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U.S. Nuclear Regulatory Commission
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Attn: Mr. Christopher Grimes

Subject: Transmittal of EPRI Reports
Reference: Project 690

Per your request, enclosed is one (1) copy of each of the following EPRI reports:

1. TR-104213, "Bolted Joint Maintenance & Applications Guide," December, 1995 (NOTE: This report has superseded NP-5067)
2. NP-7079s, "Instrument Air Systems," December, 1990

EPRI requests that you make these reports publicly accessible so that their contents can be distributed to the general public and EPRI waives all license agreement requirements to achieve that end.

Should you have any questions regarding this submittal, please feel free to contact me at 650/855-2105.

Sincerely,

John Carey

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Program Manager

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D042

NMAC

EPRI

Bolted Joint
Maintenance & Applications Guide

LICENSED MATERIAL

(continued on back cover)

EPRI Leadership in Science and Technology

REPORT SUMMARY

Bolted Joint Maintenance & Applications Guide

This guide consolidates key information previously published by NMAC. The primary focus is on the design and assembly of pressure boundary, mechanical, and structural bolted joints. This document helps plant engineering and maintenance personnel with the assembly and inspection of various types of bolted connections used in nuclear power plant applications.

INTEREST CATEGORIES

Maintenance practices
Nuclear plant operations and maintenance
Engineering and technical support

KEYWORDS

Design
Assembly
Bolted connections
Threaded fasteners

BACKGROUND In the past 20 years, the NRC has issued numerous documents highlighting concerns with the integrity of bolted connections including pressure boundary, mechanical, and structural joints. The commercial nuclear power industry has responded with initiatives to resolve the issues. Numerous Electric Power Research Institute (EPRI) and Nuclear Maintenance Applications Center (NMAC) publications were recognized by NUREG-1399 as contributing to the resolution of concerns. Significantly, the NRC emphasized the usefulness of utility plant-specific programs and the need to continue to address the issues identified in previous NRC guidance.

OBJECTIVES

- To update, augment, and consolidate previously published NMAC information
- To provide additional technical information to allow seamless integration of the material into a single document

APPROACH This document contains tables with information obtained from various referenced sources. The objective was to consolidate and reproduce information as a handy reference. Information on design and assembly includes guidance and illustrations that convey key concepts and critical success factors. Effort has been made to illustrate the relationship between the design and assembly processes.

RESULTS This document is divided in parts or topical sections for pressure-retaining joints, mechanical joints, structural joints, and threaded fasteners. Chapters in each part cover design considerations, assembly, inspection, and troubleshooting and repair. The section on threaded fasteners addresses selection, specification and procurement, and receipt inspection.

EPRI PERSPECTIVE Proper design, assembly, preload, and inspection of bolted connections remains an important activity for operators of commercial nuclear power plants. Likewise, plant leakage reduction efforts continue to receive attention at most of these generating facilities. NMAC's *Bolted Joint Maintenance & Applications Guide* addresses these areas of interest and represents both a major revision and a consolidation of several previous guidebooks dealing with general good bolting practices and guidelines for the use of threaded fasteners.

PROJECT

WO 3814-07

EPRI Project Manager: J.M. Jenco

Nuclear Maintenance Applications Center / Nuclear Power Group

Contractor: Aptech Engineering Services, Inc.

For further information on EPRI research programs, call
EPRI Technical Information Specialists (415) 855-2411.

Bolted Joint Maintenance & Applications Guide

TR-104213

Work Order 3814-07
Final Report, December 1995

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ORDERING INFORMATION

Price: $15,000

Requests for copies of this report should be directed to the Nuclear Maintenance Applications Center (NMAC), 1300 Harris Boulevard, Charlotte, NC 28262, (800) 356-7448. There is no charge for reports requested by NMAC-member utilities.

Abstract

Proper design, assembly, preload and inspection of bolted connections remains an important activity for operators of commercial nuclear power plants. Likewise, plant leakage reduction efforts continue to receive attention at most of these generating facilities. NMAC's Bolted Joint Maintenance & Applications Guide addresses these areas of interest and represents both a major revision and a consolidation of several previous guidebooks dealing with general good bolting practices and guidelines for threaded fasteners usage. The Guide is subdivided by major application into pressure-retaining joints, mechanical joints, and structural joints. Additional information on procurement and fastener receipt inspection is also included. This document will be useful to plant engineering and maintenance personnel responsible for procedures, assembly, inspection, and troubleshooting the various types of bolted connections used in nuclear power plant applications

Acknowledgements

This guide was prepared for the Nuclear Maintenance Applications Center (NMAC) under Electric Power Research Institute Research Project RP3814-07. Many individuals and organizations provided assistance and information to make this publication possible.

NMAC would like to recognize the members of the industry Technical Advisory Group (TAG) for their contributions to the project definition and ongoing technical review:

- Mark Harris — Centerior Energy
- Horst Zitzelsberger — Consolidated Edison Company
- William Monti — Consolidated Edison Company
- Jack Wade — Entergy Operations, Inc.
- John Johnson — IES Utilities
- Fred Schillinger — New York Power Authority
- Tim Lee — Niagara Mohawk Power Cooperative
- Paul Guthrie — Tennessee Valley Authority
- Colin Pilkington — Union Electric Company

NMAC would particularly like to express its appreciation to Mr. Paul Guthrie and the Tennessee Valley Authority for their support in contributing the additional co-funding necessary to initiate this project.

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1.0 Introduction

1.1 Basis

The Nuclear Maintenance Applications Center (NMAC) originally prepared and published NP-5067-V1, "Good Bolting Practices: Large Bolt Manual" in December of 1987. Subsequently, NMAC issued NP-5067-V2, "Good Bolting Practices: Small Bolts and Threaded Fasteners", and NP-6316, "Guidelines for Threaded Fasteners". These guides have been widely distributed by NMAC and the Electric Power Research Institute (EPRI) and are frequently utilized by nuclear power plant staff in developing and improving their plant bolting programs. The development objectives for this publication were to update and consolidate this existing information into a single document, and to provide additional the technical information necessary to allow a seamless integration of the material.

1.2 Background

Over the last two decades, the NRC has issued numerous documents highlighting concerns with the integrity of bolted connections including pressure boundary, mechanical, and structural joints. The Commercial Nuclear Power industry has responded with initiatives to resolve the issues. The last NRC publication of significance on the subject was NUREG-1399 - "Resolution of Generic Safety Issue 29: Bolting Degradation or Failure in Nuclear Power Plants" referenced by Generic Letter 91-17. These documents agreed that industry and NRC initiatives for resolution of the issues were appropriate and identified some concerns that utilities should address on an individual basis as a part of their response to the Generic Letter.

Numerous EPRI and NMAC publications were recognized by NUREG-1399 as contributing to resolution of the concerns. Significantly, the NRC emphasized the usefulness of utility plant specific programs and the need to continue to address the issues identified in previous NRC guidance, i.e. Bulletins (IEBs) and Notices (IENs). This guideline provides information to help address many of the issues important to ensuring continued plant compliance with regulatory requirements.

1.3 Approach

This report consolidates key information previously published by NMAC. The primary focus is a discussion of both design and assembly of pressure boundary, mechanical, and structural joints.

The manual is divided into parts or topical sections for pressure retaining joints, mechanical joints, structural joints, and threaded fasteners. Chapters in each part cover design considerations, assembly, inspection, and trouble shooting and repair. The section on threaded fasteners addresses fastener selection, specification and procurement, and receipt inspection.

1.4 Results

The information presented in this report has been checked to the maximum extent practical. However, the user should be cautioned that the material should be used as a "guide". Data presented are meant to provide (by way of example) guidance for development of programs and procedures, but cannot and should not be construed as representing Nuclear Quality Assured information. To be used in situations requiring Nuclear Quality Assurance, data must be processed through plant-specific programs.

Much of the information presented in this report in the form of tables has obtained from various sources as noted in the tables. While significant effort has been expended to avoid typographical errors in the tables, errors cannot be ruled out and users are encouraged to verify the data for specific applications. Our purpose has been to consolidate and reproduce information as a handy reference. The authors cannot be responsible for accuracy of all information presented in the tables.

Part II of this report describes the Code rules, design requirements, assembly, and leak considerations for pressure retaining flanged gasketed joints. This Part also covers inspection criteria for leaks, receipt inspection considerations, and troubleshooting leaking connections.

Part III covers mechanical joints in detail. It includes design considerations for various types of locking devices (set screws, washers, pins, and nuts). Considerations for electrical connections are provided. This section also covers various aspects of assembly, inspection, troubleshooting, and repair.

Part IV covers design, assembly, inspection, troubleshooting, and repair of structural joints.

Finally, selection, specification, procurement, testing, and receipt inspection for several types of threaded fasteners are covered in Part V. This information will be useful to design, maintenance, and procurement engineers for a wide range of applications.

1.5 Benefits

A substantial amount of information on design and assembly is provided, including guidance and illustrations to convey the key concepts and critical success factors. A concerted effort has been made to illustrate the relationship between the design and assembly processes. Illustrations have been included for troubleshooting problem assemblies and faulty designs. General guidance on inspection and troubleshooting leaking or potentially degraded bolted connections has also been provided.

1.6 Audience

This document will be useful to engineering and maintenance personnel responsible for procedures, assembly, inspection, and troubleshooting the various types of bolted connections used in nuclear power plant applications. The intention is to present the information in a manner that will be useful to both the designer and the assembler of bolted joints. A general understanding of the design process by maintenance personnel, and the appreciation of the assembly process by design or system engineers, will assist in development of robust plant programs.

1.7 Design

The design sections of the report should be useful to a designer or maintenance engineer who might, for example, be asked to specify an assembly preload or to troubleshoot a problem joint. We have attempted to provide enough insight into the design process to aid in their tasks. We have not attempted to describe the entire design or analysis procedure. The information in this report should be used to supplement and not take the place of analysis requirements that are specified by applicable codes and standards.

1.8 Assembly

The assembly sections are intended to give the assembler information that will help to improve the reliability of the bolted connections. These sections are also intended to provide information to the designer or analyst regarding actual loads that may exist in the field. This will permit the designer or analyst to perform a more realistic analysis that should ultimately result in improved assembly specifications for the user.

1.9 Program Elements

The design and assembly of bolted joints are analogous to welding in several respects. However, bolting applications rarely receive a similar level of attention to preclude failures, e.g. leaks in pressure boundary applications, etc. NMAC recommends that utilities provide the necessary focus by creation of an on-site bolting coordinator, empowered to implement a program to eliminate failures. Program elements resulting in more reliable bolted joints that have been successfully implemented at several plants include:

- Management support - Resources are necessary to develop, implement and maintain an effective bolted joint integrity program.
- Bolting coordinator - One individual is available who has the technical ability and authority to focus on both programmatic issues and day-to-day resolution of problems.
- Procurement & inventory controls - These must be implemented to assure the appropriate quality of replacement parts and supplies are maintained, the needed quantities are available in-stock, and the proper materials, including flanges, gaskets, lubricants, and fasteners are used.
- Manuals & Procedures - These must be prepared and updated as needed, and should include the specific steps to be followed for the assembly, inspection, testing, and disassembly of joints. The root cause of failed problem joints should be understood and corrections applied.

- Training - All personnel associated with bolting activities (including supervisors, engineers, and QA/QC personnel) should receive training in the use of procedures and receive hands-on training on prototypical joints. Documentation of the training that is received should be maintained by the utility.

Application of guidance contained in this document should result in a more reliable and safe plant through an improved understanding of the design and behavior of bolted joints.

2.0 Design

2.1 Introduction

The rules for the design and assembly of bolted joints given by the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code are often difficult to interpret. Although relatively few rules apply to bolting and bolted joints, those that do apply are scattered throughout the Code.

Designers and analysts who frequently utilize the Code become familiar with its requirements, applications, and limitations. Plant engineers and maintenance supervisors, however, do not routinely use the design sections of the Code, and sometimes have difficulty locating and interpreting information it contains. In addition, designers and analysts may not always have experience with field applications.

There is an important link between establishing a leak free joint and meeting requirements of the design and technical specifications for the plant that deserves attention, particularly with regard to assembly. The purpose of this section is to outline the foundation for design-basis integrity of pressure boundary joints by presentation and discussion of pertinent Code requirements, thus, establishing the basis for the link between design, maintenance, and operation. Design information from the Code as well as from other sources is presented. No attempt is made to change or extend any portion of the Code. Explanation and discussion of Code requirements and ambiguities are also presented.

2.2 Design Requirements of ASME Section VIII

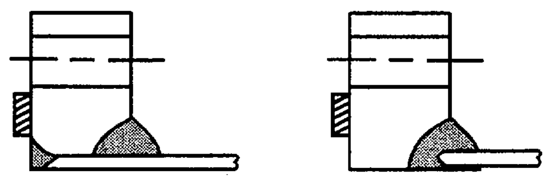
2.2.1 Section VIII Requirements

ASME Section VIII, Division 1 of the Boiler and Pressure Vessel Code provides rules for the design of bolted flange connections. The type of joints covered by the Code are illustrated in Figure 2-1 (1). Bolted flange connections covered by this Code are joints whose gaskets are entirely within the bolt circle. No flange contact occurs outside this circle.

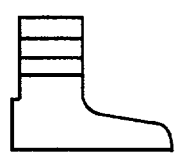
ASME Section VIII, Appendix 2 provides rules for calculating flange stresses and the amount of bolting required. This Appendix also provides information for calculating the amount of bolting required for types of unstayed flat heads and covers illustrated in Figure 2-2. Appendix 2 includes requirements for spherically dished covers with bolting flanges and for ring gaskets as well.

The flange and bolts must meet two design requirements: (1) the gasket seating assembly condition, and (2) the operating condition. The design calculations use two gasket factors:

- "y", the gasket seating stress
- "m", the gasket factor at operating conditions.

