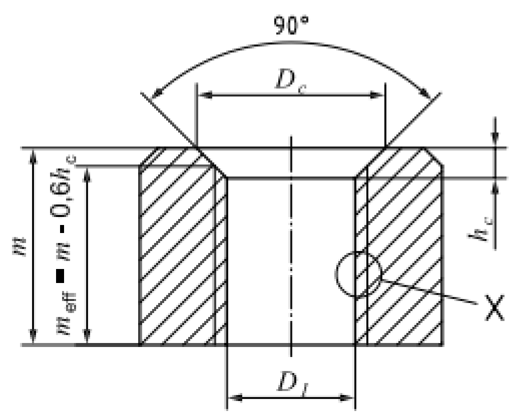


Figure 2.4: Effective Height of a Chamfered Nut (ISO, 2012)

that only half the height of the countersink/chamfer should be considered when evaluating strength.

When using the same tabulated values used in FED-STD Equation 2.6 for the minimum material condition in the ISO Equation 2.10 and neglecting the chamfer correction, these equations are exactly the same. However, the predicted external thread shear strength between the two standards are still different. This is due to corrections in ISO for nut dilation and thread bending.

2.2.3 Shear Area, Internal Threads

The shear area for internal threads is conceptually similar to that for external threads. FED-STD includes two formulas for shear area based on minimum material.

$$AS_{n,min} = \pi n d_{min} \left[\frac{1}{2n} + \frac{1}{\sqrt{3}} \left(d_{min} - D_{2,max} \right) \right] LE \tag{2.11}$$

$$AS_{n,min} = \pi d_{min} \left[\frac{7}{8} - \frac{n}{\sqrt{3}} \left(Td + TD_2 + es \right) \right] LE \tag{2.12}$$

Additionally, the standard includes a simplified formula for $d \geq 0.250$ in. based on empirical data.

$$AS_{n,min} = \pi D_{2,bsc} \frac{3}{4} LE \tag{2.13}$$

Equations 2.11, 2.12, and 2.13 are listed in FED-STD Table II.B.1 as Formulas (2a), (2b), and (3), respectively.

The equation for shear area of internal threads in ISO is given as Equation 2.14.

$$AS_n = m_{eff} n \pi d \left[\frac{1}{2n} + \frac{1}{\sqrt{3}} (d - D_2) \right] \tag{2.14}$$

where $m_{eff} = m - 0.6h_c$ for nuts with one chamfered end and $m_{eff} = m - 1.2h_c$ for nuts with two chamfered ends.

Apart from accounting for the height of the chamfers and not specifying whether maximum, minimum, or basic geometric dimensions should be used, Equation 2.14 is identical to Equation 2.11. As stated in Section 2.2.2, if using the same values as FED-STD for the minimum material condition and disregarding corrections for chamfers, the equations yield the exact same results, but differences remain between the two standards for thread shear strength due to corrections for nut dilation and thread bending.

2.2.4 Length of Engagement

FED-STD notes that if failure occurs, "it is generally more economical that the externally threaded part will break rather than that either the external or internal thread will strip." To achieve this condition, the standard provides several formulas for the length of engagement such that the tensile strength of the bolt is less by a margin than either the internal or external thread shear strength. For clarity, the required length of engagement such that tensile rupture controls the strength of the bolt is denoted as LE_r in this document.

The use of the several equations for LE_r in FED-STD depends on the relative strengths of the internally threaded part and the externally threaded part defined as R_1/R_2. R_1 and R_2 are defined in Equations 2.15 and 2.16, respectively.

$$R_1 = \frac{AS_{s,max}}{AS_{n,min}} \tag{2.15}$$

where $AS_{s,max}$ is determined using Equation 2.9 or FED-STD Table II.B.1 Formula (6a) and $AS_{n,min}$ is determined using Equation 2.11 or 2.12.

$$R_2 = \frac{UTS_n}{UTS_s} \tag{2.16}$$

Equations 2.15 and 2.16 are listed in FED-STD Table II.B.1 as Formulas (8) and (9), respectively.

If $R_1/R_2 \approx 1$ ($0.9 < R_1/R_2 < 1.1$), then combined external and internal thread shear failure is assumed to control and Equation 2.17 is used to determine LE_r. The distance from 1 for the approximation is determined from where Equations 2.17 and 2.19 meet. See Figure 2.5. Equation 2.17 is listed in FED-STD Table II.B.2 as Formula (13).

$$LE_r = \frac{4A_s}{\pi d_{2,bsc}} \tag{2.17}$$

where, A_s is calculated from either Equation 2.1 or Equation 2.2.

If $R_1/R_2 < 1$, then exterior thread shear failure is assumed to control and Equation 2.18 is used to determine LE_r. Equation 2.18 is listed in FED-STD Table II.B.2 as Formula (14).

$$LE_r = \frac{2A_s}{AS_{s,min}/LE} \tag{2.18}$$

where, A_s is calculated from Equation 2.1 or Equation 2.2, and $AS_{s,min}/LE$ is calculated from Equation 2.6 or Equation 2.7.

If $R_1/R_2 > 1$, then internal thread shear failure is assumed to control and Equation 2.19 is used to determine LE_r. Equation 2.19 is derived from FED-STD Table II.B.2 Formulas (15) and (16).

$$LE_r = \frac{2A_s}{AS_{s,max}/LE} \frac{R_1}{R_2} \tag{2.19}$$

where, A_s is calculated from either Equation 2.1 or Equation 2.2 and $AS_{s,max}/LE$ is calculated from Equation 2.9.

Example values of the required length of engagement, LE_r, computed using FED-STD equations for Grade 5 bolts and either A36 or HY-80 internally threaded parts are listed in Table 2.1. The values of LE_r assuming minimum specified strengths for each material (i.e., UTS_s = 120 ksi for Grade 5, UTS_n = 58 ksi for A36, and UTS_n = 100 ksi for HY-80). Internal thread shear failure, Equations 2.11 and 2.18, controls

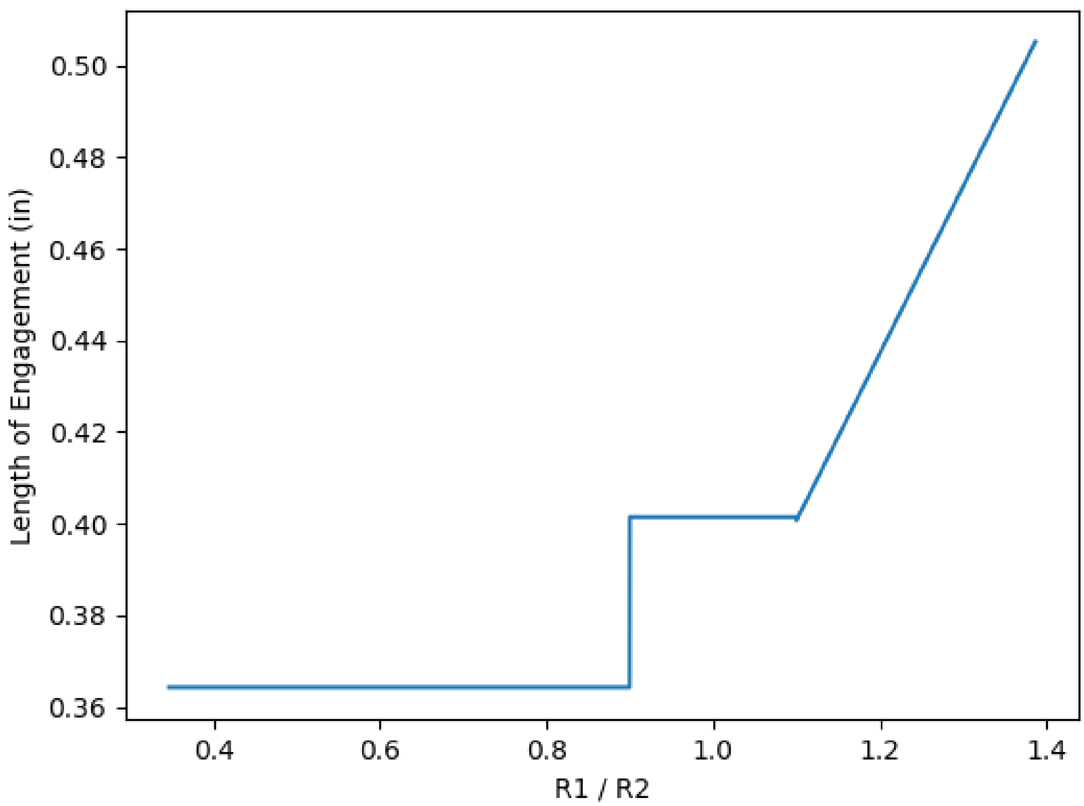


Figure 2.5: Length of Engagement vs R_1/R_2 for a 1/2-13 Bolt

Table 2.1: Required length of engagement, LE_r, for Grade 5 bolts computed using minimum specified material strengths

	A36		HY-80	
Thread Type	LE_r (in.)	LE_r/d_{bsc}	LE_r (in.)	LE_r/d_{bsc}
1/4-20	0.244	0.977	0.186	0.745
1/4-28	0.289	1.155	0.204	0.817
1/2-13	0.523	1.045	0.401	0.803
1/2-20	0.612	1.223	0.436	0.871
3/4-10	0.803	1.071	0.622	0.829
3/4-16	0.930	1.241	0.669	0.893
1-8	1.074	1.074	0.839	0.839
1-14	1.253	1.253	0.908	0.908

for the A36 internally threaded parts. Combined thread shear failure, Equations 2.11 and 2.18, controls for the HY-80 internally threaded parts.

There are no explicit equations for length of engagement provided in ISO. The formulas for tensile strength and thread shear strength can set equal to each other to determine the critical length of engagement. However, it provides near equal likelihood between the two different failure modes, whereas the FED-STD equations explicitly preclude thread shear failure. In addition, a combined failure mode is not identified in ISO, further separating itself from FED-STD. In order to identify a length of engagement for a specific chance of either failure mode occurring, ISO provides a probabilistic approach using a Monte Carlo simulation.

2.2.5 Thread Shear Strength

In ISO, there are equations for the strength of each of the failure modes. While the tensile strength equation is trivial, the equations for thread shear strength involve correction factors for nut dilation and thread bending, as previously mentioned. The equation for C_1, the nut dilation factor, is as given in Equation 2.20:

$$\begin{gathered} C_1 = -\left(\frac{s}{d_{bsc}}\right)^2 + 3.8\left(\frac{s}{d_{bsc}}\right) - 2.6 \\ \text{for } 1.4 \leq \frac{s}{d_{bsc}} < 1.9 \end{gathered} \tag{2.20}$$

for $1.4 \leq \frac{s}{d_{bsc}} < 1.9$, where s is the width across the flats of the nut.

The equation for the external thread bending correction factor, C_2, is as given in Equation 2.21:

$$C_2 = 5.594 - 13.682\left(\frac{R_2}{R_1}\right) + 14.107\left(\frac{R_2}{R_1}\right)^2 - 6.057\left(\frac{R_2}{R_1}\right)^3 + 0.9353\left(\frac{R_2}{R_1}\right)^4$$
$$\text{for } 1 < \frac{R_2}{R_1} < 2.2$$
$$C_2 = 0.897$$
$$\text{for } \frac{R_2}{R_1} \leq 1 \tag{2.21}$$

The equation for the internal thread bending correction factor, C_3, is as given in Equation 2.22:

$$C_3 = 0.728 + 1.769\left(\frac{R_2}{R_1}\right) - 2.896\left(\frac{R_2}{R_1}\right)^2 + 1.296\left(\frac{R_2}{R_1}\right)^3$$
$$\text{for } 0.4 < \frac{R_2}{R_1} < 1 \tag{2.22}$$
$$C_3 = \ = 0.897$$
$$\text{for } \frac{R_2}{R_1} \geq 1$$

In both these equations, R_1 and R_2 are as defined in Equations 2.15 and 2.16, respectively.

The inclusion of the C-factors causes the thread shear strength, as predicted by ISO, to no longer be identical to FED-STD when designing for the minimum material condition.

Chapter 3

Materials and Methods

3.1 Materials

Testing was performed on two grades of bolt material, Grade 5 and Property Class 8.8 (hereafter referred to as Grade 8.8). Grade 5 bolts are specified in SAE Standard J429. These bolts are in imperial diameters and thread forms. Grade 8.8 bolts are specified in ISO Standard 898-1. These bolts are in metric diameters and thread forms. Table 3.1 includes the relevant specified material properties for both Grade 5 and Grade 8.8 bolts. From these tables it can be seen that the two materials are comparable in strength. It was for this reason these standardized grades were chosen. Having comparable strength will keep the results in the same order of magnitude, allowing major differences in behavior to be attributed to differences in metric and imperial thread forms.

The metric and imperial thread forms are widely similar. Both use 60° threads and are defined by the same basic thread profile (see Figure 2.1). The only major differences come in ranges of tolerances and using a different base unit system (millimeters vs. inches).

For Grade 5 bolts, both coarse and fine threads were examined in commonly used sizes as to examine differences between the two thread styles. For Grade 8.8 bolts,

Table 3.1: Bolt Grade Material Specifications

Grade	**Size Range**	**Min. Yield Strength (psi)**	**Min. Tensile Strength (psi)**	**Rockwell Hardness (min/max)**
Grade 5	1/4 - 1 in	92000	120000	C25/C34
Grade 8.8	up to M16	92800	116000	N/A
Grade 8.8	over M16	95700	120000	C23/C34

commonly used pitches as close in diameter to the tested Grade 5 bolts were selected to increase comparability. The diameters and thread types of bolts tested are found in Table 3.2.

A minimum length is required such that bolts are suitable for all required tests and measurements. The minimum bolt length based on bolt diameter is listed in Tables 3.3 and 3.4. For the smaller bolts (e.g. 1/4 and M8), the minimum bolt length is controlled by the size of the tension testing apparatus. The bolt holders have dead space that must be filled, as well as accounting for the appropriately sized washers. For the larger bolts, the minimum bolt length is controlled by the turned down tension testing, as ample unthreaded length larger than the minimum requirements of the bolt holders was necessary to form coupon shapes. In some cases, it was preferable to purchase bolts larger than the minimum length as to accommodate attaching measurement devices.

Keeping lot control was a critical concern whilst procuring bolts. For example, the tensile stress area was necessarily determined from the tensile strength and the material ultimate tensile strength from two different bolts. The resulting tensile stress area would have been meaningless if the materials were not comparable. Thus, all tested bolts of one thread type were procured from the same heat treatment lot. Repeat lots were obtained for Grade 5 1/4-20 and 1/2-13 bolts. This was to evaluate the consistency and repeatability of results for one thread type given a change in material properties.

This concern over material property consistency extended to internally threaded parts fabricated for testing. The same care was taken in purchasing material where knowing the ultimate tensile strength was critical. Material for these threaded parts were selected to assist in achieving internal thread shear failure and combined thread shear failure. For internal thread shear failure, A36 was selected for its ease of availability and clear weakness in comparison to the bolt material. For combined thread shear failure, HY80 was selected due to the relevance to naval engineering (a concern of the larger project this thesis was conducted under) and its similar

Table 3.2: Thread types tested by Grade

Grade 5	**Grade 8.8**
1/4-20	M8-1.25
1/4-28	
1/2-13	M12-1.75
1/2-20	
3/4-10	M20-2.5
3/4-16	
1-8	M24-3.00
1-14	

Table 3.3: Minimum bolt lengths, unified threads

Bolt Diameter	**Minimum Length (in.)**	**Preferred Length (in.)**
1/4 in.	3	4
1/2 in.	4	4
3/4 in.	6	6
1 in.	6	7

Table 3.4: Minimum bolt lengths, metric threads

Bolt Diameter	**Minimum Length (mm)**
M8	100
M12	100
M20	160
M24	180

ultimate tensile strength to the bolt material. For external thread shear failure, Grade 2H heavy hex nuts (hereafter called strong nuts) were purchased that had strength comparable to Grade 9 bolts which have minimum 180,000 psi tensile strength, clearly larger than that of Grade 5 and 8.8 bolts.

The material properties of all bolts and internally threaded part stock was determined via coupon testing in accordance with ASTM E8 with exceptions in ASTM F606 as noted in Section 3.2.2. Further detail on the methodology of the bolt material testing is given in sections 3.2.2 and 3.2.3. The average of the results of 5 tests was taken as the defined material property for each bolt. A summary of these tested properties is given in Tables 3.5 and 3.6. Enough material to create all necessary internally threaded parts of each grade was purchased such that only one lot of each grade was necessary.

3.2 Methods

3.2.1 Naming Convention

A consistent naming convention was used for all tested bolts. Each lot of procured bolts was given a name in the format L###, where ### is a lot index, starting at 001 and increasing sequentially as lots were procured. Bolts within each lot were given a name in the format L###-X###, where L### is the lot from which the bolt was drawn, "X" is a letter indicating the type of test performed, and the last three numbers are a bolt index. The bolt index was not necessarily sequential. For example, if 10 bolts are tested with a strong internally threaded part to evaluate fastener tensile strength, they may have indices 001 through 010. If then another 10 bolts are tested in tension with internally threaded parts made of A36 steel to evaluate internal thread shear area, they may have indices 101 through 110 to more clearly distinguish the two sets. This robust naming scheme came about because the work presented in this thesis is a subset of a larger project.

Table 3.5: Average bolt material properties by lot

Bolt Type	**Yield Strength (psi)**	**Tensile Strength (psi)**	**Elongation %**
1/4-20(a)	128600	138700	11.98
1/4-20(b)	130000	144600	14.80
1/4-28	137500	148900	19.17
1/2-13(a)	101400	128100	21.90
1/2-13(b)	118900	136800	18.91
1/2-20	119100	136700	21.72
3/4-10	112500	136500	18.82
3/4-16	120100	142400	17.72
1-8	133700	144600	29.56
1-14	107900	134700	23.19
M8-1.25	129900	141200	18.71
M12-1.75	114000	122800	15.11
M20-2.50	126300	135600	23.01
M24-3.00	125300	136700	21.54

Table 3.6: Average internally threaded part material properties by lot

Material	**Yield Strength (psi)**	**Tensile Strength (psi)**	**Elongation %**
A36	51300	74100	35.13
HY80	89900	106900	29.70

3.2.2 Test Types

Several types of tension tests were performed to investigate the equations described. These tests include:

- Type 0: tests with internally threaded part of sufficient length of engagement and unmodified bolt to evaluate bolt tensile strength

- Type R: tests with internally threaded part of sufficient length of engagement and reduced diameter bolt to evaluate bolt material properties

- Types 5 and 7: tests with flattened strong nut and bolt with removed chamfer to evaluate external thread shear strength

- Types 6 and 8: tests with consumable internally threaded part to evaluate internal thread shear strength

ASTM F606 (2021) outlines standard test methods for bolts in tension. ASTM F606 Section 3.4, "Axial Tension Testing of Full-Size Products," and Section 3.6, "Tension Testing of Machined Test Specimens," were followed. Type 5, 6, 7, and 8 tests saw an exception to this standard, in that they did not develop the full strength of the bolt, as these tests had the goal of achieving thread shear failure.

Type 0 tests were the most basic test type. These were designed to achieve tensile failure. An axial extensometer was attached to the bolt during testing and kept on through the ultimate load, but was removed before failure. This was done to characterize the yield strength.

To begin the testing process, 20 Type 0 tests were performed on 1/2-13 bolts. Statistical analysis was performed to select an appropriate sample size for all future tests based on the coefficient of variation seen. It was determined that 5 tests of each type would be sufficient to establish a reasonable 90% confidence interval.

Type R tests were essentially coupon tests. These machined bolts had their shanks reduced to conform with ASTM E8, following the exceptions in ASTM F606, which limit the reduction to no more than 25% of the original diameter (or as large as feasible for smaller sizes). See Figure 3.1 for an example shop drawing of a reduced diameter bolt. A miniature axial extensometer was attached to the reduced shank during each test, and was left on through the ultimate load but removed before failure.

Type 5 tests used reduced lengths of engagement in order to ensure thread shear failure. The chamfered ends of the bolt and the strong nut were removed to prevent partial threads in these regions from affecting the strength of the bolt. An ultimate tensile strength of 180,000 psi was assumed for these strong nuts, and this assumption was confirmed to be reasonable based on hardness testing of the strong nut. Target lengths of engagement were calculated using the material strength of the bolt lot as determined in Type R tests along with the assumed strong nut material strength. These targeted lengths of engagement were achieved by placing a gage ball inside the strong nut against the flat end of the bolt. The strong nut was then tightened until a straight edge against the face of the strung nut made contact with the gage ball. See Figure 3.2 for a diagram of this process. An extensometer could not be used effectively during these tests as thread shear failure is too brittle to allow a reliable window during which the test could be paused and the extensometer removed.

Type 6 tests used consumable washers of A36 material with short enough lengths of engagement to ensure internal thread shear failure. Target length of engagement of the consumable washers were determined as 75% and 37.5% of Equation 2.19, using material strengths of the washer and material strength of the bolt lot being tested as determined in Type R tests. Washers were then fabricated for each of these lengths of engagement; see Figure 3.3 for an example. Type 8 tests were of the same pattern, using HY80 material for the fabricated washer and Equation 2.17.

Type 7 tests, i.e., the staircase method for external thread shear failure, were performed similarly to Type 5 tests, but with the target length of engagement of the first test adjusted to near 100%. If tensile failure occurred, the length of engagement was decreased by one size of gage ball and ran again. The gage ball set was in increments of 1 mm (0.04 in). If thread shear failure occurred, the length of engagement was instead increased. Testing would conclude when the failure mode

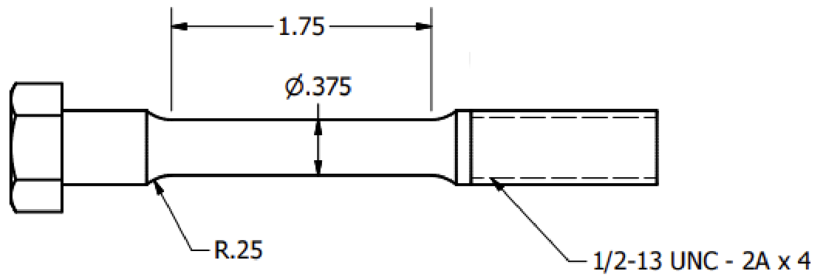


Figure 3.1: Reduced diameter bolt for Type R tests, size 1/2-13 (all units in inches)

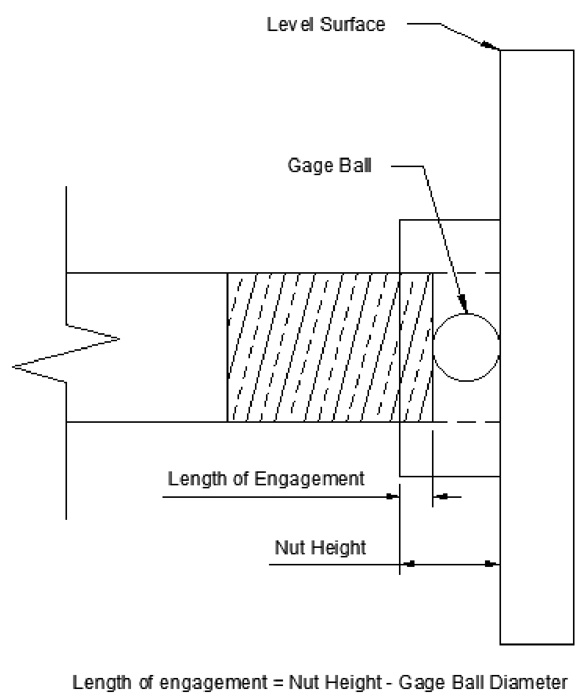


Figure 3.2: Reducing the length of engagement for Type 5 and 7 tests

Quantity	X (in.)	Z (in.)	Material
5	0.2137	.75	A36
5	0.4818	.75	A36
5	0.1338	.6	HY80
5	0.3669	.6	HY80