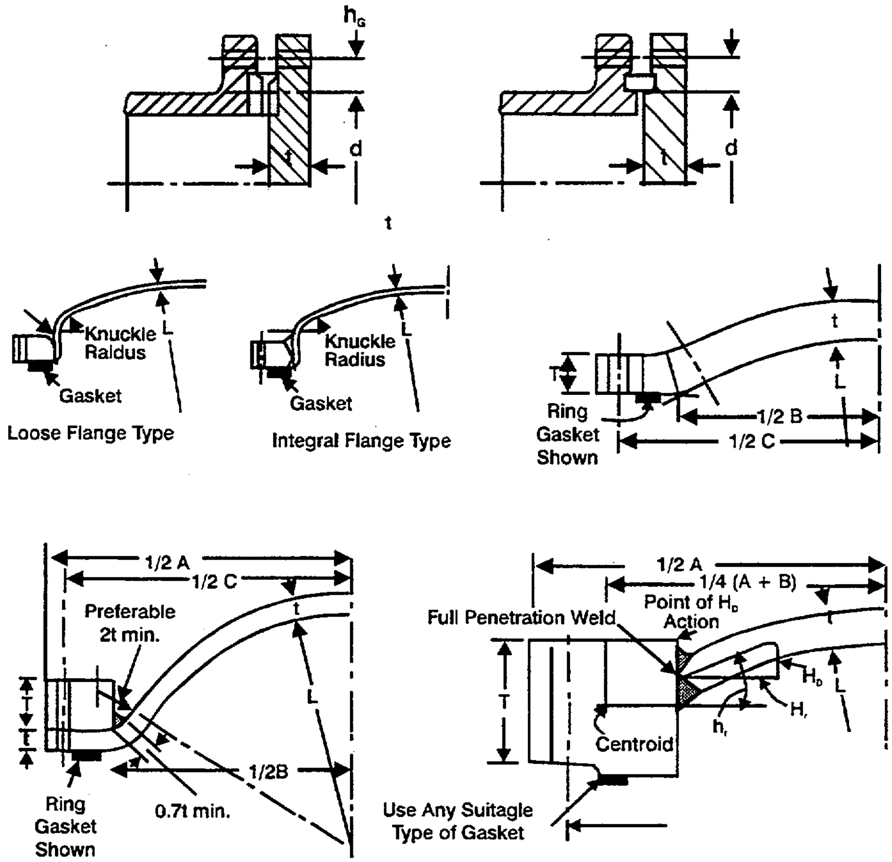


(a) Two Methods of Attaching Straight Hub Flanges



(b) Raised Face Tapered Hub, Weld Neck Flange

Figure 2-1 — Joints Covered in ASME Section VIII, Division I.



A	=	Outside Diameter of Flange
B	=	Inside Diameter of Flange
C	=	Bolt-Circle Diameter
H_D	=	Hydrostatic End Force on Area Inside of Flange
H_r	=	Radial Component of the Membrane Load in the Spherical Segment (lb.)
h_r	=	Lever Arm of Force H_r About Centroid of Flange Ring (inch)
h_G	=	Radial Distance from Gasket Load Reaction to the Bolt Circle (inch)
L	=	Inside Spherical or Crown Radius (inch)
T	=	Flange Thickness

$d = \frac{U}{V} h_o g_o^2$ for Integral Type Flanges

$d = \frac{U}{VL} h_o g_o^2$ for Loose Type Flanges

Where U = Factor Involving K

V = Factor for Inegral Type Flanges

V = Factor for Loose Type Flanges

h_o = Factor (in.) = $\sqrt{B\, g_o}$

g_o = Thickness of Hub at Small End

Figure 2-2 — Unstayed Flat Heads and Covers.

2.2.2 Concept of "y" and "m" Factors

When high gasket seating stress is achieved at assembly, better sealing performance is achieved. This is the "y" factor.

When a joint is in service, the hydrostatic end load unloads the joint, resulting in a reduction in gasket stress. Figure 2-3 illustrates this process. Under operating conditions, it is desirable to have a residual gasket stress higher than the pressure of the contained fluid. ASME Section VIII recommends that, for good sealing performance, the residual gasket stress at operating conditions be at least 2 to 3 times the contained pressure. The ratio of gasket stress to operating pressure is the "m" factor.

The relationship between initial seating stress and residual seating stress is best illustrated by a plot of gasket stress versus deflection. Figure 2-4 is a typical stress versus strain curve for a spiral wound gasket. S_{G2} is the initial seating stress and S_{G1} is the operating gasket stress. During assembly, the gasket follows the nonlinear portion of the curve from the origin to the point labeled S_{G2}. When the hydrostatic end load unloads the gasket, the gasket follows the steeper, more linear unloading curve from points S_{G2} to S_{G1}.

2.2.3 Gasket Seating

ASME Section VIII, Division1, Appendix 2 defines the minimum load required to seat a gasket (W_{m2}) as:

$$W_{m2} = \pi b G y \qquad (2\text{-}1)$$

where,
W_{m2} = Gasket seating load (pounds)
b = Effective gasket seating width (inches)
G = Mean gasket diameter (inches)
y = Gasket seating stress (psi)

The load calculated by Equation 2-1 is a function of the gasket seating stress (y) and the effective width of the gasket (b). Suggested values for "y" are given in Table 2-5.1 of ASME Section VIII, Division 2. The free body diagram of a flange under seating loads is shown in Figure 2-5.

2.2.4 Effective Gasket Seating Width

ASME Section VIII, Division 1, Table 2-5.2 provides rules for determining effective gasket width (b), which is used to calculate gasket stress. An expanded version of this is shown in Table 2-1. The concept of effective width recognizes that a uniform compressive load is not developed across the gasket width. An area smaller than the full area of the gasket supports the load. This nonuniform load is caused by flange rotation. Flange rotation increases the gasket stress on the outer edge while the inside diameter of the gasket tends to unload. Flange rotation is illustrated in Figure 2-6.

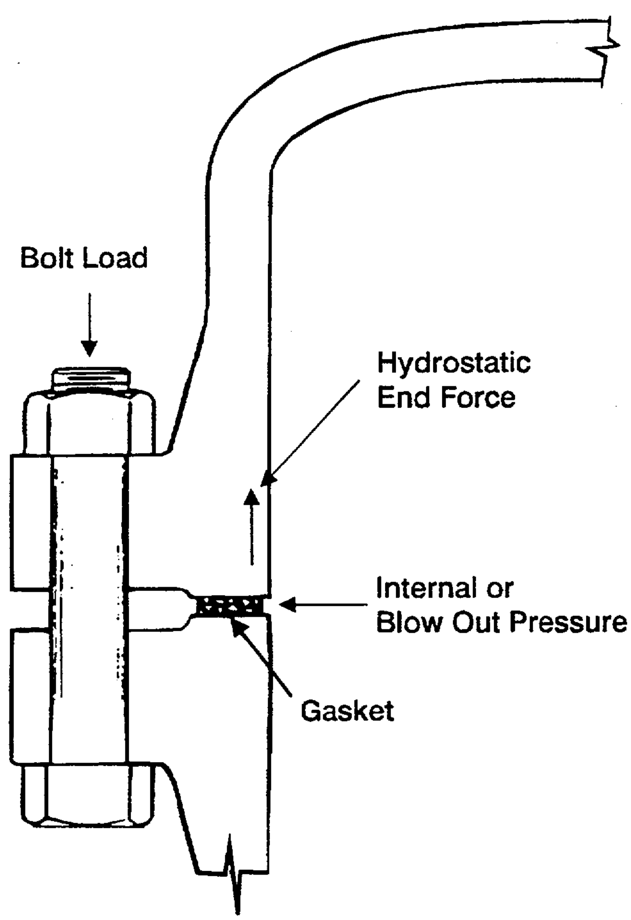
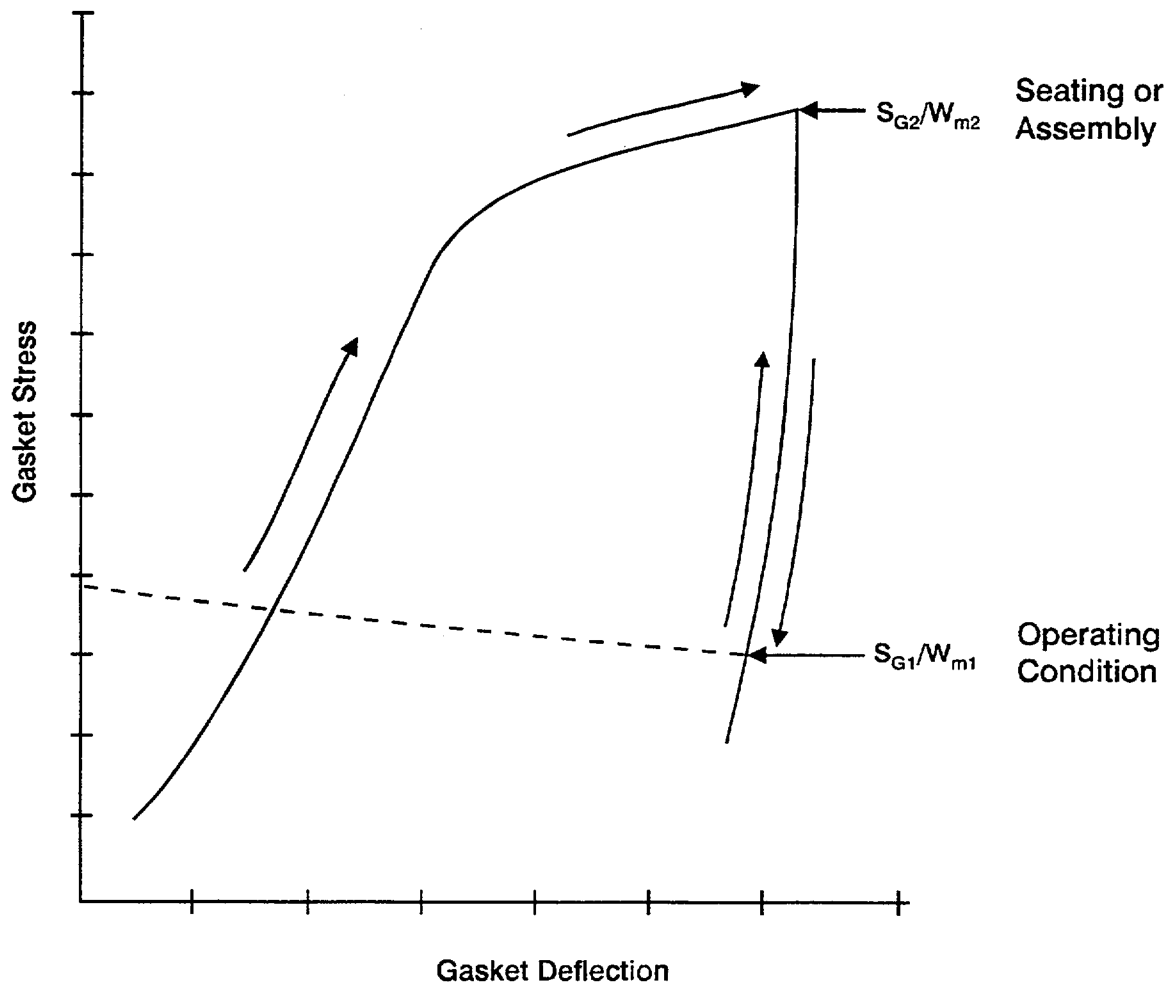
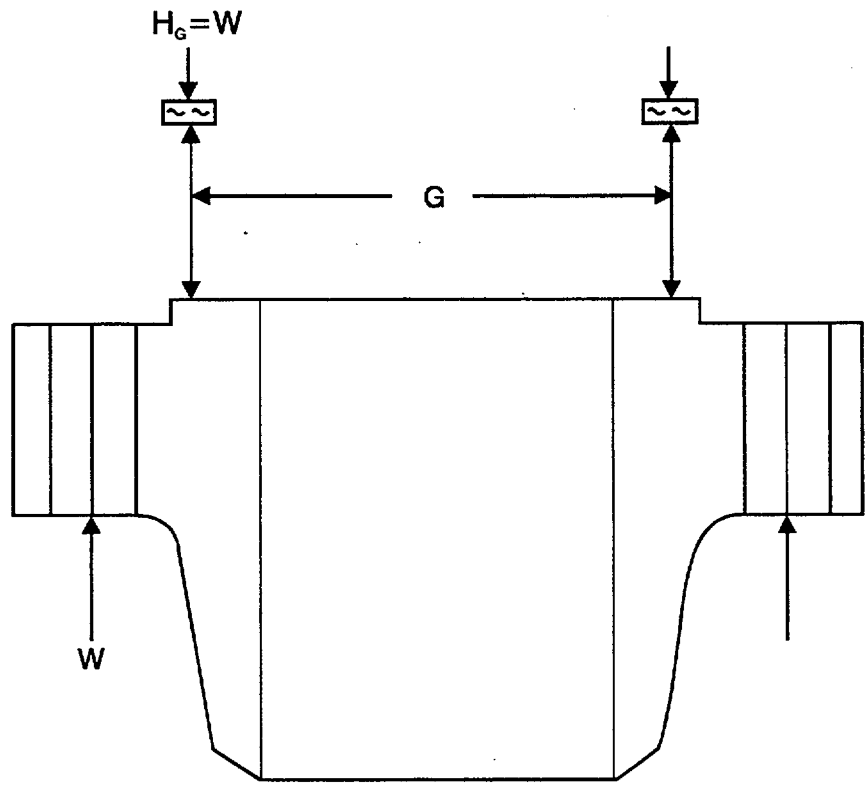


Figure 2-3 — Bolted Gasketed Joint in Service. Illustrates Reduction in Gasket Stress Due to Hydrostatic Loads.



W_{m1} = Minimum Operating Bolt Load (pounds)
W_{m2} = Gasket Seating Load (pounds)
S_{G1} = Operating Gasket Stress (psi)
S_{G2} = Initial Seating Stress (psi)

Figure 2-4 — Curve of Gasket Stress Versus Gasket Strain.



Gasket Seating Loads

G = Mean Gasket Diameter (inches)
H_G = Gasket Load (Lb.)
W = Bolt Load

Figure 2-5 — Flange Under Seating Loads.