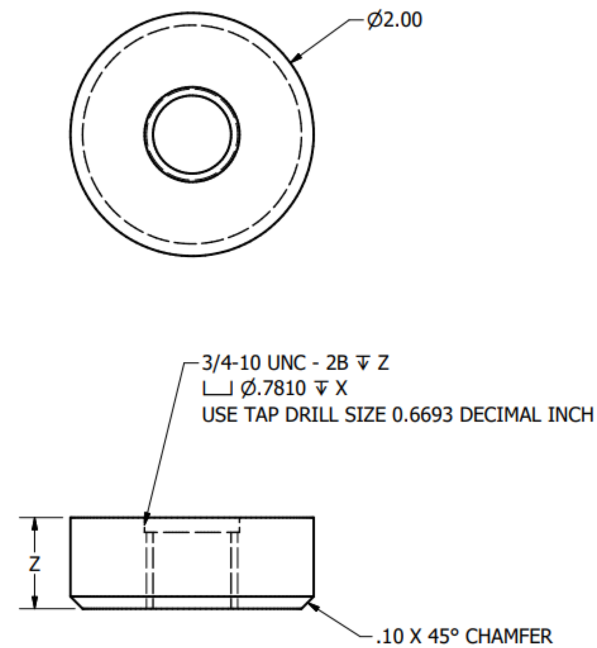


Figure 3.3: Consumable washers for 3/4-10 (all units in inches)

was consistently changing between tests, as this would trap the critical length of engagement within the 1 mm range.

3.2.3 Test Procedures

A step-by-step explanation of a single test is as follows:

1. Prepare the bolt for testing.
   (a) For Type R tests, bolts will be machined to appropriate sizes identified in ASTM E8, noting that ASTM F606 requires that the reduced diameter be no smaller than 75% of the original diameter.
   (b) For Type 5 and 7 tests, the chamfer at the end of the bolt must be sawn off.
2. Prepare the internally threaded part for testing.
   (a) For Type 0 and R tests, use threaded washer with sufficient length of engagement.
   (b) For Type 5 and 7 tests, use flattened strong nut.
   (c) For Type 6 and 8 tests, use fabricated washer of desired material.
3. Take geometric measurements in accordance with Section 3.2.4.
   (a) All tests: major external thread diameter, body diameter, and external pitch diameter.
   (b) Type 5 and 7 tests: the internal minor diameter and the height of the nut shall also be measured.
   (c) Type 6 and 8 tests: the internal minor diameter and the length of engagement (i.e., the height of the threads) shall also be measured.
4. Prepare the universal testing machine for testing (needs to be completed only once per testing session). A proper setup is shown in Figure 3.4.

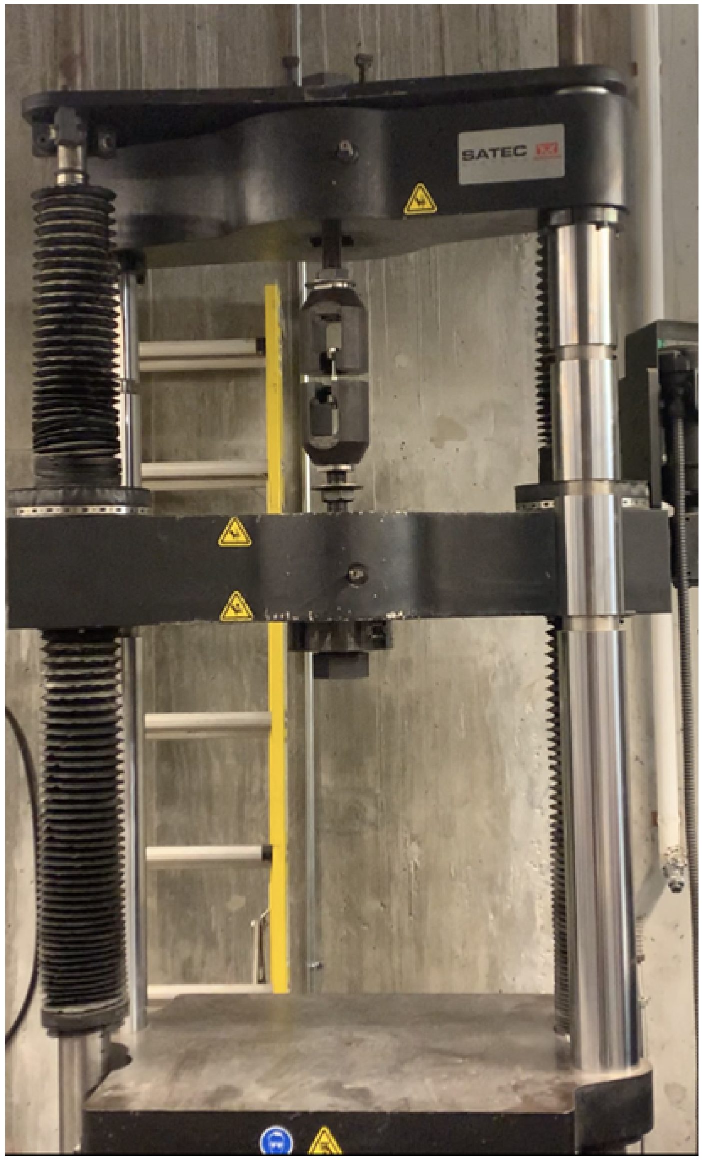


Figure 3.4: Universal testing machine with bolt properly installed.

(a) Install the spherical head rod supports at the top and bottom of the machine.

(b) Place the first spherical head rod in the upper support.

(c) Thread a nut down the threaded end of the spherical head rod past the distance the bolt holders will thread.

(d) Thread the threaded adapters into the bolt holders.

(e) Thread a bolt holder assembly (i.e., bolt holder plus threaded adapter) onto the upper spherical threaded rod with a washer. Thread the nut down to the washer.

(f) Place a square washer over the upper side of the hole in the lower crosshead, followed by a washer.

(g) Place the second spherical head rod in the lower support through the two washers.

(h) Hold the spherical head rod flush against the mount and thread the nut down so that it is roughly half an inch above the lower crosshead.

(i) Thread a bolt holder assembly onto the lower spherical threaded rod.

(j) Adjust the crosshead so that the two bolt holders are nearly touching.

(k) Prepare and select appropriate test controls.

5. Place the bolt in its unthreaded washer.

6. Attach the appropriate internally threaded part for the test type to the bolt, leaving enough space between the threaded and unthreaded part to allow placement of the bolt in the bolt holders.

7. For Type R tests, attach the miniature axial extensometer.

8. Install the bolt in the bolt holders.

9. Adjust the internally threaded part.

(a) For Type 0 tests, leave 6 threads exposed between the inner faces of the internally threaded part and the unthreaded washer.

(b) For Type 5 and 7 tests, measure the nut height and select a gage ball of a height that reduces the nut height to the targeted length of engagement for the test. Place the gage ball inside the nut on the sawn-off face of the bolt and cover with a flat surface. Tighten the assembly until the gage ball makes contact with the flat surface.

(c) For Type 6 and 8 tests, ensure the chamfer of the bolt is beyond the internal threads.

10. Adjust the position of the crosshead so that the lower bolt threaded rod is nearly seated.

11. For test Type 0 and R, attach and zero the extensometer reading in accordance with the device's instructions.

12. Run test with appropriate controls for testing machine.

13. If using an extensometer, remove the device once appropriate elongation has been reached.

14. Test the specimen until failure.

Testing speeds were deformation controlled and kept at 0.05 in./min. for 1/4 in. and M8 bolts and 0.1 in./min. for the larger sizes. Extensometer removal for Type R tests was programmed to occur at an elongation of roughly 15% of the original nominal diameter of the specimen. Extensometer removal for Type 0 tests was programmed to occur at 0.025 in. for 1/4 in. and M8 bolts and 0.05 in. for larger sizes. This was determined experimentally for the 1/4 in. bolt and limited by the used device's maximum range for the larger sizes.

3.2.4 Geometric Measurements

A variety of geometric measurements are required to characterize the bolts enable comparisons to existing design equations.

Major External Thread Diameter

1. Place the arms of a caliper on the threads, parallel to the axis of the bolt
2. Do not let the caliper arms touch the unthreaded portion of the bolt
3. Take measurements in the same orientation while the bolt is pinned between each set of hex-head faces (see Figure 3.5)

Body Diameter

1. Place the arms of a caliper on the unthreaded body roughly 1 diameter away from the threaded section
2. Keep the caliper arms perpendicular to the axis of the bolt
3. Rotate the bolt to seat the caliper
4. Repeat the steps above for each of the three pairs of hex-head faces (see Figure 3.5)

External Pitch Diameter (Three Wire Method)

1. Take out three of the appropriately sized wires for the bolt's TPI – consult chart found with wire set
2. Record the diameter of the wires (this will be constant for all bolts of one TPI)
3. Clamp the bolt in place
4. Adjust the micrometer to slightly larger than the expected measurement over the wires

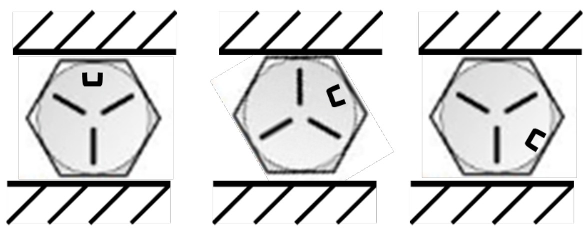


Figure 3.5: Pairs of Faces for a Grade 5 Bolt

5. Place two wires into adjacent thread notches on the bolt (see diagram found with wire set for appropriate placement) near the middle of the threaded length
6. Hold these in place against the bolt with the stationary anvil of the micrometer
7. Check that the two bottom wires are being held against the threads equally*
8. Slip the third wire into its place on the opposite side fo the bolt threads (see diagram)
9. Tighten the micrometer across all three wires using the smallest part of the adjusting screw†
10. Take your reading once the micrometer has been tightened until at least three audible clicks from the small end of the micrometer have been heard without a change in the displayed measurement
11. Calculate the pitch diameter as

$$E = M + 0.86603P - 3W \tag{3.1}$$

    where E = pitch diameter, M = measurement, P = pitch, and W = wire diameter.
12. Figure 3.6 shows the three wire method performed correctly.
13. Take measurements in the same orientation while the bolt is pinned between each set of hex-head faces (see Figure 3.5)

*Try to slide them around, and if one slides while the other doesn't, then the micrometer is not flat against them. Adjust the angle of the micrometer until both are held stationary. Make sure this does not change between steps.

†Ensure that the wires are nominally parallel, and that the micrometer is measuring across the middle of the bolt

Figure 3.6: Three Wire Method

Internal Minor Diameter

1. Insert the internal micrometer's arms into the hole of the part
2. Rest one of the arms against one side of the hole over as many threads as possible
3. Open the micrometer up as far as possible, resting the opposite arm against as many threads as possible
4. Repeat the steps above for each of the three pairs of hex-head faces (see Figure 3.5)

3.2.5 Determination of Yield and Ultimate Load

For all tests (excluding Type R, which is governed by ASTM E8), the peak load from the UTM output file shall be taken as the peak before the first significant drop in load (more than 5% of overall maximum load). This decision comes from the behavior of some bolts during Type 6 and 8 tests, where the initial peak load was surpassed after significant load loss due to the interaction of remaining intact threads on the bolt and edges of the stripped threads of the washer. See Figure 3.7 for an example plot of load vs crosshead displacement where this occurs.