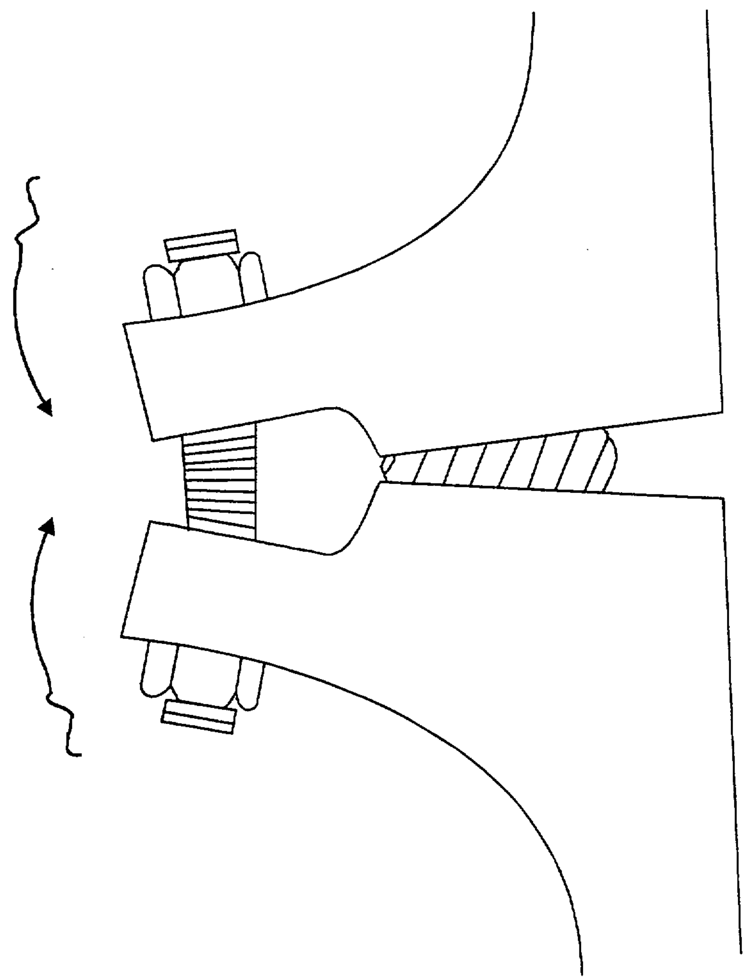


Figure 2-6 — Flange Rotation.

The effective gasket width is 50% of the actual width for gaskets less than 1/2 inch wide. For gaskets greater than 1/2 inch wide, the effective gasket width is 25% to 41% of the actual width. In theory, the use of effective gasket width is valid. In practice, however, the flange design loads calculated using the effective gasket width are lower than the loads normally applied at assembly (see Chapter 2, Section 2.5.2).

Table 2-1*

ASME Section VIII, Division I, Table 2-5.2
Comparison of Effective Gasket Width to Actual Width

ASME Equation for Effective Gasket Width			Actual Width (N)	Effective Width (b)	b/N
b = N/2	for	N < 1/2 inches	1/2	1/4	50%
$b = \sqrt{(N/2)}/2$	for	N > 1/2 inches	3/4	0.31	41%
			1 1/4	0.40	32%
$b = [\sqrt{(N/2)}]/2.8$			1 1/2	0.44	29%
			2	0.51	25%

**Developed by Mike Looram.*

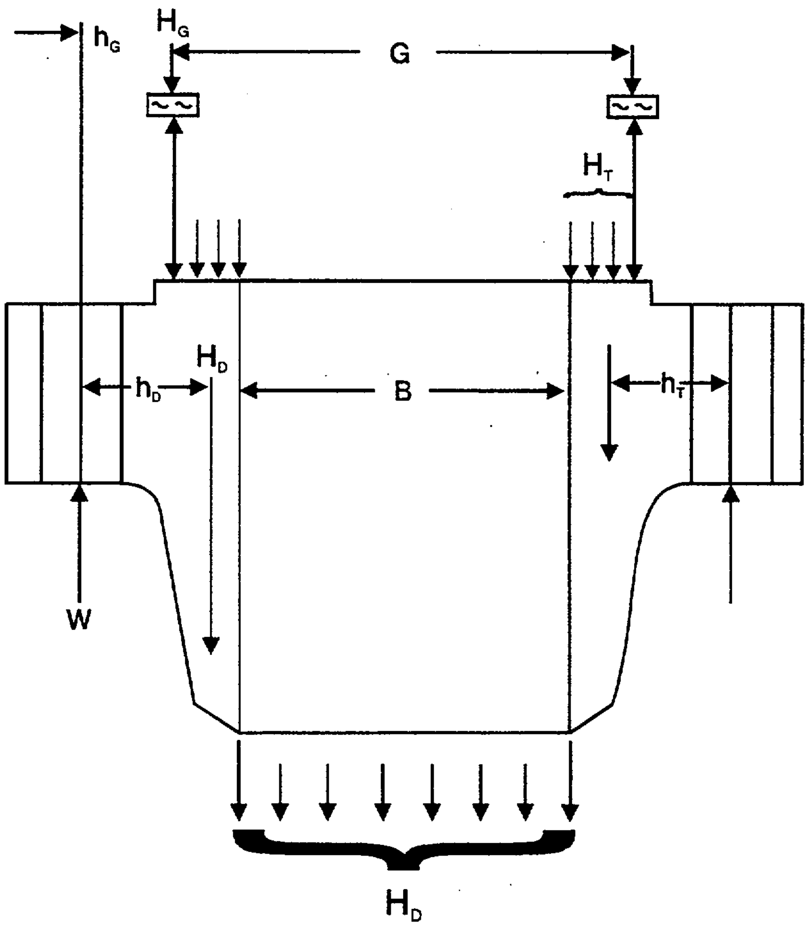
2.2.5 Operating Conditions

The minimum total bolt load at the operating condition (W_{m1}) is defined as:

$$W_{m1} = 0.785G^2P + (2b\pi GmP) \qquad (2\text{-}2)$$

where, W_{m1} = Minimum operating bolt load (pounds)
P = Design pressure (psi)
m = Gasket factor related to the minimum required load on the gasket for a tight joint at the design pressure

Equation 2-2 includes two factors which merit discussion. The factor "$0.785G^2P$" is the hydrostatic end load on the joint. The factor "2b x 3.14GmP" is the joint contact load. The equation states that the operational flange load (W_{m1}) must be at least equal to the hydrostatic end load plus a residual contact load at the sealing surface. Figure 2-7 illustrates the operating loads on the flange.



B	=	Inside Diameter
G	=	Mean Gasket Diameter (inches)
H_G	=	Gasket Load (lb.)
h_G	=	Radial Distance from Gasket Load Reaction to the Bolt Circle (inch)
h_D	=	Radial Distance from Bolt Circle, to the Circle on Which H_D Acts (inch)
H_D	=	Hydrostatic End Force on Area Inside of Flange (lb.)
H_T	=	Difference Between Total Hydrostatic End Force and the Hydrostatic End Force on Area Inside of Flange (lb.)
h_T	=	Radial Distance from the Bolt Circle to the Circle on Which H_T Acts
W	=	Flange Design Bolt Load (lb.)

Figure 2-7 — Flange Under Operating Loads.

In Equation 2-2, the hydrostatic end load reduces the gasket load by an amount equal to the end load when the design pressure is applied. A residual load of "$2b\pi GmP$" remains on the gasket. Sufficient residual gasket stress at operating conditions is required for a tight joint. **Note that ideally, the operating gasket stress is higher than the pressure of the contained fluid by the "m" factor.**

2.2.6 Required Bolt Area

The total minimum required cross sectional area of the bolts is the greater of W_{m2}/S_a and W_{m1}/S_b.

where, S_a = Allowable bolt stress at atmospheric temperature (psi)
S_b = Allowable bolt stress at design temperature (psi)

ASME Section VIII, Appendix 2 recommends that bolts and studs have a nominal diameter of not less than 1/2 inch. Bolts or studs smaller than 1/2 inch must be made of alloy steel.

2.2.7 Historical Perspective of "y" and "m" Factors

During the 1940s, Roberts (2) and others (3) suggested the concept of using "m and "y" factors to determine flange design loads. Widespread user experience supported the concept, although limited test data were available.

The ASME Boiler and Pressure Vessel Code (1) adopted a design analysis procedure employing "m" and "y" factors in determining flange design loads. Values of "m" and "y" were assigned for various gaskets with little experimental justification. These values were estimated based on the experience of ASME committee members. Table 2-5.1 of ASME Section VIII defines "m" and "y" factors and specifies in footnote that these values are suggested only and not mandatory.

Over the years, the values of "m" and "y" factors were adjusted to reflect further understanding of joint performance. At one point, tests performed by the Canadian Atomic Energy Commission indicated that the actual "y" factor for spiral wound asbestos filled gaskets was approximately four times higher than the 4500 psi suggested by ASME. ASME increased the "y" factor for spiral wound asbestos filled gaskets to 10,000 psi. Experimental work by the Pressure Vessel Research Council (PVRC) suggests that even higher values may be required to provide leak free performance. One oil company, for example, uses 1.5 times the ASME "m" and "y" factors for design.

2.2.8 PVRC Gasket Factors

Over the last fifteen years, the PVRC has conducted tests on gaskets and gasketed joints. The objective of these tests has been to reduce leaks and to improve the understanding of gasket sealing performance.

Recent PVRC test results, as discussed by Bickford (4), confirms that high gasket seating stresses are beneficial. Figure 2-8 illustrates a typical gasket stress-strain curve with lines of constant leak rate superimposed, and is used to determine the assembly gasket stress and operating gasket stress that will maintain a given leak rate. The curve suggests that

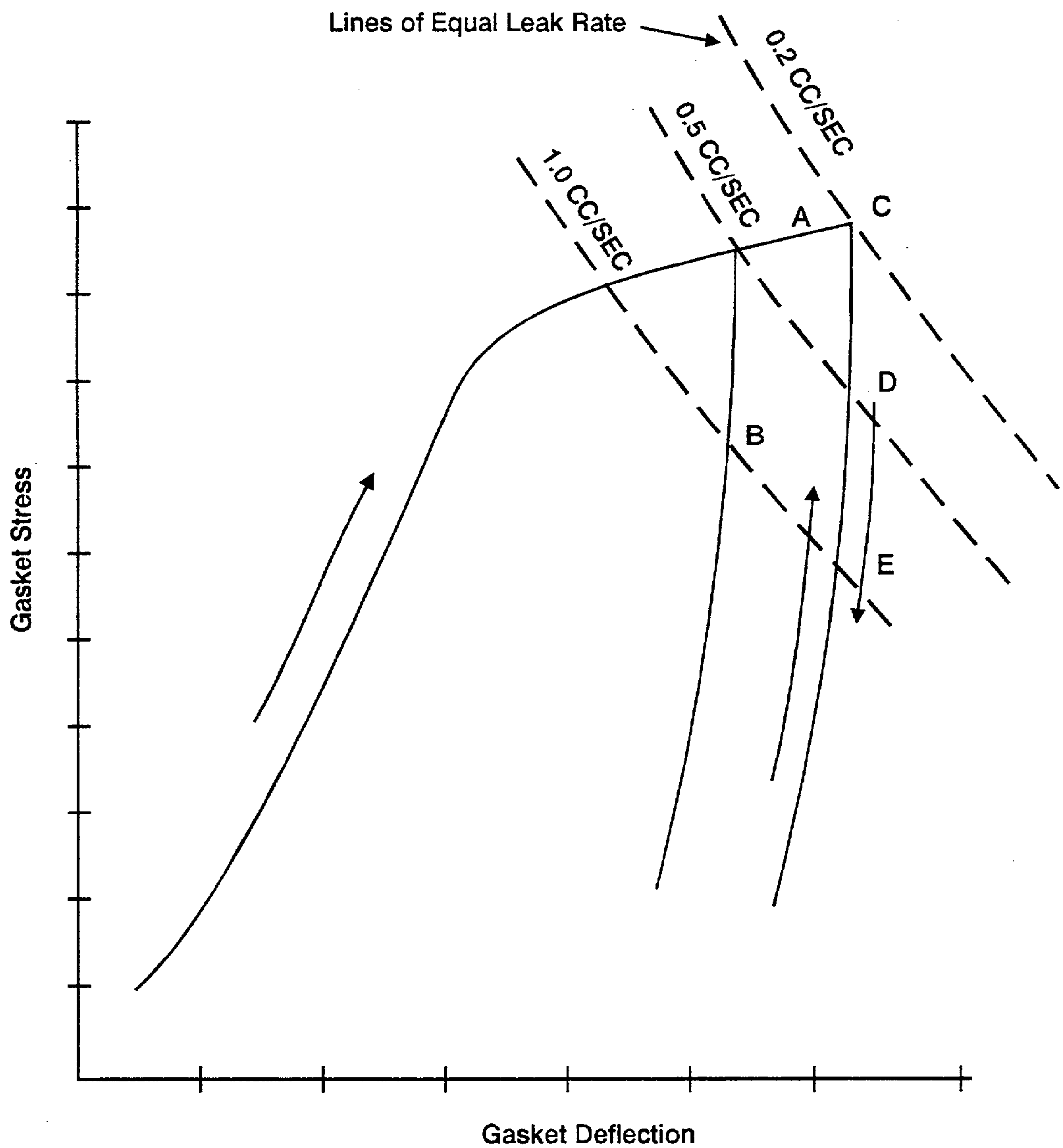


Figure 2-8 — Gasket Stress and Deflection/Leak Rate Relationship.

many combinations of seating stress and operating gasket stress provide identical leak tightnesses. The graph also suggests that for a given leak rate, the higher the initial seating stress, the lower the permissible operating gasket stress (i.e., greater load loss can be tolerated before a leak occurs).

For example, referring to Figure 2-8, to maintain a leak rate of less than 1.0 cc/sec, the operating gasket stress must be to the right of the 1.0 cc/sec leak rate line. As the gasket stress decreases during pressurization, the leak rate increases with the loss of gasket stress and deflection. The desired leak rate can be maintained in several ways. The following two scenarios give examples of how to achieve a desired leak rate of 1.0 cc/sec. Figure 2-8 illustrates the process.

- Assemble the initial gasket stress to level A. The pressure unloads the gasket to level B, thus achieving a 1 cc/sec leak rate.
- Assemble the initial gasket stress to level C. The pressure unloads the gasket to level D, resulting in a leak rate of 0.5 cc/sec. The gasket may relax to E before the leak rate exceeds 1 cc/sec.

The second scenario demonstrates that higher gasket seating stress provides a larger margin of error for sealing.

The ASME is currently assessing a new calculation system for flange design which incorporates the PVRC findings. One gasket vendor's documentation "Gasket Design Criteria" (5) provides an overview of the calculation process.

2.2.9 ANSI B16.5 Flange Connections

Similar considerations as described in foregoing paragraphs on ASME VIII bolted joints apply to ANSI B16.5 flanges. Figure 2-9 shows a typical ANSI B16.5 raised-face flange with spiral wound gasket and solid metal gage ring. The gage ring is added for the purpose of obtaining proper gasket deflection and is not intended to carry load. The user is cautioned not to overload the gage ring, this may cause rotation of the flange and leakage (see Section 2.6.2). Figure 2-9 illustrates possible detrimental effects of high preload. This is normally not a problem when proper procedures are used.

2.3 Design Requirements of ASME Section III

2.3.1 Introduction

ASME Section III, Division 1, Subsection NB (6) provides rules for the design of bolted gasketed joints for Class 1 service in nuclear power plants. This Subsection requires that bolted gasketed joints be analyzed for the following:

- Assembly loads
- Fatigue
- Bending, shear

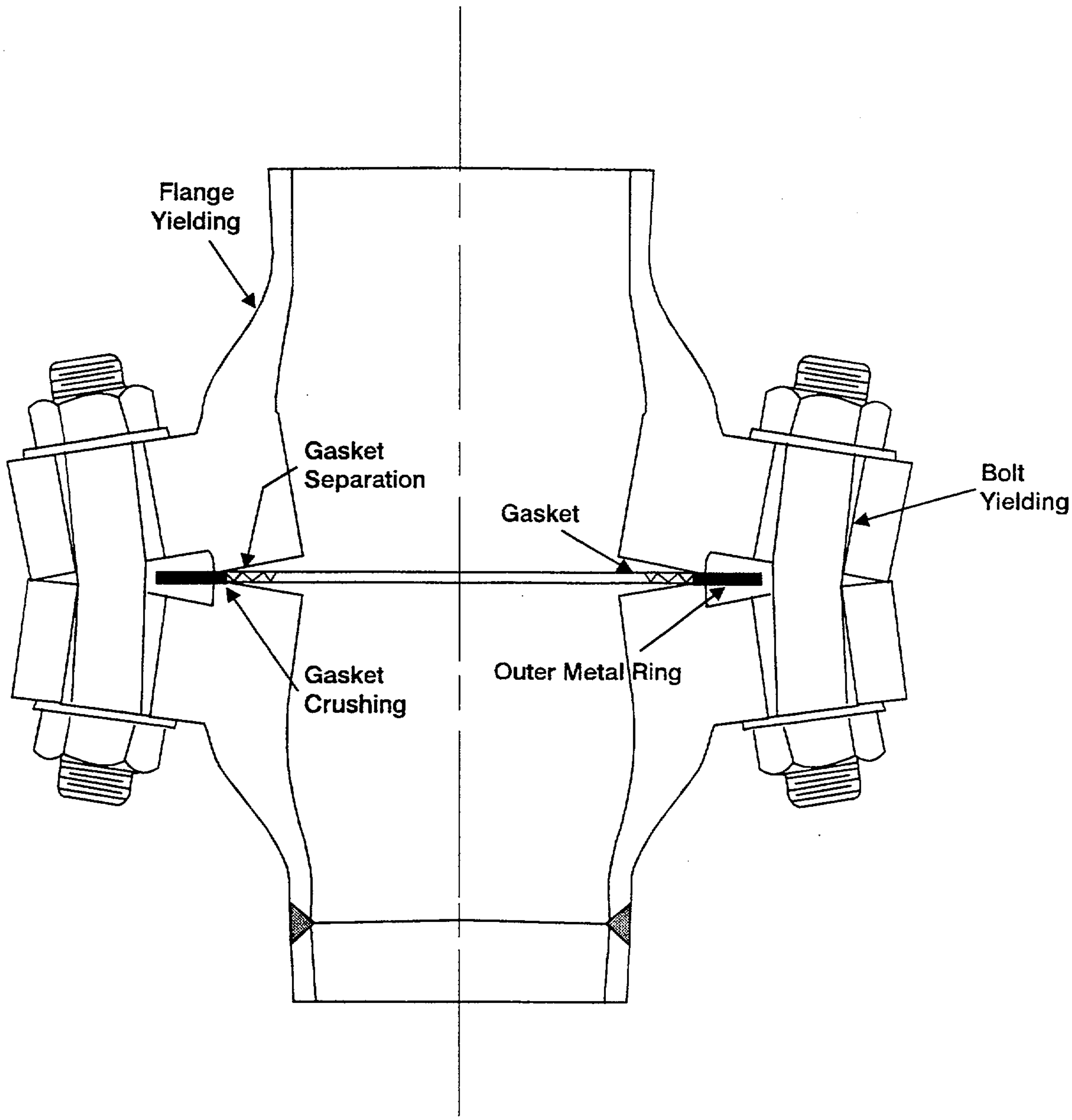


Figure 2-9 — Possible Detrimental Effects of High Preload Force.

- Stress concentrations
- Stress intensities (maximum principal stress difference)
- Operating loads, including:
  — Thermal loads
  — Pressure loads
  — External loads and moments.

2.3.2 Section III Requirements

ASME Section III, Division 1, Paragraph NB-3000 provides design rules for components. The Paragraph's "design-by-analysis" approach requires the calculation of principal stresses, the use of stress concentrations, and the consideration of operating and assembly loadings for each component. The design rules require that a component be evaluated for fatigue.

The design rules of Paragraph NB-3000 are grouped according to the type of component:

- Vessels: Paragraph NB-3300
- Pumps: Paragraph NB-3400
- Valves: Paragraph NB-3500
- Piping: Paragraph NB-3600.

2.3.3 Scope of ASME Section III

Paragraph NB-1100 provides rules for the design, materials, fabrication, examination, testing, pressure relief system, marking, stamping, and preparation of reports for Class 1 components.

Paragraph NB-1130 defines Class 1 components as those components which are contained in the reactor coolant pressure boundary, including reactor vessels, piping, pumps, and valves. Paragraph NB-1131 defines the component boundary as the first circumferential joint in a welded connection, the face of the first flange in a bolted connection, or the first thread in a threaded joint. The design specification is required to define the Class 1 components. Paragraph NCA-2000 system safety criteria as well as regulatory guides aid the facility in classifying components.

2.3.4 Design Loads

Paragraph NB-3100 provides guidance on design loads, including the following:

- Loading conditions. Paragraph NB-3111 specifies the loading conditions which must be considered in the design of components. These conditions include pressure, impact, weight of components, wind, snow, vibration, earthquake, reaction loads, temperature, and superimposed loads.

- Design loads. Paragraph NB-3112 specifies that design loads should be established in accordance with Paragraph NCA-2142. Paragraph NB-3113 requires design service limits for design loads as defined in Paragraph NCA-2142.2. A brief discussion of service limits follows below:
  - — Level A service limits are those which must be satisfied for all loading conditions to which the component may be subjected. These loading conditions are generally the <u>normal</u> operating conditions for the component and are the focus of Paragraph NCA-2142.2.
  - — Level B service limits are those which the component must withstand without damage that requires repair. These limits are sometimes referred to as <u>upset</u> conditions. The maximum duration of conditions that meet Level B service limits must be included in the design specification. The following conditions can cause Level B limits:
    - — Operation Based Earthquake (OBE): OBE is a seismic event usually on the order of one half the magnitude of safe shutdown earthquake (SSE). The component is designed to operate through this event.
    - — Transient conditions: Examples of transient conditions are power level changes on the order of 10% to 20%.
  - — Level C service limits are <u>emergency</u> conditions. The component is designed to withstand these conditions, but large deformations in areas of discontinuity are created. Removal and/or repair of the component is permissible. Level C limits must not cause more than 25 stress cycles having a stress amplitude of greater than that for 10^{10} cycles on the appropriate fatigue diagram.
  - — Level D service limits are <u>faulted</u> conditions for which gross general deformation with a loss of dimensional stability is permitted. The component may require repair or removal after the event. Examples of events which can impose Level D limits are SSE, pipe break, or a combination of the two events.

2.3.5 Design by Analysis

The design by analysis philosophy of ASME Section III (<u>6</u>) is based on the following factors:

- The component may be subjected to cyclic loads.
- Superior reliability of the component is required due to the nature of the contained fluids.
- Periodic inspection of the component may be difficult or impossible.

Paragraph NB-3214 requires that a stress analysis of all major structural components be prepared. This stress analysis should be in sufficient detail to show that the stress limits given in Paragraphs NB-3220 and NB-3230 are satisfied when the component is subject to the design loadings given in Paragraph NB-3110.

The analysis required for a Class 1 bolted joint may be called interaction analysis (7). An interaction analysis considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. Margins of safety are assessed by comparing calculated stress to a predicted failure using the maximum shear theory of failure.

2.3.6 Class 1 Analysis Flow Diagram

The flow diagram illustrated in Figure 2-10 gives an overview of the analysis of a Class 1 joint.

The first task, illustrated in Block 1, is to determine whether the component is subject to fatigue loading. Langer (8) made the following observation concerning fatigue analysis of a pressure vessel:

> *The fatigue analysis of a vessel is apt to be the most laborious and time consuming part of the design procedure, and this effort is not warranted for vessels which are not subjected to cyclic operation. However, no obvious distinction between cyclic and non-cyclic operations exist. No operation is completely non-cyclic, since start up and shut down are cycles themselves. Therefore, fatigue can never be completely ignored.*

2.3.6.1 Block 1: Is Fatigue Analysis Required?

Paragraph NB-3222.4(d) prescribes a set of rules and limits which are used to determine the necessity of a fatigue analysis. The determination is based on a comparison of the average service stress due to start up and shut down, service pressure fluctuations, thermal gradients, mechanical loads, and thermal stresses to stress limits defined in this section.

The design load conditions and the limits of Subsection NB-3222.4(d) are as follows:

- Start up and shut down. The number of cycles from zero pressure (i.e., at start up) to design pressure must be less than the cycles allowed for a stress amplitude (S_a) which is three times the allowable stress (S_m). For example, an SA193 B7 material at 700°F has an S_m of 26.8 ksi. Multiplying S_m by 3 gives S_a = 80.4 ksi. Figure 19.4 of ASME Section III, Division I, Subsection NB shows that the allowable number of cycles is 700.

- Normal service pressure fluctuations. The full range of pressure fluctuations should not exceed the quantity (1/3 x design pressure x (S_a/S_m)).

  According to Rodabaugh and Moore (9), the assembly and disassembly cycles have the largest stress amplitudes and are not considered load cases for fatigue evaluation.

- Thermal gradients. Thermal stresses due to cyclic operation are evaluated under three conditions: (1) start up and shut down; (2) normal service; and (3) when

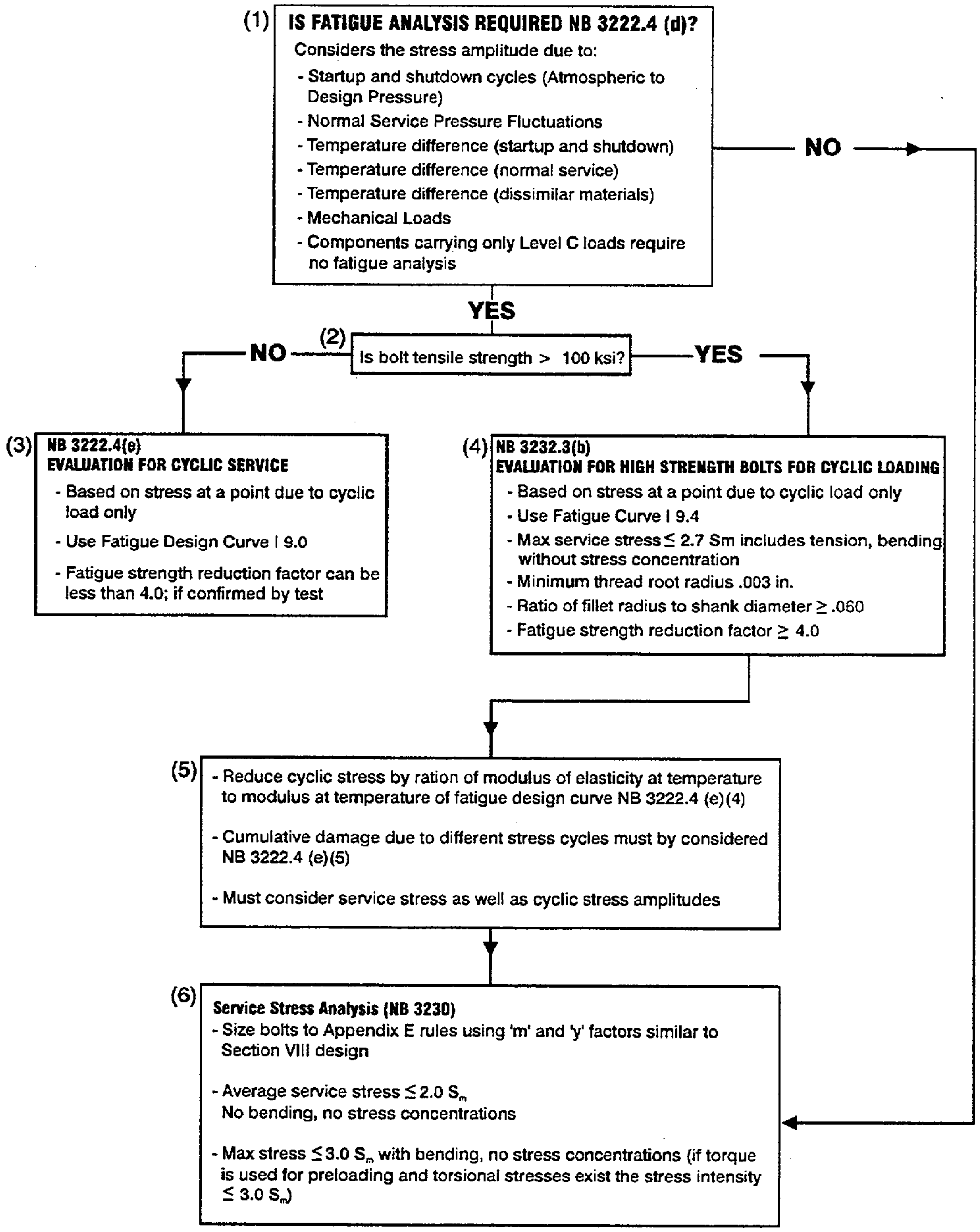


Figure 2-10 — Class 1 Analysis Flow Diagram of a Class 1 Joint.

dissimilar materials are used. The following equations demonstrate the calculation of thermal gradients for each condition:

— Start up and shut down

$$\Delta t \leq S_a \,/\, 2Ea \qquad (2\text{-}3)$$

— Normal service

$$\Delta t \leq S_a \,/\, 2Ea \qquad (2\text{-}4)$$

— Use of dissimilar materials

$$\Delta t \leq S_a \,/\, [2(E_1a_1 - E_2a_2)] \quad (2\text{-}5)$$

where, Δt = Temperature difference between adjacent points (°F)
S_a = Stress amplitude (psi)
E = Modulus of elasticity (psi)
a = Coefficient of thermal expansion (inches/(inches/°F))
Subscripts 1 and 2 refer to Materials 1 and 2, respectively.