Yield load was determined only for tests with an extensometer (Type 0 and Type R). This criterion was calculated following the 0.02% offset method in accordance with ASTM F606.

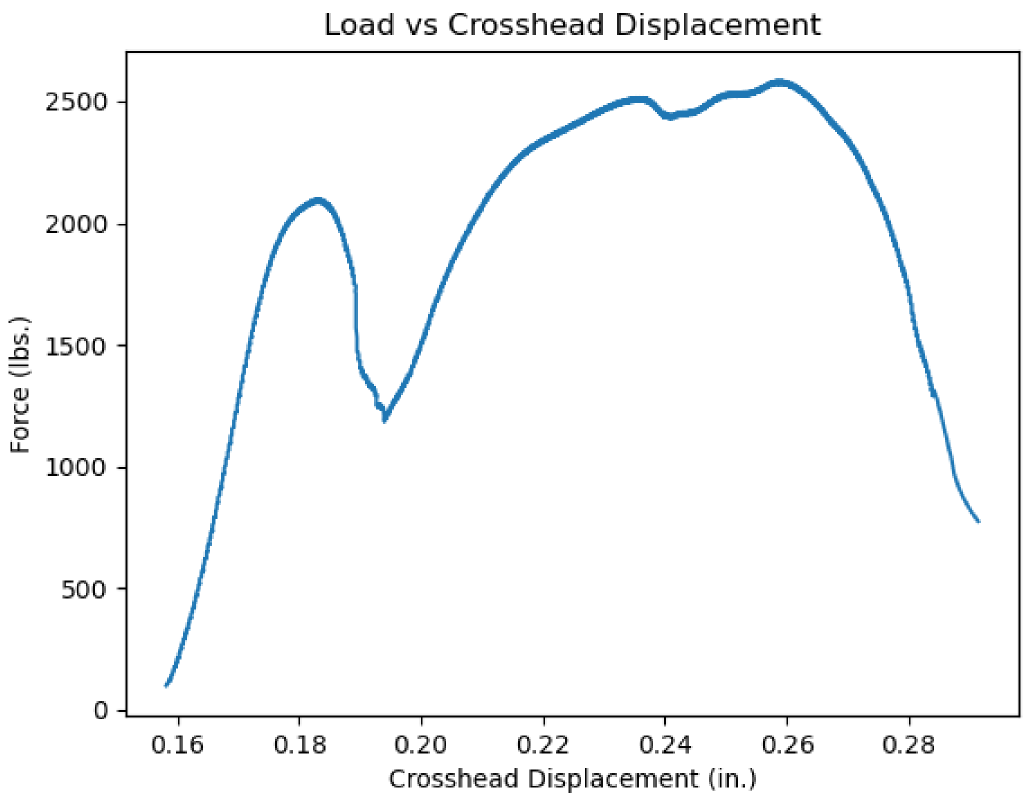


Figure 3.7: Test L003-T806. The initial peak load is surpassed, but the initial peak is the value relevant to the behavior of interest.

Chapter 4

Results and Discussion

4.1 Results

4.1.1 Tensile Stress Area

The experimental tensile stress area was calculated as the average load from Type 0 tests divided by the average material strength from Type R tests. Several plots comparing the experimental results to the predicted tensile stress area as calculated in Equation 2.1 for each lot of bolt tested. The error bars in these figures represents the propagated uncertainty using the 90% confidence intervals from Type R and Type 0 tests. The first plot presents the results as calculated. Each progressive plot normalizes the data in various ways. First the results are divided by the nominal area. Then the results are recalculated as the tensile stress diameter by solving the equation of circular area for diameter. These tensile stress diameters are then normalized according to Equation 2.3. The y-axis for this plot is presented in fractions of H, the height of the sharp v-thread. This was to make comparison to the tensile stress area equations easier, as they adjust the tensile stress area circle with some fraction of H.

This progression of normalization is presented for the original eight Grade 5 bolts tested. The final iteration of the figures are also presented for the two repeat Grade

5 bolts (with their original counterparts) and the four Grade 8 bolts. All six of these figures are shown in Figures 4.1 through 4.6.

4.1.2 External Thread Shear Area

A series of graphs were generated for the results of Type 5 tests. First, on a lot-by-lot basis, the strengths for the tests at each target length of engagement were grouped together and the 90% confidence interval was calculated. A linear regression was then performed with the mean of these confidence intervals and the origin on axes of strength vs length of engagement. Two graphs were produced, one for each standard examined. The FED-STD graph included equations 2.6 and 2.8, as well as the predicted length of engagement. The ISO graph included Equation 2.10, evaluated at the minimum material condition and without including the correction for chamfer or bell mouthing (as the chamfer on both the bolt and nut were removed). Both graphs included the linear regression with the confidence intervals and the mean ultimate tensile strength from Type 0 tests. All values were normalized by nominal diameter, d_{bsc}.

Type 7 results were also included on these initial graphs. The bounding tests (where the failure mode was changing each test) were used to determine a range of length of engagements and a range of strengths. These ranges were graphed as a box over the Type 5 results. The boundaries of this box were determined as follows:

- The left bound of this box was the average length of engagement for the largest increment for which thread shear failure was consistently observed.
- The right bound of this box is the average length of engagement for the smallest increment for which tensile failure was consistently observed.
- The bottom bound of the box is the average of the strengths for the largest increment for which thread shear failure was consistently observed.

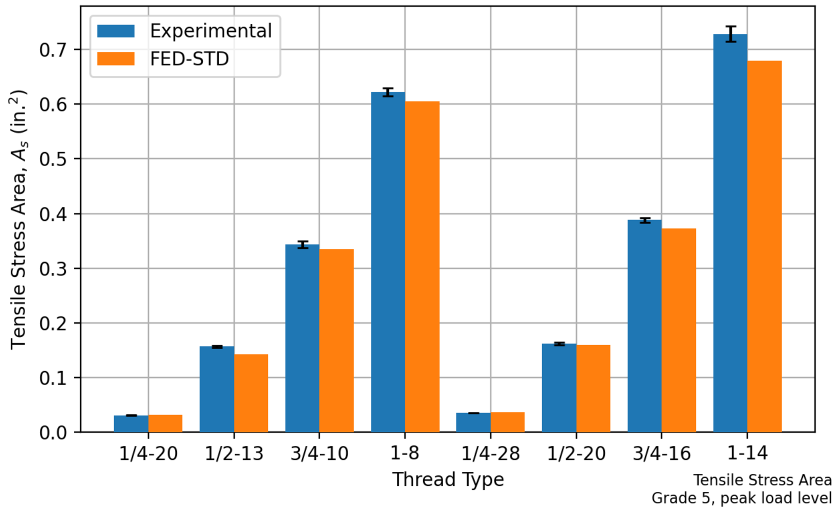


Figure 4.1: Tensile stress area

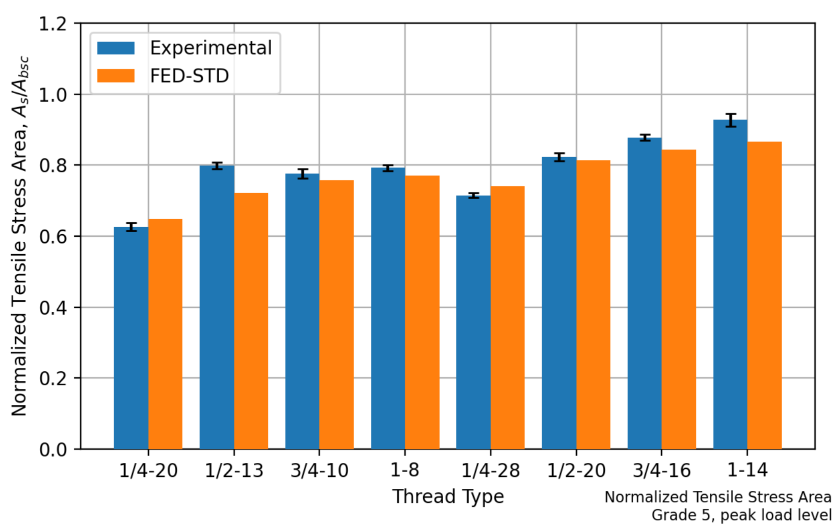


Figure 4.2: Normalized tensile stress area

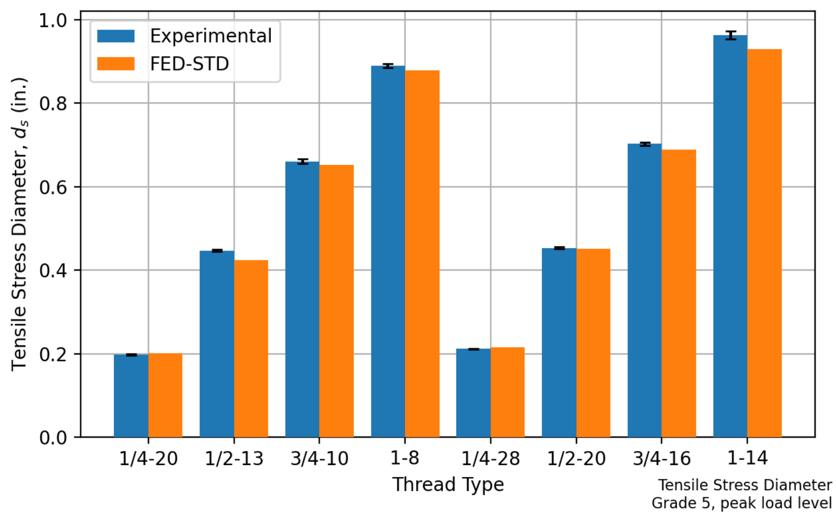


Figure 4.3: Tensile stress diameter

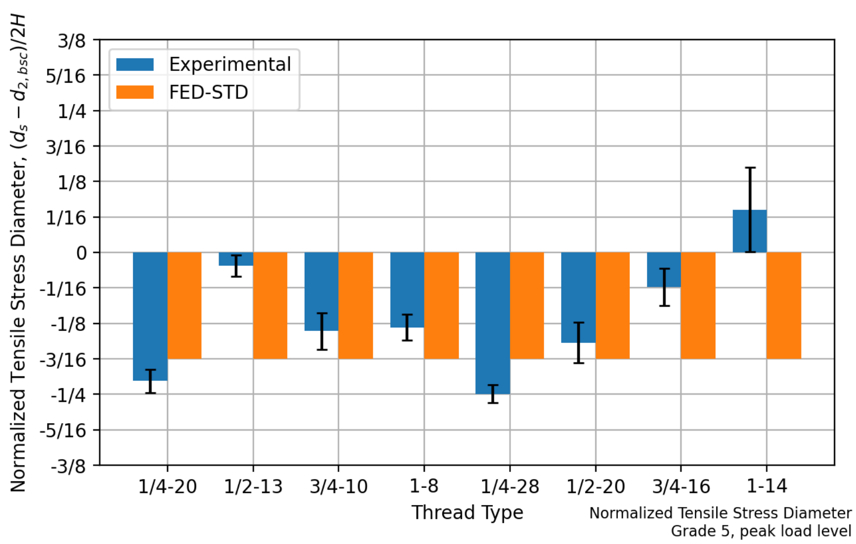


Figure 4.4: Normalized tensile stress diameter

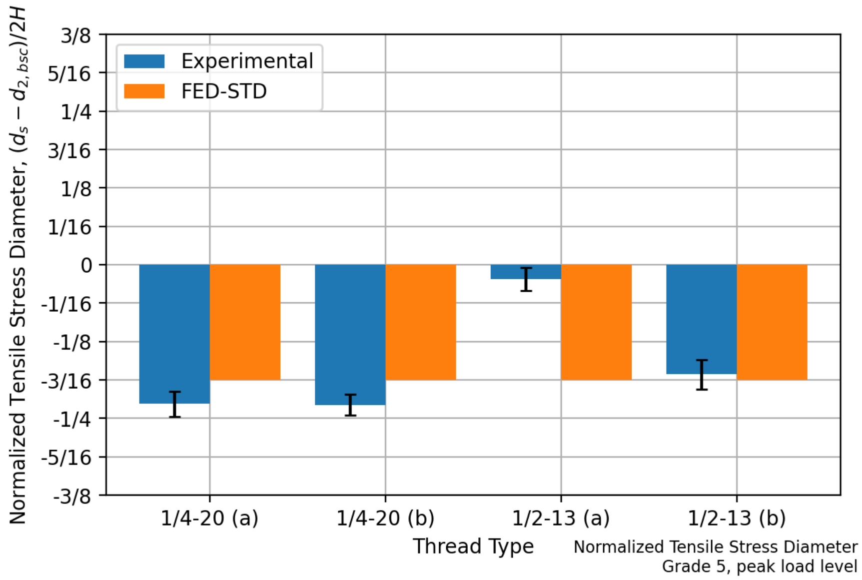


Figure 4.5: Normalized tensile stress diameter, repeat lots

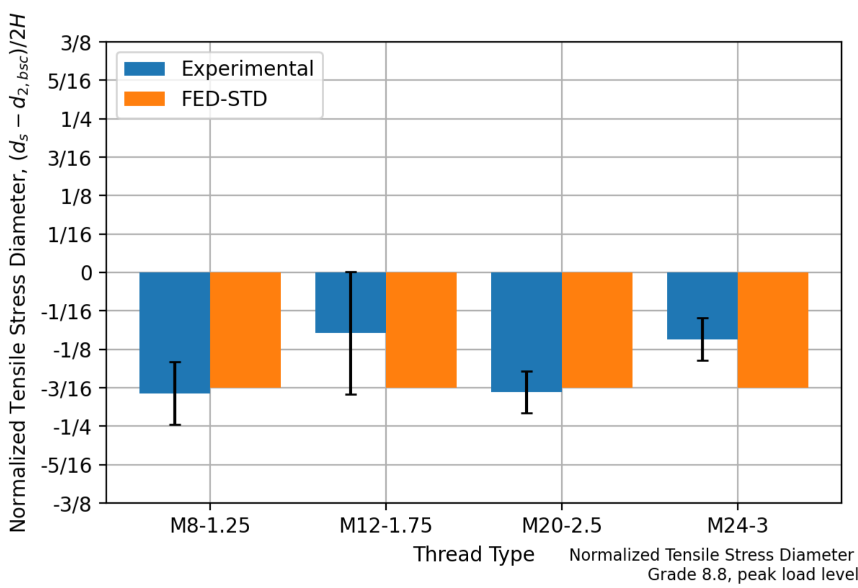


Figure 4.6: Normalized tensile stress diameter, metric lots

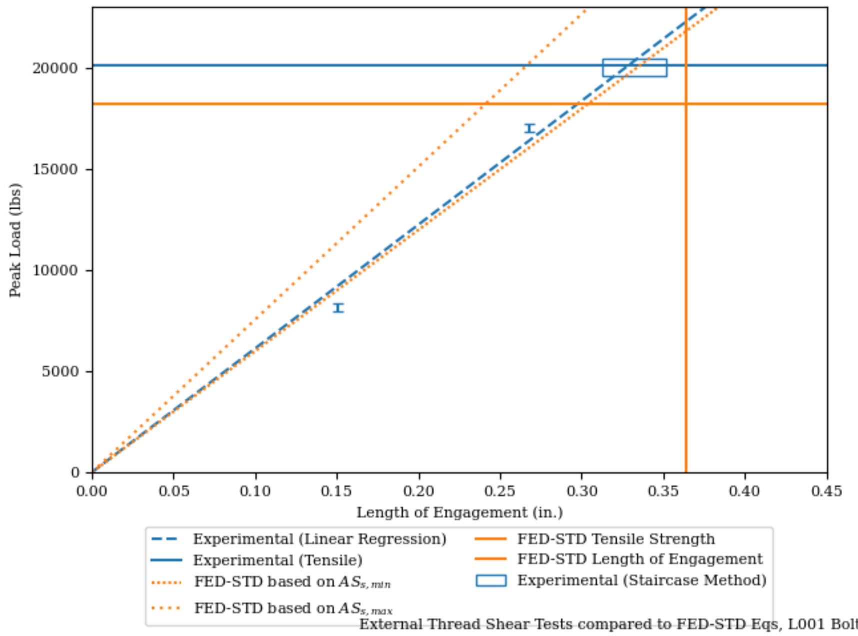
- The top bound of the box is the average of the strengths for the smallest increment for which tensile failure was consistently observed.

Figure 4.7 is an example of these single lot graphs.

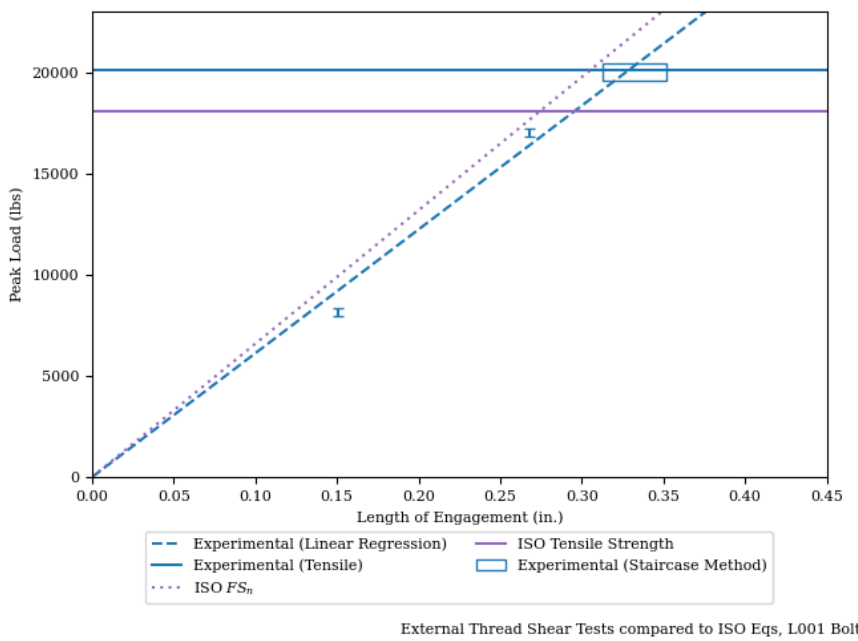
A second set of graphs were generated to assist comparison of the equations across lots. Two graphs were made. As shown in Figure 4.8, the slope of the linear regression is divided by the material strength of the lot, giving the tensile stress area per unit length of engagement. FED-STD Equation 2.11 and ISO Equation 2.14 are included and evaluated at the same conditions as the previous graphs. A second instance of the ISO equation is included, this time multiplied by appropriate C-factors, C_1 and C_2. All of these values are normalized by nominal diameter, d_{bsc}. As shown in Figure 4.9, the experimental ultimate tensile load was divided by the slope of the linear regression to achieve an estimation of the experimental length of engagement. The predicted length of engagement per FED-STD and the length of engagement predicted by ISO found through reverse engineering were also included. All values were again normalized by nominal diameter, d_{bsc}.

4.1.3 Internal Thread Shear Area

The series of graphs generated for the results of Type 6 and Type 8 tests is much the same as presented in Section 4.1.2. Two initial graphs were generated for every lot and every nut material (for a total of four plots). The first included FED-STD Equations 2.11 and 2.13 and the predicted length of engagement, and the second included ISO Equation 2.14 evaluated at the minimum material condition without the correction for bell mouthing. Both included the experimental results (generated the same as described above), the mean ultimate tensile strength from Type 0 tests, and the predicted tensile strength. All values were normalized by nominal diameter, d_{bsc}. Example graphs are not provided, as they are formatted identically to Figure 4.7.



(a) FED-STD equations



(b) ISO equations

Figure 4.7: External thread shear test results, 1/2-13

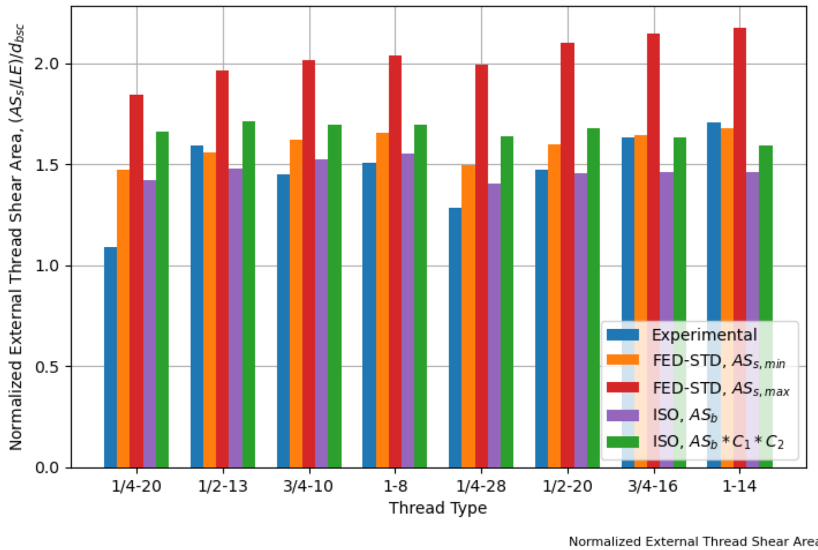


Figure 4.8: Normalized external thread shear area comparison by lot

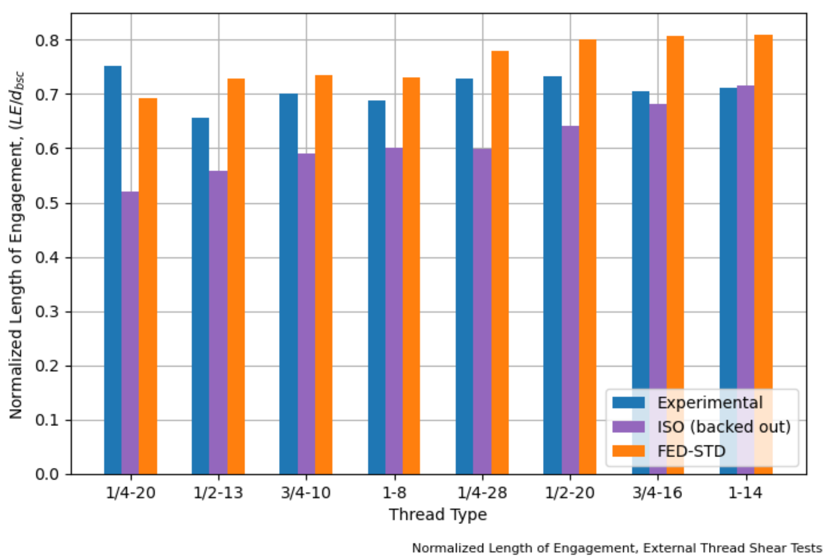


Figure 4.9: Normalized external thread shear length of engagement comparison by lot

As before, a second set of graphs were generated to assist comparison of the equations across lots. Two graphs were made per nut material (for a total of four plots). The first set takes the slope of the linear regression and divides it by the material strength of the lot, giving the tensile stress area per unit length of engagement. FED-STD Equation 2.11 and ISO Equation 2.14 are included and evaluated the same as described in the previous plots. A second instance of the ISO equation is included, this time multiplied by appropriate C-factors, C_1 and C_2. All of these values are normalized by nominal diameter, d_{bsc}. For the second set of graphs, the experimental ultimate tensile load was divided by the slope of the linear regression to achieve an estimation of the experimental length of engagement. The predicted length of engagement per FED-STD and the length of engagement predicted by ISO found through reverse engineering were also included. All values were again normalized by nominal diameter, d_{bsc}. The four graphs are depicted in Figures 4.10 through 4.13.