The adjacent points referred to in defining Δt are along a surface or across a wall section. For bolting, the temperature difference which has significance is the difference in temperature between the clamped parts and the fastener.

- Mechanical loads. The effect of mechanical loads, excluding pressure but including pipe reactions, is considered. The range of stress caused by the mechanical loads is not to exceed the stress amplitude from the fatigue curve in Figure I 9.4 of ASME Section III, Division 1, Subsection NB.

If the component is not subject to fatigue loading, the designer may go directly from Block 1 to Block 6 to evaluate the service stress.

2.3.6.2 Blocks 3, 4, and 5: Fatigue Analysis

If the results of the considerations in Block 1 indicate that a fatigue analysis is required, continue to Blocks 3, 4, and 5 in the flow diagram. These blocks provide important points for evaluation of Class 1 joints. The following guidelines are given in Blocks 3 and 4:

- Fatigue life is based on cyclic stress amplitude at a point.
- Fatigue curves of stress amplitude versus the number of cycles to failure (S-N curves) are provided.
- The thread root radius must be a minimum of 0.003 inches. The ratio of bolt fillet radius to shank diameter must be at least 0.060.
- The fatigue strength reduction factor (i.e., stress concentration) must be at least 4.0.

- The cyclic stress amplitude is reduced if the modulus of elasticity decreases at design temperature. Service stress as well as cyclic stress must be considered.

After fatigue analysis is completed, the designer compares the service stress on the fastener to the criteria given in Block 6.

2.3.6.3 Block 6: Service Stress

Paragraph NB-3230 gives design conditions for Class 1 joints:

- The number and cross-sectional geometry of bolts required to resist the design pressure is prescribed in Appendix E of ASME Section III. The largest bolt load is designated the design mechanical load.

- The Appendix E approach is identical to that in Section VIII, which considers the seating condition (using a "y" factor) and the operating condition (using an "m" factor). These conditions are discussed in Section 2.2.2.

Paragraph NB-3232 gives service limits for Class 1 joints. The service stress in bolts, produced by a combination of preload, pressure, and differential thermal expansion, may be higher than the values given in Table I 1.3 of the ASME Code.

- The maximum value of service stress averaged across the bolt section, not including bending and stress concentrations, must be less than or equal to twice the allowable stress (i.e., 2 x S_m).

- The maximum service stress at the periphery of the bolt, including bending but excluding stress concentrations, must be less than or equal to three times the allowable stress (3 x S_m).

- If torque is used in preloading, the stress intensity must be less than or equal to (3 x S_m).

2.3.7 Stress Analysis Terms

Terms relating to stress analysis are given in Paragraphs NB-3213.1 through NB-3213.34. A brief description of important terms follows below:

- Stress intensity is the difference between the algebraically largest and smallest principal stresses at a point.

- Local structural discontinuity is a material or geometric discontinuity which affects the stress distribution (e.g., a fillet or a thread root).

- Normal stress is the component of stress normal to the plane of reference.

- Average stress is the uniform distribution of normal stress across a section.

- Shear stress is the component of stress tangent to the plane of reference.
- Bending stress is the variable component of normal stress across a section.
- Primary stress is a stress developed by an imposed load. A primary stress is not self limiting.
- Secondary stress is a normal or shear stress developed as the result of a constraint of adjacent material or self constraint of the structure (e.g., general thermal stress).
- Thermal stress is a self balancing stress produced by a non-uniform distribution of temperature or differing coefficients of thermal expansion.
- Peak stress is an increment of stress which is additive to primary and secondary stress. Peak stress is caused by either local discontinuities or local thermal stress, including stress concentrations.
- Total stress is the sum of primary, secondary, and peak stresses.
- Operational cycle is the initiation and establishment of new conditions followed by a return to the conditions that prevailed at the beginning of the cycle.

2.3.8 Maximum Shear Stress

The maximum shear stress at a point is defined as one half of the algebraic difference between the largest and the smallest of the three principal stresses. If the principal stresses from largest to smallest are s_1, s_2, and s_3, then the maximum shear stress, t_{xy}, is defined as:

$$\tau_{xy} = (1\ /\ 2)\,(s_1 - s_3) \qquad (2\text{-}6)$$

The maximum shear stress theory of failure states that yielding in a component occurs when the maximum shear stress reaches a value equal to the maximum shear stress at the yield point in a tensile test. At the yield point in a tensile test, $s_1 = S_y$ and $s_3 = 0$. Substituting into Equation 2-6, the maximum shear stress equals $S_y/2$. Therefore, yielding in the component occurs when:

$$(1\ /\ 2)\,(s_1 - s_3) = 1\ /\ 2\ S_y \qquad (2\text{-}7)$$

Multiplying both sides of Equation 2-7 by two simplifies the equation and results in a new term, "equivalent intensity of combined stress" or "stress intensity". The stress intensity is twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal stresses. Thus, the stress intensity is directly comparable to strength values determined in tensile tests (10).

2.3.9 Calculation of Stress Intensity

The stress intensity at a point of interest on a structure is determined using the following procedure:

1. Set up an orthogonal coordinate system. For bolting, the directions of interest will be along the axis of the bolt and the tangential.

2. Calculate the stress components (both normal and shear) in each direction for each type of loading. The stresses for bolting will usually be axial tension, bending, and shear stress.

3. Assign each set of stress values to one or more stress types given in Table 2-2.

Table 2-2

Stress Types for Calculating Stress Intensity

Stress Type	Symbol	Reference Paragraph
Primary membrane stress	P_m	NB-3213.8
Primary bending stress	P_b	NB-3213.7
Secondary stress	Q	NB-3213.9
Peak stress	F	NB-3213.11
Expansion stress	P_e	NB-3213.20

The stresses in the bolting of a raised face gasketed joint, for example, would be categorized as follows:
— Axial tension (P_m) due to assembly preload and hydrostatic end load
— Bending (P_b) due to flange rotation
— Thermal stress (Q) due to thermal gradient and dissimilar materials
— Peak stress (F), which considers stress concentration due to thread roots, thread runout, and shank-to-bolt head fillet (11).

4. Translate the stress components for the tangential (t), longitudinal (l), and radial (r) directions into principal stresses, s_1, s_2, and s_3. In many pressure component calculations, the t, l, and r directions may be chosen such that the shear stress components are zero and s_1, s_2, and s_3 are identical to s_t, s_l, and s_r.

5. Calculate the stress differences, S_{12}, S_{23}, and S_{31}, from the relations:

$$S_{12} = \sigma_1 - \sigma_2 \qquad (2\text{-}8)$$

$$S_{23} = \sigma_2 - \sigma_3 \qquad (2\text{-}9)$$

$$S_{31} = \sigma_3 - \sigma_1 \qquad (2\text{-}10)$$

The stress intensity, S, is the largest absolute value of S_{12}, S_{23}, and S_{31}.

An example of a stress intensity calculation by Rodabaugh and Moore (9) for the fasteners of an 8 inch 900# gasketed joint is given below.

Problem statement: Two 8-inch 900# ANSI B16.5 flanges are to be assembled using a fastener assembly stress of 40,000 psi. Calculate the fastener stress intensity at operating conditions.

Operating conditions:

- Internal pressure = 1410 psi (this is a proof test pressure)
- Equivalent pressure due to pipe moment = 2700 psi
- Pipe temperature = 700°F
- Ring temperature = 650°F
- Bolt temperature = 600°F
- Modulus of elasticity ratio, assembly to operating temperature = 0.90
- Allowable fastener stress (S_m) = 28,400 psi.

Analysis: An interaction analysis, using the FLANGE program gives the following bolt operating stresses:

- Average bolt stress (σ_{xx}) = 34,780 psi
- Bending stress (σ_b) = 9,520 psi
- Shear Stress (τ_{xy}) = 14,400 psi.

Section 2.4.3 contains more information on FLANGE.

Calculation of stress intensity: The stress intensity is two times the maximum shear stress of the fastener.

- $SI = 2 \times \textit{maximum shear stress}$ (2-11)
- $SI = [(\sigma_{xx} + \sigma_b)^2 + 4(\tau_{xy})^2]^{1/2}$ (2-12)
- $SI = [(34{,}780 + 9520)^2 + 4(14{,}400)^2]^{1/2} = 52{,}800$ psi.

The allowable stress intensity for this fastener is $3S_m$ = 85,200 psi.

2.4 Analysis of Flanged Gasketed Joints

2.4.1 General Model

Consideration of a model of a flanged joint will provide a better understanding of how the joint works and why it leaks, refer to Figure 2-11. The model includes information about the loads, stresses, and distortion of the joint under operating conditions. This model is suitable for analysis required by ASME Section III, Division I, Subsection NB. Features of the model given in Figure 2-11 include:

- Complete joint assembly including fastener, gasket, and flanges, is modeled.
- The stiffness and thermal expansion properties of the components, including the coefficient of expansion and changes in the modulus of elasticity with temperature, are considered.
- An interaction analysis may be performed using the model as a basis by the following procedure:
  — Introduce assembly loads into the system.
  — Calculate the stress, strain, and deflection of the components.
  — Apply operating loads to the model. The assembly loads are altered by the operating conditions and the system response to these loads.
  — Calculate the operating stresses.

2.4.2 Analysis of the Model

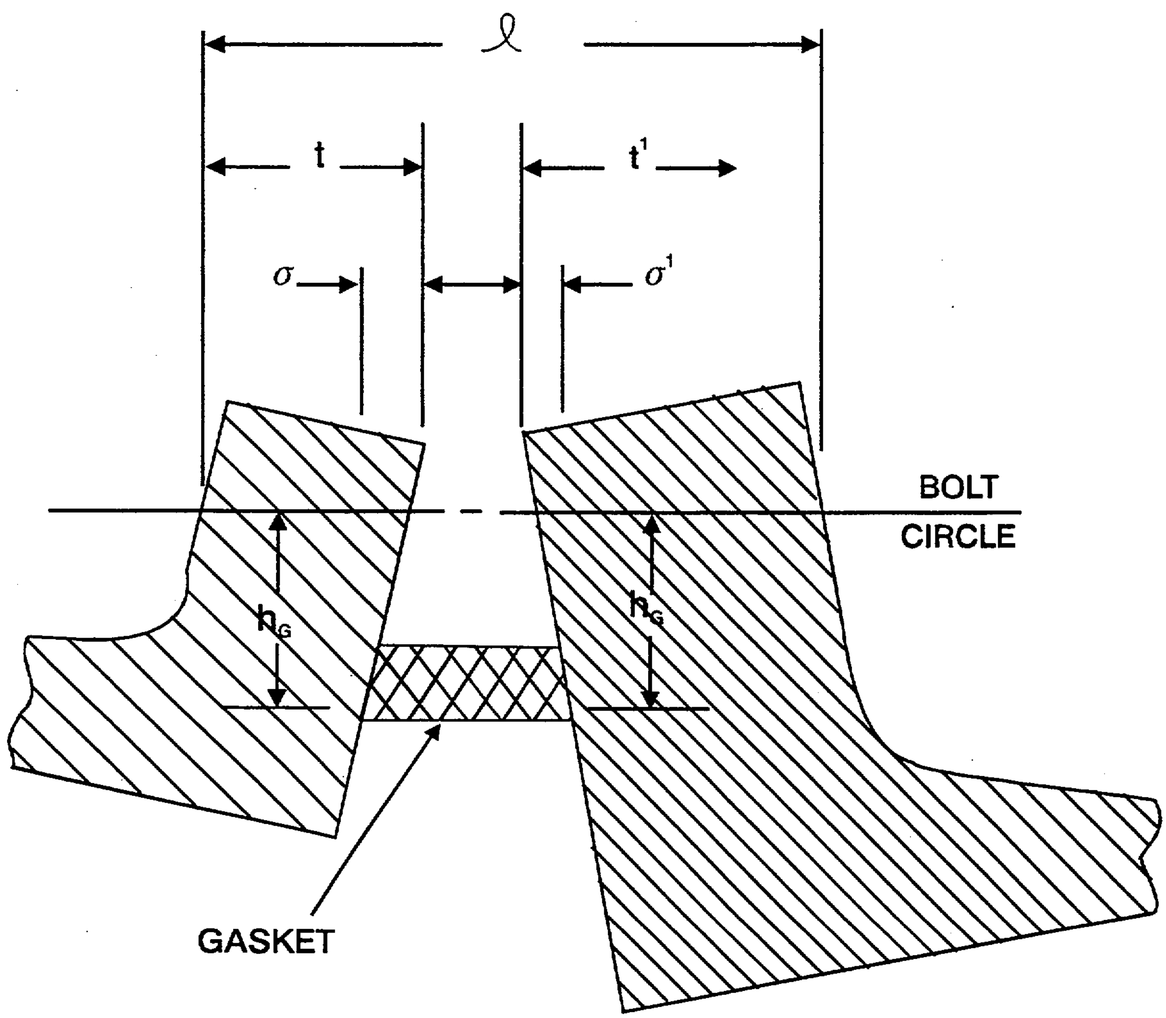
When an assembly bolt load is applied across a joint, the gasket compresses and the flanges rotate. The gasket compressive load is equal to the fastener preload. The flange rotation is a function of the stiffness of the flange (principally the ring thickness), the magnitude of the bolt load, and the resultant moment produced by the bolt load.

Calculating the response of a bolted joint system to applied loads is difficult. Fortunately, some researchers have published papers on this subject.

In 1951, Wesstrom and Bergh (12) developed the idea that flange rotation, caused by internal pressure, resulted in a loss of fastener load. In their paper, they developed equations to calculate flange rotation and the effect on bolt load. This concept is summarized by the following equation:

$$W_2 = W_1 + \alpha p \tag{2-13}$$

where,
W_2 = Operating bolt load (pounds)
W_1 = Assembly bolt load (pounds)
α = Flange factor (inches^2)
p = Internal pressure (psi)



ℓ = Bolt Length (in.)
σ = Relative Axial Displacement Between Gasket Center and Bolt Circle
h_G = Radial Distance From Gasket Load Reaction to the Bolt Circle (in.)
t = Ring Thickness

Figure 2-11 — General Model of a Bolted Joint.