4.2 Discussion

4.2.1 Tensile Stress Area

The normalized stress diameter plots show that the FED-STD Equation (and by extension the ISO Equation) for tensile stress area under-predicted the strength for all but the 1/4, M8, and M20 bolts. For a few cases (namely 1/2-13 and 1-14) the equations under-predicted strength to a greater degree than others. These two lots had the largest coefficient of variation in peak loads from Type 0 testing, which could impact these results.

The repeat lots of Grade 5 bolts affirm that the equations over-predict strength for 1/4-20. Seeing that the questionable results are mostly contained to the smaller bolt sizes, it is possible that smaller bolts require a special adjustment or separate equation that predicts smaller tensile stress areas. Further results are necessary to

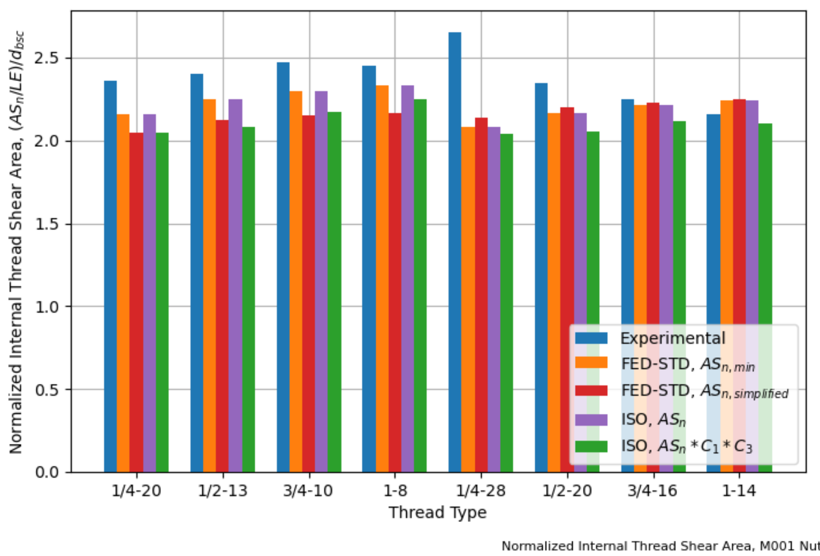


Figure 4.10: Normalized internal thread shear area for A36 washer comparison by lot

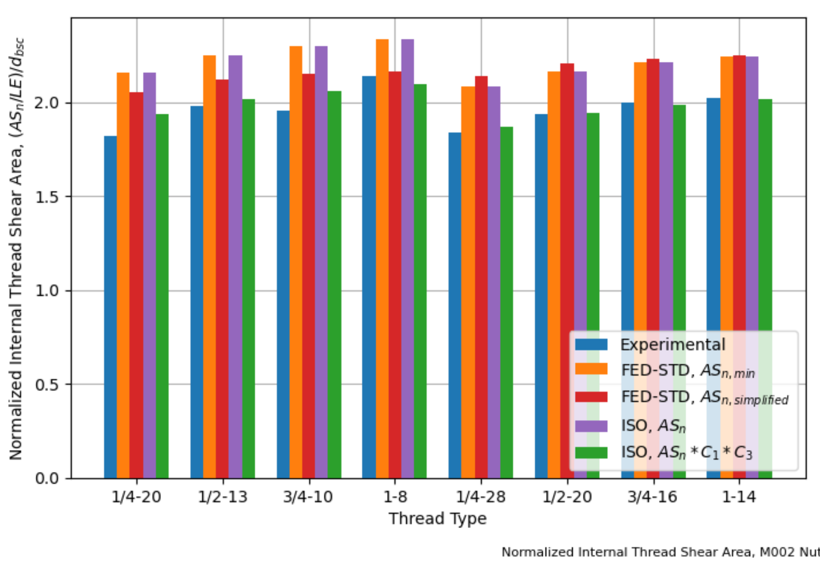


Figure 4.11: Normalized internal thread shear area for HY80 washer comparison by lot

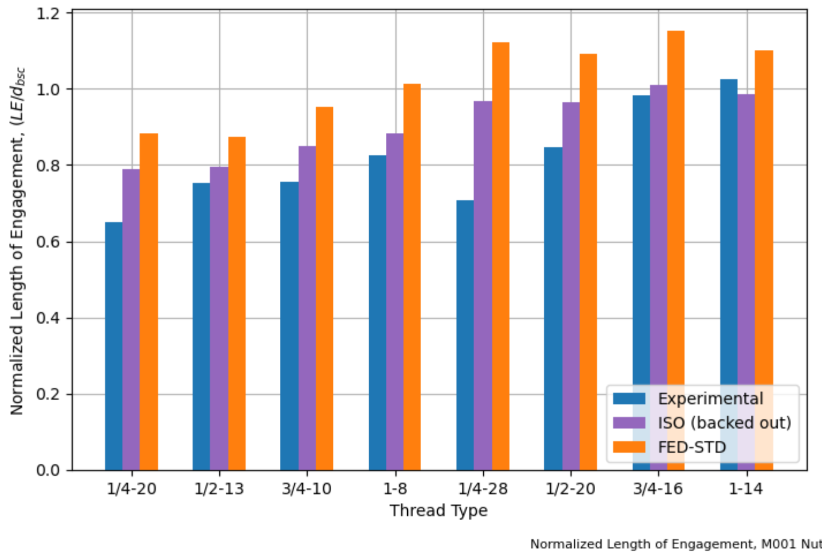


Figure 4.12: Normalized internal thread shear length of engagement comparison by lot for A36 results

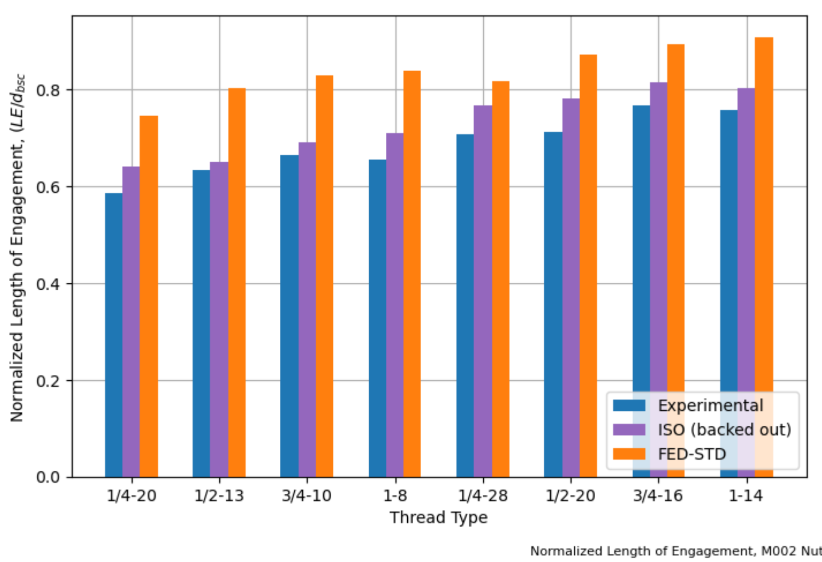


Figure 4.13: Normalized internal thread shear length of engagement comparison by lot for HY80 results

propose specific adjustments or explanations for this behavior, especially considering that M20 bolts also under-predicted results.

The variation between the two lots of 1/2-13 bolts is interesting. Considering group (b) has a tested material strength roughly 10,000 psi than group (a), there is a strange lack of increase in Type 0 test strength. The two groups also displayed different ratios of yield strength to tensile strength, with group (a) being 1.26 and group (b) being 1.15. A difference in elongation was also measured, with group (b) elongating roughly 3% less on average. Which of these material differences had the greatest impact on this variation between results is unclear.

Overall, the tensile stress area equations show good agreement with the experimental results. In most scenarios they yield acceptable and conservative predictions without excessive margins of error.

4.2.2 External Thread Shear Area

As seen in plots like Figure 4.7, the external thread shear area was certainly proportional to length of engagement. FED-STD Equation 2.8 for the maximum material condition overestimated all other results by a wide margin. This made sense, as the vast majority of measured dimensions were on the lower end of the allowable range of tabulated dimensions. However, the minimum material condition overestimated most experimental results. FED-STD is more focused on giving a length of engagement that precludes thread shear failure, not in accurately predicting the thread shear strength. Should the length of engagement equations that use the external thread shear area accurately provide a required length of engagement that makes tensile failure more likely, an inaccurate measure of thread shear strength is less problematic.

The ISO equation showed varying degrees of accuracy across lots. Once multiplied by C-factors however, the predicted area increased somewhat dramatically, overestimating the area for all but the largest fine threaded lots. The critical length of engagement per ISO was consistently lower than the experimental results. The only difference between the uncorrected ISO equation and the FED-STD equation is the correction for bell-mouthing. This correction worked to make the prediction more conservative, but was undone by the C-factors.

The results of Type 7 tests often showed tensile failure loads being lower than previously seen in Type 0 tests. While this could have been due to variation in bolt material strength, the loads were outside the 90% confidence interval found during Type 0 tests. According to Alexander (1977), the only major effect on tensile load comes from applying torque to the bolt, which was not done in any testing for this thesis. Given that these tests were near the critical length of engagement, interaction between tensile and shear failures could have had an effect.

Interestingly, the results for the external thread shear area are seemingly less consistent than those for the tensile stress and internal thread shear areas. The FED-STD length of engagement equation is moderately acceptable. However the rest of the equations differ from the experimental results and show variation across the thread forms, reducing confidence in their accuracy.

Some measured internal minor diameter measurements exceeded the maximum tolerance. This would lower the external thread shear area of these nuts and thus the thread shear strength. Additionally, for some of the larger diameters, the prongs of the micrometer used to measure the internal minor diameter did not cover many threads, potentially introducing error into the measurement. It is unclear how much of an effect these phenomena had on the results.

4.2.3 Internal Thread Shear Area

The internal thread shear area was also proportional to length of engagement. FED-STD Equation 2.11 and ISO Equation 2.14 for internal shear area behaved the same, due to being identical when evaluated without accounting for chamfers at the minimum material condition. For A36 internal threads, these equations were conservative for all but the 1-14 bolts. The simplified FED-STD Equation 2.13 was further separated from these experimental results for the coarse bolts, but was in a closer margin then Equation 2.11 for the fine bolts. The ISO equation for strength (including the C-factors) was more conservative than all other equations for all lots. For HY80 internal threads, the equations for internal shear area were unconservative for all lots. However, the ISO equation for strength nearly exactly matched the experimental results for all lots.

The FED-STD length of engagement equation for the A36 nuts were conservative for all lots. For HY80 internal threads the equation was also conservative and there was less variability among the lots. The ISO length of engagement very closely matched the experimental results for both types of internal threads. This was to be expected, as this equation was backed out, and does not have any intentional buffer like FED-STD. However, it does demonstrate the consistency of the ISO equations and the predictability of the experimental results.

These results are very unsurprising and consistent, and lend high confidence to the equations from both standards. The reliability of the internal thread shear results casts some further doubt on the behavior of the external thread shear results. Mild improvements to the internal thread shear areas could be made, and some excess of the length of engagement for internal thread shear failure could be reduced. In contrast, it is much more difficult to identify direct improvements to the external thread shear area equations. Further investigation is necessary to increase the accuracy of the external thread shear equations. Better understanding of the C-factor for external thread bending and the material behavior of threads under shear would be pertinent.

Chapter 5

Probabilistic Evaluation of Length of Engagement

5.1 Background

ISO and Alexander (1977) both present a method for the determination of the minimum nut height for a bolt and nut combination with given material properties. In this method, a probabilistic analysis following a Monte Carlo simulation is used. Various parameters, both geometrical and material, of the bolt and nut are assumed to be randomly distributed inside their tolerances nearest to the minimum material condition. The distributions of the thread shear failure load and the tensile failure load for a nut height equal to one diameter of the bolt are then calculated for these sampled properties. Assuming a perfectly linear relationship between the stripping load and nut height, a line is drawn from the origin to the mean of the thread shear load. Where this line is equal to 95% of the mean tensile failure load (the 5% reduction is to account for torque loading) is taken to be the critical nut height. A random normal distribution is generated for this critical nut height. The minimum nut height is then set to the height where the probability of tensile failure is 10%. Tolerances are then applied to the minimum nut height to determine the range of nut heights for the relevant ISO standard. See Figure 5.1 for an illustration of this process.

The objective of the Monte Carlo simulation in this thesis is to quantify the probability thread failure at the FED-STD length of engagement. This probabilistic method followed many of the same assumptions outlined in the ISO method, but included statistical data from this study including on the accuracy of the strengths formulas themselves, a source of uncertainty that is not considered in ISO.

5.2 Method

The goal of a single Monte Carlo simulation in this thesis is to calculate the probability of the different possible tensile failure modes for a given bolt, nut, and, length of engagement.

The probability of failure of each failure mode is the number of realizations for which that failure mode was controlling divided by the total number of realizations. A failure mode was controlling if it had the lowest computed strength. Tensile strength was computed using Equation 2.4 multiplied by the ultimate tensile strength of the bolt and $X_{tensile}$, a random variable that describes the ratio of experimental tensile area to that predicted by Equation 2.4. This type of random variable is referred to as a "professional factor" in structural engineering literature (Ravindra and Galambos, 1978). External thread shear strength was computed using Equation 2.10 multiplied by C-factors C_1 and C_2, 60% of the ultimate tensile strength of the bolt, and a professional factor for external thread shear area $X_{thread,ext}$. Internal thread shear strength was computed using Equation 2.14 multiplied by C-factors C_1 and C_3, 60% of the ultimate tensile strength of the nut, and a professional factor for internal thread shear area $X_{thread,int}$.

To generate the professional factors, ratios were first calculated across lots. The mean and standard deviation of these ratios were then used to generate the random variable. The ratio for $X_{tensile}$ was of experimentally determined tensile stress area

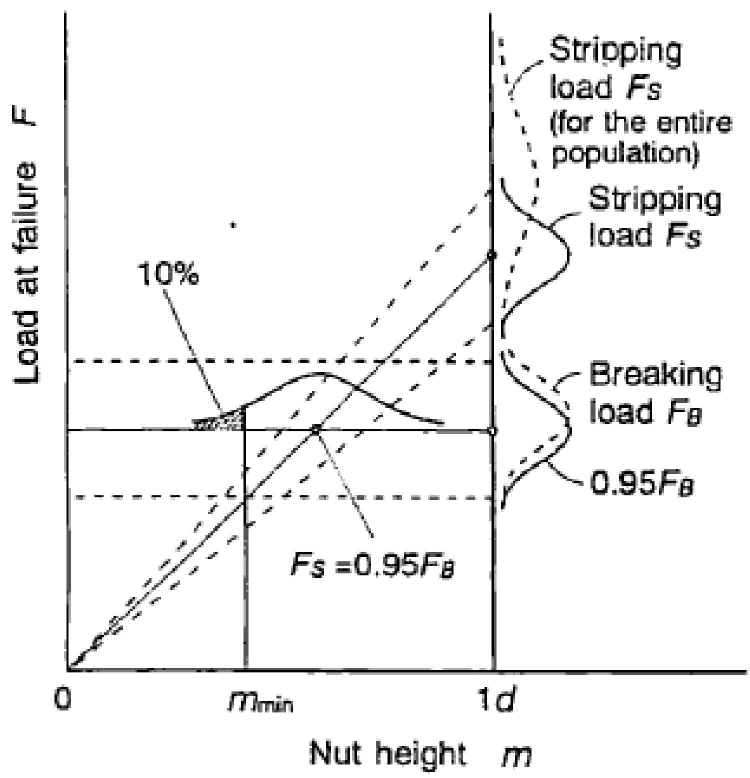


Figure 5.1: Original Monte Carlo method (Hagiwara et al., 1995)

to the predicted area as per Equation 2.4. The ratio for $X_{thread,int}$ was of the experimental results of both Type 6 and Type 8 testing to the predicted area as per Equation 2.14. The ratio for $X_{thread,ext}$ was of the experimental results for Type 5 tests to the predicted area as per Equation 2.10.

Thread pitch, length of engagement, and width across the flats of the nut were assumed to be deterministic. Internal and external pitch diameter, Internal and external minor diameter, the external major diameter, the chamfer diameter, and the ultimate tensile strength of the bolt and the nut were assumed to be random variables.

In line with the ISO method, the variables describing geometrical properties were assumed to have total deviations (six standard deviations) as percentages of the tolerance described in ASME (2019). These deviations are reported in Table 5.1. For internal geometry, the mean was calculated as three standard deviations subtracted from the maximum tolerance. For external geometry, the mean was calculated as three standard deviations added to the minimum tolerance. This was done to approximate minimum material conditions.

Internal pitch diameter, internal minor diameter, and the chamfer diameter were assumed to be perfectly correlated to each other. External pitch diameter, external minor diameter, and external major diameter were assumed to be correlated to each other.

The bolt and nut material strength variables used the ultimate tensile strength multiplied by the expected tensile strength ratio as the mean. The expected tensile strength ratio for bolts was calculated as the ratio of tested bolt material strength to specified minimum strength. This ratio was taken as 1.15. The expected tensile strength ratio for nuts was taken as 1.2 based on data for A36 material. This data is from Liu et al. (2007), which contains expected tensile strength ratio data for many common steels used in building construction.

Tables 5.2 and 5.3 show the expected tensile strength ratios and professional factor data for every lot of bolts, separated by bolt grade.

Table 5.1: Assumed deviations for variables

Variable	For Nuts	For Bolts
Major diameter	-	20% of the tolerance
Pitch diameter	30% of the tolerance	25% of the tolerance
Minor diameter	50% of the tolerance	100% of the tolerance
Countersink diameter	$0.01D_{bsc}$	-

Table 5.2: Lot and average error adjustment ratio data for Grade 5 bolts

Bolt Type	Tensile Strength (psi)	Tensile Strength Ratio	Tensile Stress Area Ratio	External Shear Area Ratio	Internal Shear Area Ratio, Type 6	Internal Shear Area Ratio, Type 8
1/4-20(a)	138700	1.156	0.950	0.657	1.153	0.941
1/4-20(b)	144600	1.205	0.948	-	-	-
1/4-28	148900	1.241	0.953	0.783	1.302	0.983
1/2-13(a)	128100	1.067	1.091	0.928	1.153	0.982
1/2-13(b)	136800	1.140	0.993	-	-	-
1/2-20	136700	1.139	1.003	0.879	1.142	0.997
3/4-10	136500	1.137	1.015	0.854	1.138	0.949
3/4-16	142400	1.187	1.033	1.001	1.061	1.006
1-8	144600	1.205	1.017	0.890	1.088	1.022
1-14	134700	1.123	1.065	1.070	1.025	1.003
Mean	139200	1.160	1.007	0.883	-	1.059
St. Dev.	6011	0.050	0.048	0.127	-	0.097
COV	0.043	0.043	0.048	0.073	-	0.092

Table 5.3: Lot and average error adjustment ratio data for Grade 8.8 bolts

Bolt Type	Tensile Strength (psi)	Tensile Strength Ratio	Tensile Stress Area Ratio
M8-1.25	141200	1.217	0.981
M12-1.75	133900	1.154	1.040
M20-2.50	135600	1.168	0.987
M24-3.00	136700	1.178	1.029
Mean	136900	1.180	0.999
St. Dev.	3117	0.027	0.026
COV	0.023	0.023	0.026

The calculations described above were implemented in a Python function. Each simulation used 100,000 realizations. This function was called many times over a range of lengths of engagements. Probability curves were generated alongside vertical lines representing the critical length of engagement as per ISO and the required length of engagement as per FED-STD.

5.3 Results and Discussion

Throughout this section, several plots will be shown. Certain parameters of the simulation not previously specified will have consistent values for many of these plots. These parameters and their default values are as follows: the bolt minimum specified tensile strength was $UTS_s = 120000$; the number of data points along the length of engagement range was 50.

Figure 5.2 shows the plot generated for a scenario in which external thread shear is expected (Minimum nut tensile strength, $UTS_n = 180000$psi) for a 1/2-13 bolt. As expected, the probability of thread shear failure is very high for short lengths of engagement and gradually decreases as length of engagement increases. The two failure modes cross at 50%, indicating where the equations have found equal strengths and thus the two failure modes are equally likely.

The difference between the length of engagement where 100% thread shear occurs and 100% tensile failure occurs is roughly 0.25 inches. This is quite large in comparison to a similar plot presented in Alexander (1977), shown in Figure 5.3, in which the probability curves cover roughly 0.6 millimeters (0.024 inches). This increased spread is due to the professional factors introduced, capturing uncertainty in the equations that was not captured previously. To demonstrate this further, a plot is generated without including the professional factors, shown in Figure 5.4. This plot was generated with 200 data points for the probability lines to increase curve clarity. Without the professional factors, the probability spread is now roughly 0.03 inches,

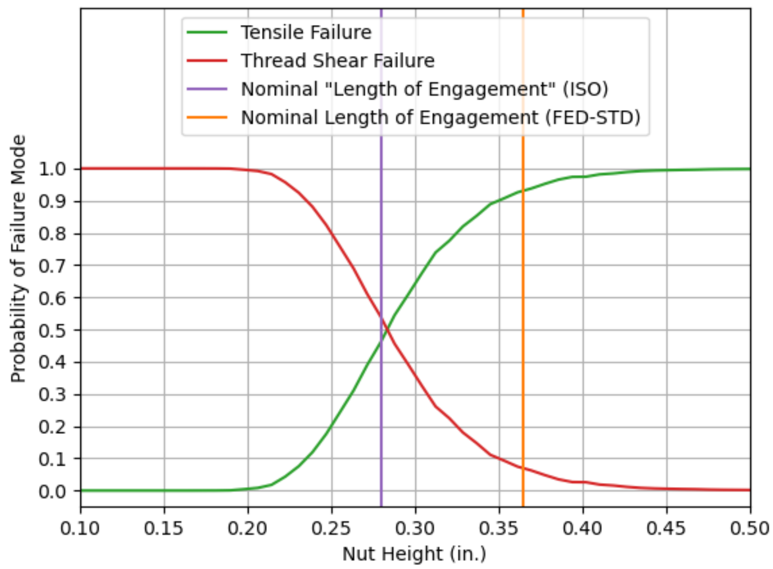


Figure 5.2: Probability curve for external thread shear conditions

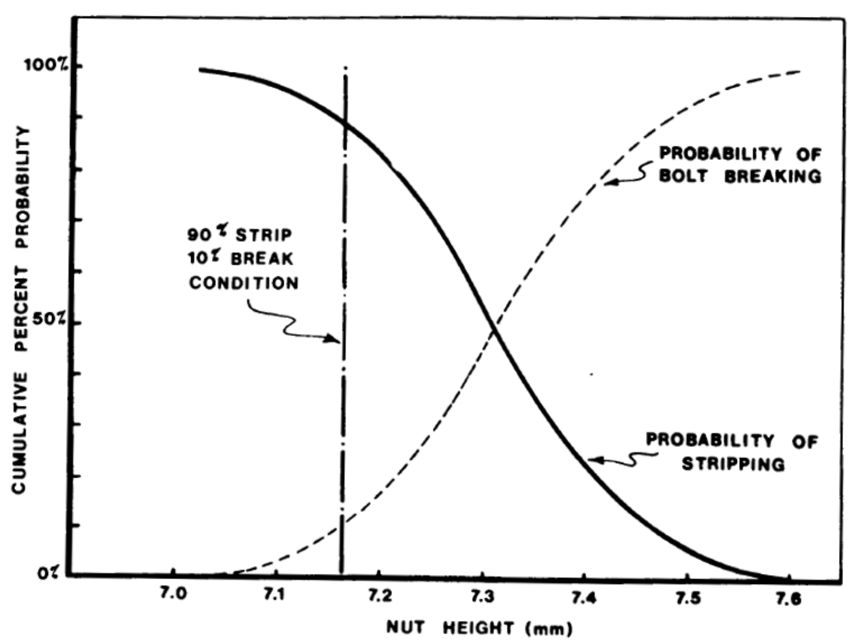


Figure 5.3: Probability curve from Alexander (1977)