For large flanges, which are generally quite flexible, a can be on the order of -203. A flange factor of this magnitude can equate to a 20% to 30% loss of preload when the system is pressurized.

Other investigators have confirmed Wesstrom and Bergh's findings. Rodabaugh and Moore (7) calculated the loss of load at operating conditions and verified this loss and corresponding value for a by tests on a heat exchanger joint. Andreosso (14) demonstrated a 20% loss in bolt load due to operating conditions for a pressurized water reactor (PWR) pressurizer manway joint. His calculation method employed a stiffness matrix approach, and the results were verified by tests. Andreosso concludes:

> *It is an important precaution to verify the effect of cyclic loadings on the tightness of a closure. In the special case of the PWR plant pressurizer manway, we have shown that the parts deformations during a pressure build up lead to a loss of tightening which has to be compensated for by an initial overtightening.*

2.4.3 Computer Programs For Flange Analysis

2.4.3.1 FLANGE Program

In 1976, Rodabaugh and Moore (7) published a paper discussing a computer program that performs flange analysis. This program, called FLANGE, uses and expands upon the calculation methods and equations developed by Wesstrom and Bergh (14).

The FLANGE program analyzes the following applications:

- Section VIII Division 1 and 2
- Section III, Subsections NB and NC
- ANSI B31.1.

Transient as well as steady state thermal loads can be modeled by FLANGE. This modeling capability allows the analyst to calculate the bolt load and gasket stress during transient conditions, which are most challenging to sealing.

Engineers can use the program to answer the important question — *What is the preload?*. The FLANGE program can be used to calculate the flange, bolt, and gasket stresses and deflections at assembly. The analyst can impose loads which simulate a transient start up condition, including a temperature gradient across the flange. Steady state operation, operating transients, and shut down can also be simulated.

The FLANGE program may be used in the following applications:

- Straight, taper-hub, or blind raised-face flanges
- Combinations of dissimilar flanges (e.g., straight-to-taper, straight, or taper-to-blind)

- Ring type joints whose gasket or sealing surface is within the bolt circle.

The following is a list of calculations the FLANGE program can perform:

- Flange stress due to moment loads on the ring (ASME Section VIII analysis types)
- Flange rotation
- Flange stress and bolt load resulting from the following loading conditions and their combinations:
  — Initial assembly
  — Internal pressure
  — Temperature gradient from the hub to the ring and from the ring to the bolt
  — Axial thermal expansion of the components
  — Temperature effect on the modulus of elasticity
  — Non-axisymmetric pipe loading of the flange.

The following is an example of a problem statement which may be analyzed by the FLANGE program:

Problem Statement: Analyze a joint composed of a pair of identical 8"-900# ANSI B16.5 flanges, fasteners, and gaskets for rated temperature and pressure and complete the following:

1. Evaluate the suitability of the system for ASME Section VIII, Division 1; ASME Section VIII, Division 2; and ASME Section III, Division 1, Subsection NB.
2. Determine the assembly preload and justify the adequacy of the preload to maintain a leak free joint (Sections 2.5.3.1 and 2.5.3.2 discuss this subject in detail).
3. Show that the preload is in compliance with the design Codes. Consider the following:
   — Design by analysis
   — Operating and assembly loads
   — Principal stress and stress intensity
   — Bending
   — Thermal effects
   — External loads
   — Fatigue due to load fluctuations
   — Torsional stress if torqued.

The FLANGE program is available in open literature. Report No. ORNL 5035, "A Computer Program for the Analysis of Flanges Joints with Ring Type Gaskets", (17) provides the FLANGE program. This report is available from:

National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

2.4.3.2 Finite Element Analysis

For some joints, finite element analysis by an experienced designer may prove to be a useful tool for evaluation of flange design and assembly parameters. In particular, finite element analysis has been used to solve tough problems associated with problem leaking joints, and to demonstrate margins of safety when degradation is suspected. A state-of-the-art general purpose program should be adequate for bolted joint interaction analysis.

2.5 Assembly Design Considerations

2.5.1 Introduction

A designer or plant engineer supplies the assembly preload for a gasketed joint. In this section, the practical and Code considerations which affect selection of an appropriate preload are discussed.

2.5.2 Bolt Preload

Appendix 2 of ASME Section VIII applies only to the design of bolted flange connections with gaskets entirely within the bolt circle. Applying the design rules leads to the minimum amount of bolting area required. The Appendix provides no guidelines for preload nor does it limit the bolt assembly (preload) stress. The bolt assembly stress often must exceed the allowable stress if the joint is to be tight at the design pressure or is able to pass a hydrostatic test.

The last statement in the above paragraph is supported by the following case: assume that (1) the amount of bolting required by Section VIII design is governed by operating conditions (see Equation 2-2), and (2) the total cross sectional area of the bolts provided just equals the total required area. Then:

$$W_{m1} = A_m S_b \tag{2-14}$$

where, W_{m1} = Gasket seating load (pounds)
A_m = Required bolting area (inches^2)
S_b = Allowable bolt stress at operating temperature (psi)

Bolts stressed to their allowable stress develop a total load equal to W_{m1}. Accordingly, the residual load on the gasket at the design pressure, given by Equation 2-2, is equal to only $2b\pi GmP$. However, the actual bolt stress at the design pressure is usually less than W_{m1}. Thus, the residual load on the gasket is less than $2b\pi GmP$. This is due to the pressure causing additional rotation of the flanges. Accordingly, the assembly will leak at the design pressure. For the case just cited, the assembly bolt stress must exceed 1 1/2 times W_{m1} or 1 1/2 times the allowable bolt stress to pass a hydrostatic test of 1 1/2 times the design pressure.

Appendix S of ASME Section VIII, Division 1 of the Boiler and Pressure Vessel Code, "Design Considerations for Bolted Flange Connections", (1) recognizes that design loads may not seal and states:

> *...In any event, it is evident that an initial bolt stress higher than the design value may, and in some cases must, be developed in the tightening operation, and it is the intent of this Division that such practice is permissible.*

Appendix S recommends that the assembler tighten sufficiently to withstand the test pressure. It has been suggested that 1 1/2 times the Code flange design load may be required to pass a hydrostatic test. The test is not likely to be as challenging to the joint as the service conditions. This implies that more load may be required to meet long term service conditions.

Appendix S also cautions the designer that a distinction exists between assembly design values and the load which is required for joint integrity at service conditions. The assembly of the flanged joint is likely to be less than perfect, and the design values of the flanged bolt load may never be achieved. For example, using torque assembly and three cross bolting passes, the residual preload may have a 2 to 1 scatter with an average load of 50% to 60% of the design load (see Chapter 3, Section 3.7.1 for additional discussion on this issue).

As noted above, bolt load often decreases during operation due to flange rotation, relaxation of the gasket flange, and bolt material. This loss of clamping load increases the potential for leaks.

Appendix S approves the use of simple wrenching without verifying the actual bolt stress. This Appendix states that, using simple wrenching, the probable bolt stress is given by

$$S = 45{,}000 \,/\, \sqrt{d} \qquad (2\text{-}15)$$

where, S = Bolt stress (psi)
d = Nominal diameter of the bolt (inches)

Appendix Y of ASME Section VIII, which applies to flat face flanges with metal-to-metal contact outside the bolt circle, adopts the same philosophy as Appendix S: simple wrenching without verification.

Article 3-5 of ASME Section VIII, Division 2 offers guidance on assembly loads similar to that in Section VIII, Division 1, Appendix S. The article suggests that high bolt assembly loads may cause local yielding at this point states that locally over stressing the flange due to bending at the small end of the hub is acceptable. The flange will redistribute the loads as long as it has sufficient capacity in the ring and the tangential and radial ring stresses are reasonably low. This subject is discussed in more detail in Section 2.5.3.2.

The Electric Power Research Institute (EPRI) video, "Leaks and the Engineer" (15), includes a panel discussion on allowable bolt stress and assembly stress. In this discussion, R. W. Schneider suggests that a designer should not specify a torque or preload unless an interaction analysis has been performed to determine the required load to keep the joint tight. In essence, Appendix S and Article 3.5 of ASME Section VIII, Division 2 (16) allow tightening to levels that are deemed sufficient for service. The upper limit for tightening is that limit which does not excessively distort the flange or grossly crush the gasket.

2.5.3 What is the Preload?

Section 2.5.2 establishes that the flange design load (W_{m1} or W_{m2}) is not necessarily the assembly load. Based on experience, the target (nominal) preload should usually be higher than the design load. What, then, is the appropriate preload?

2.5.3.1 Practical Considerations for Assembly Stresses

Practical considerations when recommending bolt preload and torques include the following:

- For carbon steel flanges having allowable stress of approximately 20,000 psi with low alloy, quenched, and tempered bolting, an appropriate preload is between 45,000 psi and 67,000 psi (9).
- Field experience suggests that bolt stress of 45,000 psi is effective for sealing joints.
- Gasket seating stress on spiral wound gaskets should be at least 10,000 psi and should not exceed 25,000 psi, based on the full width of the gasket. Note that the ASME Section VIII calculation uses an effective gasket width which is approximately half the actual gasket width (Section 2.2.4).
- The design assembly stress may not be achieved in the field due to uncertainties in torquing. Approximately 50% of the design load across the joint may be attained, with a scatter of ±40% due to joint interactions. Section 3.7.1 provides additional information about this subject.
- The assembly stresses will change based on operating conditions (i.e., temperature and pressure). The stresses tend to decrease due to:
  — Flange rotation
  — Stress relaxation
  — Gasket creep
  — Redistribution of loads due to localized yielding of flange gasket, and fastener material.

- Recommended torques are calculated using the short form torque/preload relationship, discussed in Section 3.2.1. Section 3.2 provides details for the torque calculation procedure, including:
  - — The short form torque/preload equation
  - — Tables of nut factors (K) for various lubricants
  - — A table of torque versus nut factors to achieve 40%, 70%, and 85% based on the root area for various size and grade of fasteners
  - — The computation of the nut factor (K), given the friction coefficient (μ).

2.5.3.2 Procedure for Specifying Preload for Class 2 and Class 3 Joints

Although the ASME Boiler and Pressure Vessel Code does not require specification of the preload for ASME Section III,.Division 1, Classes 2 and 3 joints, it is often desirable to do so. The following is a procedure for specifying bolt preloads for doing this:

1. Document the following information:
   - — Class of the joint
   - — Materials (e.g., flange, gasket, fasteners, washers, lubricant)
   - — Design conditions (e.g., temperature and pressure)
   - — Geometry of the joint.
2. Determine the allowable stresses (e.g., gasket, bolt, flange).
3. Perform an ASME Section VIII Code calculation. The FLANGE computer program may be used (see Section 2.4.3).
4. Evaluate the resulting stresses:
   - — Gasket seating and operating stresses
   - — Flange stresses.
5. Calculate the average gasket stress using Equations 2-16 and 2-17. Use a bolt root stress of 45,000 psi as a first estimate.

$$S_g = F_p \; / \; \pi G n \qquad (2\text{-}16)$$

and

$$F_p = S_b A_r \qquad (2\text{-}17)$$

where,
S_g = Average gasket stress (psi)
F_p = Preload (pounds)
G = Mean gasket diameter (inches)
n = Gasket width (inches)
S_b = Bolt root stress (psi)
A_r = Bolt root area (inches^2)

The average gasket stress, calculated using Equations 2-16 and 2-17, should be less than the allowable gasket stress. Values for allowable gasket stress may be obtained from the gasket manufacturer.

The average gasket stress, calculated using Equations 2-16 and 2-17, should be less than the allowable gasket stress. Values for allowable gasket stress may be obtained from the gasket manufacturer.

Flange stresses are calculated in accordance with rules of ASME Section VIII, Division 2, of the Boiler and Pressure Vessel Code (16). An assembly bolt stress of 45,000 psi to 60,000 psi can result in high indicated stresses. The longitudinal stress (S_H) at the small end of the hub shown in Figure 2-12 often exceeds the flange material allowable and sometimes exceeds the yield strength.

Although these high stresses are calculated, they may not actually exist in the flange. The following paragraphs give possible explanations for the discrepancy:

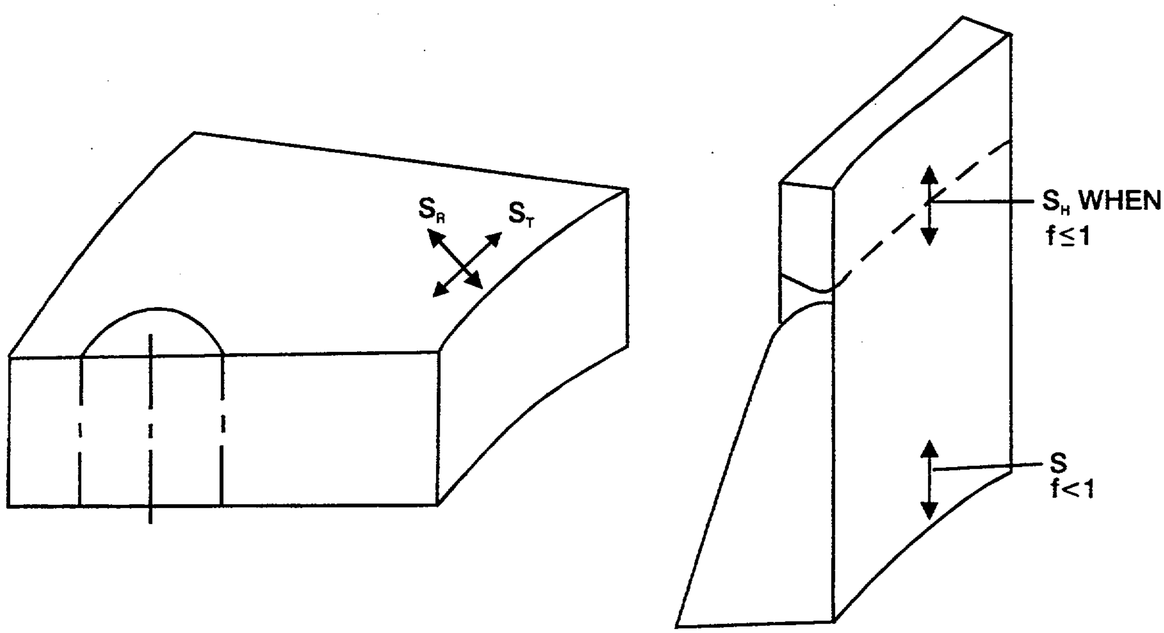
- The bolt loads are introduced one at a time in several passes around the flange. Elastic interaction between fasteners during assembly serves to reduce the average load on the flange (see Chapter 3, Section 3.7.1). Relaxation of the gasket, distortion of the flange, and localized yielding of the flange also tend to limit the amount of load applied at assembly.