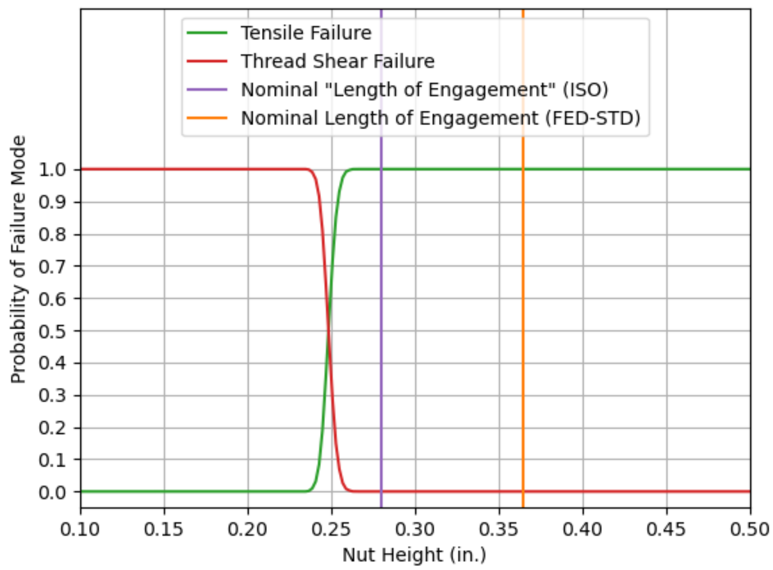


Figure 5.4: Probability curve for external thread shear conditions, sans professional factors

which is much closer to Alexander's original method. This provides some confidence that the code is properly following the original process.

Returning to Figure 5.2, the lengths of engagement from ISO and FED-STD line up in logical places in relation to the probability curves. Because the ISO equation for length of engagement has been reverse engineered as the location of equal likelihood for the two failure modes, it is unsurprising to see it cross the probability curves at near 50%. Adding to this, the ISO equations form the basis of the Monte Carlo simulation, giving even more cause for this alignment. The FED-STD equation on the other hand includes a built in factor increasing the required length of engagement, as it is designed to preclude bearing failure. This is reflected in Figure 5.2, as the FED-STD equation lines up with about a 93% chance of tensile failure.

These trends can be seen once again in Figures 5.5 and 5.6, which show the plot for nut materials of minimum tensile strength 58000 psi and 110000 psi (emulating A36 and HY80 materials, respectively). For Figure 5.5, the probability of tensile failure seen from the FED-STD equation is markedly lower than seen in the other plots, reading around 70%. This would likely be an undesirably high probability of thread shear failure.

Table 5.4 collects the probability of tensile failure at the FED-STD required length of engagement for Grade 5 bolts tested for the three scenarios presented so far (external, internal A36, and internal HY80).These results show that the FED-STD equation can very quite a bit in its predicted probability. Whether this is an indication of the equation needing improvement or the probabilistic model needing refinement is hard to say. However, it demonstrates some uses of the model. Were an engineer to use these results in the design of a threaded connection, they may justify increasing the required length of engagement to further avoid tensile failure. Additionally, this would provide a set of testing conditions a future study could investigate with the purpose of evaluating both the accuracy of this probabilistic model and that of the FED-STD equation.

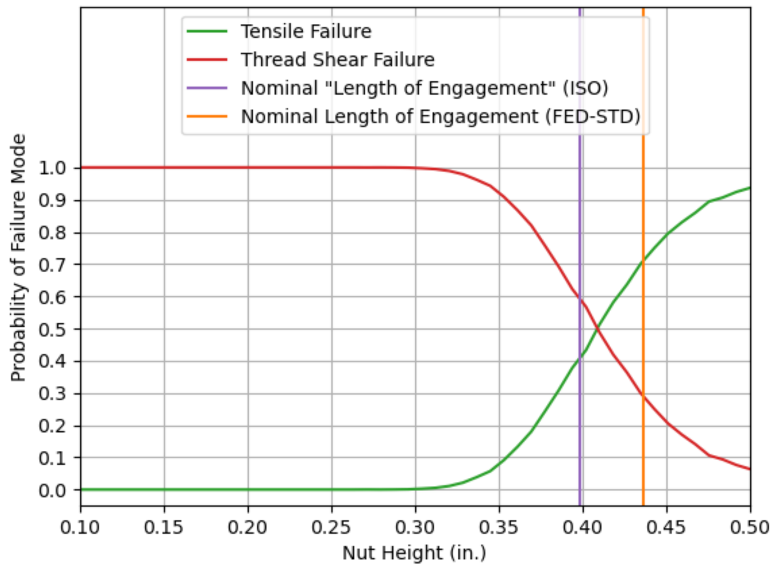


Figure 5.5: Probability curve for internal thread shear conditions, A36 nut material

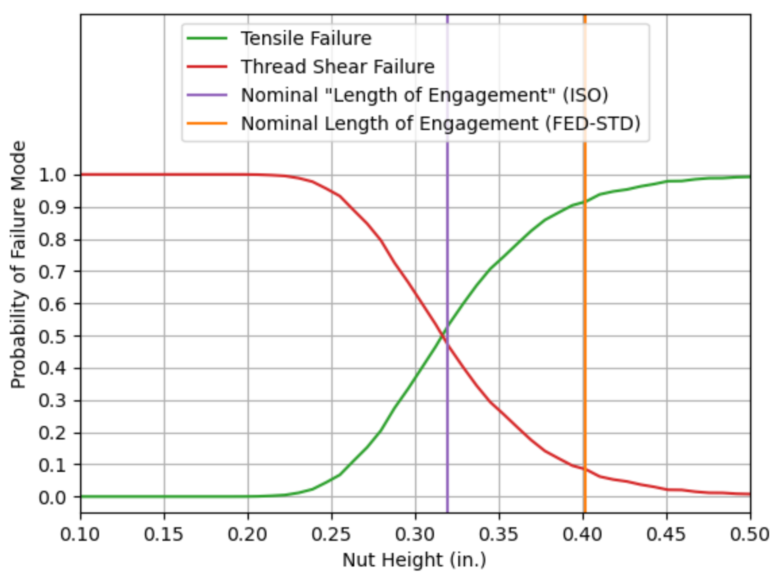


Figure 5.6: Probability curve for internal thread shear conditions, HY80 nut material

Table 5.4: Predicted probability of tensile failure at LE_r as per FED-STD. All values as percentage

Bolt Type	UTS_n = **180ksi**	UTS_n = **100ksi**	UTS_n = **58ksi**
1/4-20(a)	95.0	92.7	89.7
1/4-20(b)	95.2	92.8	94.2
1/4-28	94.0	87.0 96.8	
1/2-13(a)	93.3	91.8	70.9
1/2-13(b)	93.1	91.9	85.3
1/2-20	65.0	55.4	48.6
3/4-10	54.7	68.9	38.9
3/4-16	55.3	51.7	52.3
1-8	65.1	70.1	55.3
1-14	73.5	75.1	63.6

Chapter 6

Conclusions

This thesis sought to validate existing design equations for bolts in tension through physical testing. The differences in form and underlying assumptions between FED-STD and ISO were identified and contrasted. An array of tests were employed to generate a wealth of physical data for the limit states of tensile failure, external thread shear failure, and internal thread shear failure. A probabilistic approach for determining a desirable length of engagement using Monte Carlo simulation was performed, adapting the established method and giving further insight into the performance of the design equations.

Several conclusions were drawn from the results presented:

- Physical results for the tensile stress area showed good agreement with the existing equations.
- Physical results for the internal thread shear area showed good agreement with the existing equations.
- The C-factors in the ISO equations worked to increase accuracy of the equations for internal thread shear results.
- The external thread shear area equations showed poorer agreement with the existing equations

- Potential sources of discrepancy include the C-factors specific to external thread shear and the behavior of threads under shear

- The FED-STD length of engagement equations over-predicted length of engagement for internal thread shear failure

- The FED-STD length of engagement equations over-predicted strength for external thread shear failure

- The probabilistic evaluation of length of engagement shows potential as a guide for future investigations into the FED-STD equations.

Overall, these results contribute to an increased confidence in the design of bolts in tension, leading towards increased safety. Furthermore, the data generated will be made available for reference, ensuring long-term availability for comparison and evaluation by others. The methodology established in this thesis can also be used in future studies to evaluate other grades of bolts, determine if an adjustment to the tensile area equation is necessary for smaller bolt sizes, and further investigate the accuracy of the C-factors for external thread shear failure.

Bibliography

Alexander, E. M. (1977). Analysis and Design of Threaded Assemblies. *SAE Transactions*, 86:1838–1852. x, 7, 8, 57, 59, 64, 65

ASME (2019). Unified Inch Screw Threads (UN, UNR, and UNJ Thread Forms). ASME B1.1-2019, ASME, New York, N.Y. 62

Dyson, C., Hopkins, W., Aljeran, D., Fox, M., and Priest, M. (2023). Tribological considerations of Threaded Fastener Friction and the importance of Lubrication. *Tribology International*, 191. 7, 8

Grimsmo, E., Aalberg, A., Langseth, M., and Clausen, A. (2016). Failure modes of Bolt and Nut assemblies under Tensile loading. *Journal of Constructional Steel Research*, 126:15–25. 8, 9

GSA (1991). Federal Standard: Screw-Thread Standards for Federal Services Section 2 Unified Inch Screw Threads - UN and UNR Thread Forms. FED-STD-H28/2B, Federal Supply Services, General Services Administration, Washington, DC. 2, 9

Hagiwara, M., Itoh, S., and Kondoh, S. (1995). Loadability of Bolt-Nut Assemblies – Determination of Critical Nut Height by Staircase Method. *International Journal of the Japan Society for Precision Engineering*, 29(4):356–360. x, 9, 61

ISO (1994). Mechanical Properties of Fasteners - part 6. ISO 898-6:1994, International Organization for Standardization, Switzerland. 9

ISO (2012). Technical Aspects of Nut Design. ISO/TR 16224:2012, International Organization for Standardization, Switzerland. ix, 2, 9, 17

Liu, J., Sabelli, R., Brockenbrough, R., and Fraser, T. (2007). Expected yield stress and tensile strength ratios for determination of expected member capacity in the 2005 AISC seismic provisions. *Engineering Journal*, First Quarter:15–25. 62

Ravindra, M. K. and Galambos, T. V. (1978). Load and Resistance Factor Design for Steel. *Journal of the Structural Division*, 104(9):1337–1353. 60

Yang, F., Vljkovic, M., and Liu, Y. (2021). Fracture simulation of partially threaded bolts under tensile loading. *Engineering Structures*, 226. 8

Vita

Having grown up in middle Tennessee, Paxton Lifsey was always familiar with the University of Tennessee. He chose to attend UT for his undergraduate career and received a Bachelor of Science degree in civil engineering in 2022. Originally, Paxton was enrolled in the 5-year Master's program. However, after working under Dr. Denavit in undergraduate research his senior year, he was given the opportunity to become a graduate research assistant. Thus, he pursued his Master of Science degree in structural engineering over 2 years. After graduation, Paxton aims to find employment with a structural design firm and eventually earn his Professional Engineer license. He is deeply grateful for the guidance and mentorship from Dr. Denavit, as well as the support from his friends and family as he continues pursuing his goals.