- The usually high longitudinal stress, S_H, is a bending stress at the small end of the hub. As the actual stress in the flange at this location reaches the material yield, a "plastic hinge" develops. The stress is redistributed into the surrounding material of the hub. The following paragraph is an excerpt from ASME Section VIII, Division 2, Article 3-5 (16). The paragraph addresses the issues of localized plastic behavior, redistribution of load, and flange failure.

  Theoretically, the margin against flange yielding is not as great. The design values for flange materials may be as high as five eighths to two thirds of the yield strength. However, the highest stress in a flange is usually the bending stress in the hub or shell, and it is more or less localized. It is too conservative to assume that local yielding is followed immediately by overall yielding of the entire flange. Even if a "plastic hinge" should develop, the ring portion of the flange takes up the portion of the load the hub refuses to carry. Yielding is far more significant if it occurs first in the ring, but the limitation in the rules on the combined hub and ring stresses provides a safeguard.

2.5.4 Analysis Method to Determine Preload

Using a flange model as described in Section 2.4.1 and the FLANGE computer program, Rodabaugh and Moore (7) performed a study on the full range of sizes and pressure ratings of ANSI B16.5 flanges to determine a suitable assembly preload. The objective of the study was to determine the preload required to give leak free performance and meet requirements



S	=	Bolt Stress
S_H	=	Calculated Longitudinal Stress in Hub (PSI)
S_R	=	Calculated Radial Stress in Flange (PSI)
S_T	=	Calculated Tangential Stress in Flange (PSI)
f	=	Hub Stress Correction Factor for Integral Flanges

Figure 2-12 — Flange Stresses on a Hub.

for ASME Section III, Class 1 service. The following items were required to perform the study:

- A model of the joint. Rodabaugh and Moore used the general model described in Section 2.4.1 and Figure 2-11. This model allowed them to model preload conditions; introduce operating loads, such as pipe bending moments and thermal gradients; and calculate the effect of the external loads on the bolt, the gasket, and the flange.
- A set of operating conditions. The authors developed a set of operating conditions, shown in Table 2-3. These conditions represent a worst case scenario for bolt stress, gasket stress, and leak potential.

Table 2-3

Loadings Used in Estimating Preload Average Bolt Stress

Condition	Item
a.	Internal pressure equal to the primary rating pressure
b.	Decrease in modulus of elasticity from 30,000,000 psi to 23,000,000 psi
c.	Bolt temperature 50°F higher than flange ring temperature
d.	Pipe and hub temperatures 100°F higher than flange ring temperature
e.	Pipe moment load producing a bending stress, S_{pb}, of 8,750 psi in the attached pipe

2.5.4.1 Discussion of Loadings

If the bolt temperature is 50°F higher than the flange ring temperature, given in condition (c) of Table 2-3, a decrease in bolt stress and gasket stress will occur. The bolts expand faster than the ring. Pipe and hub temperatures 100°F higher than flange ring temperature, given in condition (d) above, cause flange rotation in the direction that reduces bolt load. Conditions (c) and (d) do not normally occur simultaneously; assuming both occur is a worst case for reduction in bolt load and gasket stress.

Rodabaugh and Moore (7) first defined the operating gasket stress at which leaks occur. They defined leakage as the condition in which the hydrostatic end load is equal to the bolt load in operation. The bolt stress at the leak condition is called S_{bc}. Under this condition, a leak is produced because the compressive load on the gasket is zero. This condition provides a non-conservative view of leakage since leaks actually occur long before the residual load on the gasket reaches zero. Despite the definition of leakage, the study determined how much the preload degrades due to operating conditions.

The results of Rodabaugh and Moore's study are illustrated in Table 2-4. The table compares the calculated preload bolt stress needed to seal (S_{b1}), the operating bolt stress at leakage (S_{bc})the bolt stress equivalent to the design allowable bolt stress per Section VIII of the ASME Code ($S_{b\,ASME}$), and the bolt stress calculated by Petrie's ($S_{b\,Petrie}$) Equation, Equation 2-15. Comparisons of the bolt preloads given in Table 2-4 indicate:

- The preload calculated by Rodabaugh and Moore is lower than that calculated by Petrie's Equation.
- The Rodabaugh and Moore preload is approximately two to five times higher than the ASME Code allowable load.
- $S_{b\,ASME}$ assembly bolt stresses based on the higher of the Boiler and Pressure Vessel design loads W_{m1} or W_{m2}, are inadequate. In most cases, the critical bolt stress at leakage is higher than the assembly bolt stress.
- The difference between the assembly stress (S_{b1}) and the operating bolt stress at leakage (S_{bc}) represents the loss of bolt load due to operating conditions. The flange is assembled to a bolt load of S_{b1}; operating conditions (i.e., pressure, temperature gradient, flange rotation, etc.) cause a reduction in bolt load to a value of S_{bc}. Comparison of the S_{bc} and S_{b1} columns in Table 2-4 shows that the bolt load is reduced by 50% to 70% from assembly to operating conditions.

  This 50% to 70% reduction in bolt stress is conservatively high due to challenging loading conditions but is similar to results reported by Wesstrom and Bergh (17) and others.

Table 2-4 (7)

Initial Bolt Stresses (psi)

Class	Size (in)	S_{bc}	S_{b1}	$S_{b\,ASME}$	$S_{bPetrie}$
150	4	9400	26300	12900	56900
	8	17500	46600	16900	52000
	16	13800	36500	9200	45000
	24	13700	38600	7400	40300
300	4	10100	27200	8700	5200
	8	14400	37000	8200	48100
	16	11900	30800	5900	40300
	24	14000	34800	6700	36700
400	4	9000	28600	6600	48100
	8	14000	35800	8300	45000
	16	12400	31300	6400	38500
	24	13000	31700	6300	34100

Table 2-4 (7)					
Initial Bolt Stresses (psi)					
Class	Size (in)	S_{bc}	S_{b1}	$S_{b\ ASME}$	$S_{bPetrie}$
600	4	12600	29700	9900	48100
	8	15100	35900	9500	42500
	16	14800	34400	7800	36700
	24	16400	38200	8100	32900
900	4	10300	23800	8500	42500
	8	13700	30300	8900	38500
	16	18000	38600	9800	35300
	24	15400	35000	7900	28500
1500	4	12600	26800	11100	40300
	8	15000	32100	10300	35300
	16	14200	30300	8000	28500
	24	15300	32700	8000	24100
2500	4	13100	26500	12300	36700
	8	15100	30000	10800	31800
	12	15900	31500	10000	27200

In his discussion of Table 2-4, Rodabaugh (20) stated that

> *This table is not intended to give specific recommended values of bolt stress that should be used in assembly of B16.5 flanged joints. Rather it is intended to show that an initial bolt stress of 40,000 psi is usually needed and applied.*

Rodabaugh's choice of 40,000 psi for assembly bolt stress compares well with the experiences of others. Bickford (20) reports successful joint performance on a variety of process vessel joints using a bolt prestress of approximately 37,500 psi. Bickford controlled and verified the bolt stress levels by stretch measurements. If a less accurate preload control is employed, such as torque, a higher target prestress may be required to achieve the same average stress and performance.

Recent NMAC studies indicate that 50,000 psi to 60,000 psi assembly bolt stress may be required to suitably preload fasteners used in gasketed joints, subject to the following qualifications:

- Flanges are ANSI B16.5 or designed in accordance with ASME Section VIII, Division 1, Appendix II.
- Fastener material has a yield strength in excess of 100,000 psi.
- The gasket is not over stressed.

2.6 Leaks

2.6.1 Gasket Seating Stress — Factors and Misconceptions

Designers often rely on the suggested gasket factors listed in the ASME Boiler and Pressure Vessel Code to determine the gasket seating stress and the gasket stress at operation. Many designers use the values suggested in the Code and fail to consider the effects of cyclic and steady state pressure and temperature on the gasket and fastener stress.

The suggested "y" and "m" gasket factors listed in the ASME Boiler and Pressure Vessel Code have been discussed and debated by the technical community for decades. Much of the discussion has focused on the magnitude of "y" and "m" (see Section 2.2.2). Many experienced designers and maintainers of bolted joints use higher gasket factors. They multiply the values suggested in the Code by a factor of 1.5 to 2.0. Others increase the suggested "y" value and use the full width of the gasket rather than the effective seating width suggested by the Code. This method increases the gasket seating stress.

Early papers and recent Pressure Vessel Research Council (PVRC) (4) tests have demonstrated that higher initial gasket seating stress prevents leaks. The following arguments support the use of higher initial gasket seating stress:

- Assembly efficiency. When standard bolting tools and procedures are employed to assemble joints, the resulting bolt load is scattered (see Chapter 3, Section 3.8.1 for more detail). A difference in load of 40% from bolt to bolt is not uncommon. The average load across the flange is often only 50% of the design load. To overcome the problem of a scattered bolt load, the bolt assembly stress must be high enough to allow for this assembly inefficiency.
- Relaxation. Relaxation occurring after assembly causes a loss of bolt load and gasket stress. The principal source of this relaxation is the gasket. The magnitude of the relaxation depends on the length of time, load level, material, temperature, and nature of the loading (i.e., steady or cyclic).

The ASME Boiler and Pressure Vessel Code committee is currently considering the incorporation of an assembly efficiency factor into the gasketed joint design procedure. The committee has suggested an efficiency of 80% for torque controlled assembly. For this efficiency, the assembly load should be increased by 25%.

2.6.2 Flange Rotation & Leaks

As noted previously, flange rotation can lead to leakage due to reduced gasket stress. Some utilities have reported that certain ANSI B16.5 flanges have been susceptible to leaks due to over preloading and subsequent flange rotation. The ASME Code is considering putting a limit on flange flexibility. The FLANGE program discussed previously or finite element analysis can be useful for calculating maximum preload limits to preclude overstressing, flange rotation, and leaks.

3.0 Assembly

3.1 Introduction

The last chapter focused on design considerations for pressure boundary joints. This chapter begins with an outline of general assembly procedures, and discusses:

- Considerations for establishing torque values, and development of torque tables to obtain desired gasket stress.
- Hydraulic tensioning, stretch of fasteners, and stretch control assembly procedures.
- Gasket compression and torquing assembly procedures.

As discussed in the previous chapter, the question of how much preload is appropriate is practically speaking somewhat ambiguous, and at best may be confusing. A key point was made however, in Section 2.5 that merits reiteration, i.e., the intent of the Code (specifically ASME VIII and ANSI B31.1) is to allow tightening to levels that are deemed sufficient for service, and the upper limit is that which does not excessively distort the flange or grossly distort the gasket. This statement is in general also true for safety related joints with the clarifier that an interaction analysis is required by the Code to preclude excessive loads.

Note: Section 3.3.7 presents one plants' successful approach towards dealing with how much preload is enough and appropriate. Incidents of leakage at this facility are reportedly very low. The user of this guide is encouraged to review and consider this success story for obtaining the compromise between enough preload and design considerations.

3.2 General Assembly Procedures

Equally important to the design of a bolted joint in determining performance is the assembly procedure. If the joint is not properly assembled, it will not perform as intended. Many variables affect the performance of a joint. Examples of these variables include smoothness and lubricity of all surfaces, condition of the parts (e.g., rust, tool marks, defects, etc.), hardness of the parts, calibration of the tools used on the parts, accessibility of the bolts, and environment in which the mechanics operate.

The following are guidelines for bolting assembly procedures:

- Be consistent. Do not magnify the variables that affect joint performance with inconsistent assembly procedures. Whenever possible, the mechanic should use the same tools in the same way and in the same sequence for each assembly.
- Train the bolting crews. Explain why good work practices are important. Warn the crews of problems that will be encountered if procedures are not followed. Training improves bolting results.

- Supervise the work, especially on critical joints.
- Keep tools in good repair. Tool repairs waste time and are counter productive. Calibrating and rebuilding the tools periodically ensures that they perform as required. A written procedure for assembling joints should be developed and include the following:
  - Joint identification, including number, system, location, material, size, etc.
  - Fastener identification, including size and grade
  - Tool identification and verification of current calibration sticker
- Detail assembly steps:
  - Cleaning of parts (e.g., solvents, etc.)
  - Lubrication: type, grade, and application
  - Visual inspection of components
  - Tool settings (e.g., pressure, torque, or turn)
  - Specification of the tightening sequence and tool setting for each pass
  - Check list with sign off blocks for bolting crew, supervisor, and quality assurance
  - Preparation of joint mating surfaces, including the following:
    - Clean the mating surfaces with a suitable solvent and wire bristle brush. Use stainless steel bristles on alloy components.
    - Inspect the seating surface for defects such as burrs and corrosion.
    - Inspect the surfaces for signs of warping.
    - Wire brush studs and nuts when necessary to remove any dirt from the threads. Use stainless steel bristles on alloy materials.
    - Inspect studs and nuts for burrs. Nuts should turn freely on studs to the in-service makeup position. If any burrs are present, one of the following steps should be performed.
    - Burrs of a minor nature may be filed off. Files used to remove burrs from alloy materials should not have been used previously on carbon steel materials.
    - If burrs are too numerous or too large to be filed off, the nut or stud should be returned to the store room for replacement.
  - Coat studs with an approved lubricant.
  - Coat the bearing surface of the turned element (the nut or the bolt head) with an approved lubricant.
  - If torque is used for assembly, use hardened washers between the turned element and the joint surface. Although hardened washers are not mandatory, they are helpful.

3.3 Torque

3.3.1 Short Form Torque/Preload Equation

A torque wrench is one of the most common tools used in a power plant to assemble a bolted joint. The short form torque/preload equation calculates the torque (t) required to produce a desired preload. This equation is given by

$$\tau = KDF_P \qquad (3\text{-}1)$$

where, τ = Torque (inch-pounds)
K = Nut factor
D = Nominal diameter (inches)
F_P = Preload (pounds)

Equation 3-1 gives the relationship between preload and torque. The two are related by the nominal diameter (D) and the nut factor (K).

The long form equation is discussed in Section 3.3.6.

3.3.2 Torque Preload Uncertainty

An Air Force study (4) identified 74 factors which significantly affect the relationship between torque and preload. Some of the more significant factors include:

- Lubricant and the method of application
- Hardness
- Surface finish
- Plating
- Material of fastener, joint, and washer
- Combination of materials
- Angularity of parts
- Thread fit.

Figure 3-1 gives the results of experiments used to determine the relationship between torque and tensile stress. Note the numerous torque/preload lines and the variability of stress for any given torque.

The variability of preload resulting from controlled torque is generally accepted to be ±30% for a metal-to-metal joint. For gasketed joints, the scatter may be as high as ±50%. NMAC has found that use of hardened steel washers in flanged joints signifiantly increases the resultant preload obtained for a degree of applied torque. This finding remains true even where improper lubrication, nut-reversed (nut grade markings down - bearing surface up), or angular misalignment conditions are found. Hardended steel washers also serve to preserve the surface condition of both the nut bearing surface and flange face after repetitive tensioning/detensioning cycles.

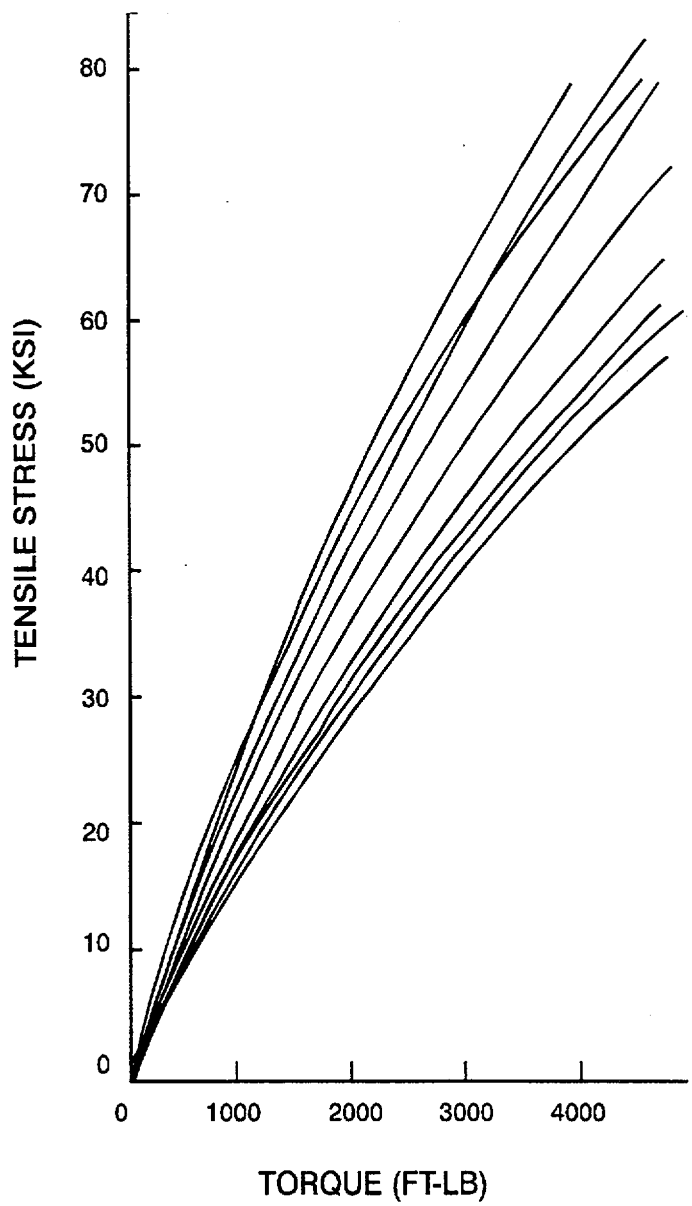


Figure 3-1 — Relationship of Torque and Tensile Stress

3.3.3 Nut Factors

The nut factor (K) is an experimentally determined factor which characterize the relationship between the torque applied to the nut and the preload achieved. It accounts for the many variables affecting the relationship between torque and preload. As a result, the nut factor is subject to wide variation, depending on the specific conditions under which it is measured.

Table 3-1 lists nut factors for various lubricants. The lubricant is not the only variable affecting the nut factor, however. The nut factor also depends on bolt diameter and material, tightening speed, thread fit, and operator skill. When accuracy is very important, the actual nut factor for a given application should be determined experimentally.

Table 3-1*

Reported Nut Factors for Various Materials and Lubricants

Metal/Lubricant	Minimum Nut Factor	Mean Nut Factor	Maximum Nut Factor
As Received Alloy or Mild Steel Fasteners	0.158	0.2	0.267
As Received Stainless Steel Fasteners	---	0.3	---
Cadmium Plate (Dry)	0.106	0.2	0.328
Copper Based Anti-Seize	0.08	0.132	0.23
Cadmium Plate (Waxed)	0.17	0.187	0.198
Fel-Pro C54	0.08	0.132	0.23
Fel-Pro C-670	0.08	0.095	0.15
Fel-Pro N 5000 (Paste)	0.13	0.15	0.27
Machine Oil	0.10	0.21	0.225
Moly Paste or Grease	0.10	0.13	0.18
Never-Seize (Paste)	0.11	0.17	0.21
Neolube	0.14	0.18	0.20

Table 3-1*			
Reported Nut Factors for Various Materials and Lubricants			
Metal/Lubricant	Minimum Nut Factor	Mean Nut Factor	Maximum Nut Factor
Phos-Oil	0.15	0.19	0.23
Solid Film PTFE	0.09	0.12	0.16
Zinc Plate (Waxed)	0.071	0.228	0.52
Zinc Plate (Dry)	0.075	0.295	0.53

**Reference EPRI NP5067 "Good Bolting Practices".*

The nut factor should be chosen carefully and should best approximate the assembly conditions. The following can significantly increase the nut factor:

- A bolt tightened into tapped holes, especially those with long thread engagements
- No hardened washer under the turned element
- Lack of perpendicularity of parts which causes high contact forces at the outer edge of the nut.

3.3.4 Lubricants

Lubrication of the fastener threads and the bearing surface of the turned element is essential when torque is used to control preload. Figures 3-2 and 3-3 illustrate the importance of proper lubrication. Figure 3-2 shows the galled threads of a 3-inch diameter A354 stud which was repeatedly preloaded to 450,000 lb by applying a torque of 20,000 foot pounds. The damage to the threads was due to the high surface loads and frictional forces on the threads. Figure 3-3 shows the bearing surface of a 3-inch diameter nut which was torqued to 20,000 foot pounds without lubricant. The surface damage was caused by the high bearing loads and the frictional forces acting on the surface during torquing.

In an unlubricated joint, 25% of the torque is used to overcome friction at the thread surfaces. When threads gall, however, there is no relative rotation between the nut and the bolt. Therefore, no increase in preload is achieved. Similarly, as much as 70% of torquing effort is used to overcome friction at the nut bearing surface.

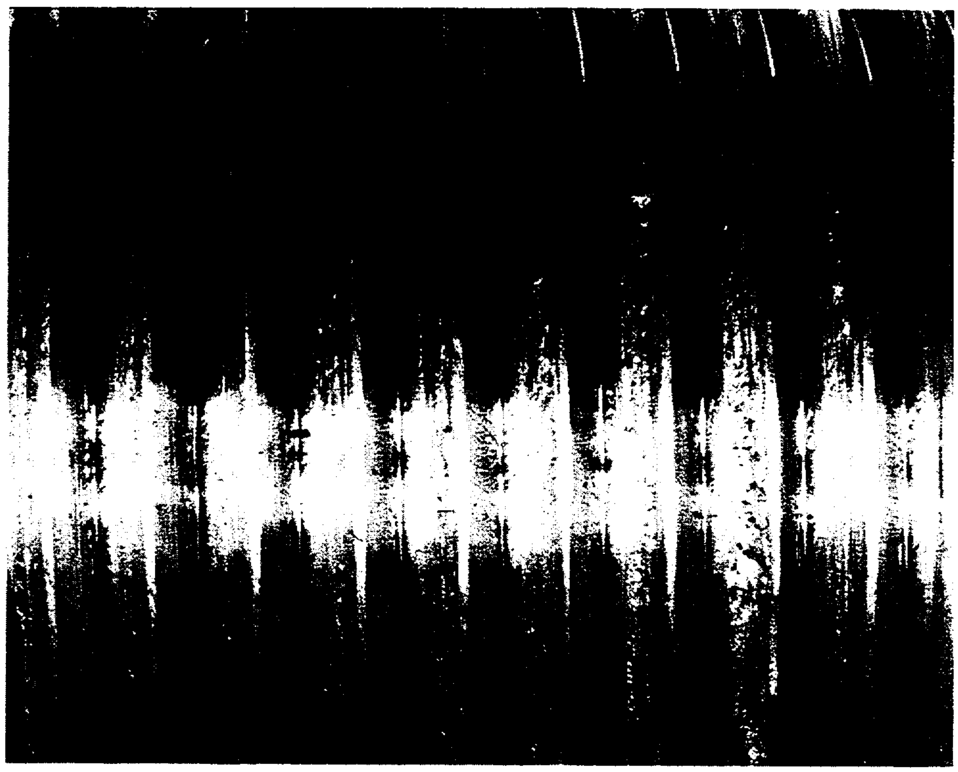


Figure 3-2 — Galled Threads of an A354 Stud.

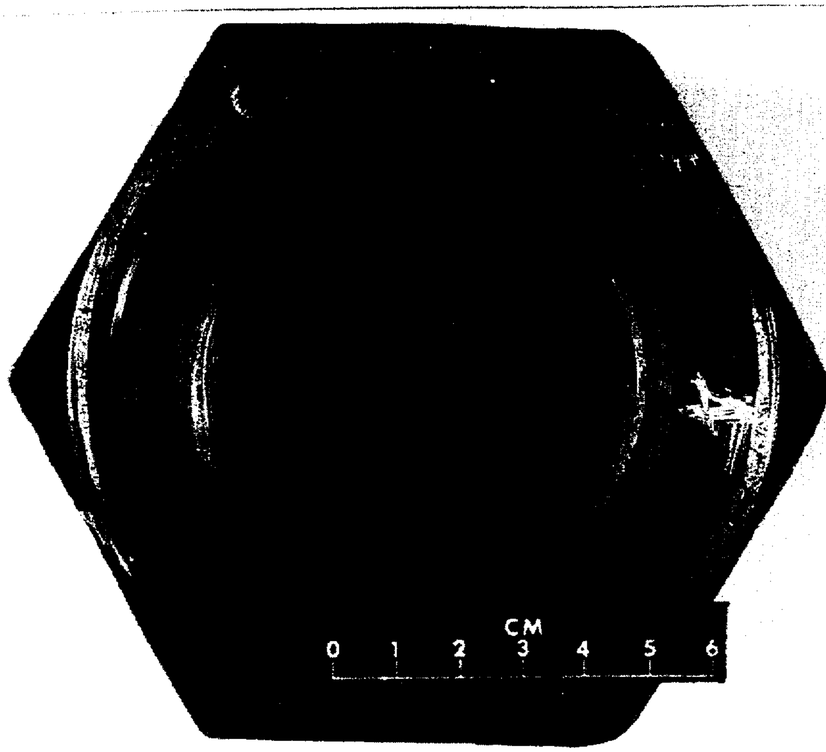


Figure 3-3 — Galled Surface of a Nut Torqued Without Lubricant.

Consider the following when selecting a lubricant:

- Compatibility. The lubricant must be compatible with the fastener material and with the contained fluid. Chlorides, fluorides, and sulfides are undesirable since they contribute to stress corrosion cracking. Copper based lubricants can contaminate primary reactor coolant fluids.

- Lubricity. Table 3-1 gives a wide range of nut factors for various lubricants. A lower nut factor is indicative of a more effective lubricant.

- Temperature. Each lubricant has a recommended temperature limit. Consult the manufacturer for additional information.

Consider the following when using a lubricant:

- Use only specified or approved lubricants on assemblies.

- Apply the lubricant in a consistent manner:
  - Lubricate threads as well as bearing surfaces.
  - Avoid over lubricating since the efficiency of the lubricant may be reduced.
  - Apply a thin uniform coating of lubricant to the parts.

- Close the lubricant container when not in use and store it in a clean, controlled area.

3.3.5 Torque Tables

Torque tables are generally developed from the short form torque/preload equation given by Equation 3-1. The assumptions made in creating torque tables should be stated in the table, including:

- The nut factor. Many tables are developed under the assumption that K = 0.2 (the "as received" condition).

- The basis used for the stress calculation. If the tables relate the torque to fastener stress, the cross sectional area used in the calculation (e.g., root, tensile, or nominal) should be stated.

Tables 3-2 through 3-4 list the torque required to develop 40%, 70%, and 85% of yield strength for various materials. The values given in the tables are based on a nut factor of 0.2 and the root area of the fastener.

Note that if the actual stress level and/or nut factor is different from that given in Tables 3-2 through 3-5, the torque value listed in the tables must be adjusted to fit these conditions. To determine the adjusted torque value corresponding to an actual stress level, multiply the listed torque value by the ratio of the actual stress level and/or nut factor to the listed stress level to the listed stress level and/or adjust for nut factor as shown below:

$$\tau_a = \tau_t(AS_b \; / \; AS_t)(NF_b \; / \; 0.2) \qquad (3\text{-}2)$$

where,
- τ_a = Adjusted torque (foot pounds)
- τ_t = Torque value listed in the table (foot pounds)
- AS_b = Actual stress for the bolt material to be torqued
- AS_t = Stress listed in the table
- NF_b = Nut factor for the bolt to be torqued

Table 3-5 provides torque tables for ASTM A307 Grade B material. Only one value of torque is given. Note that A307 Grade B material has no yield strength specified, consequently only a maximum torque based on tensile strength is given.

Table 3-2 (22)

Bolt Torque for ASME SA-193 Grade B7 Material

Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Code Torque (ft-lb)	S_y (psi)	Torque for 40%S_y (ft-lb)	Torque for 70%S_y (ft-lb)	Torque for 85%S_y (ft-lb)
1/2	0.50	0.4084	27	105000	46	80	97
5/8	0.63	0.5168	55	105000	92	161	195
3/4	0.75	0.6309	98	105000	164	287	349
7/8	0.88	0.7427	158	105000	265	464	564
1	1.00	0.8512	237	105000	398	697	846
1 1/8	1.13	0.9792	353	105000	593	1038	1260
1 1/4	1.25	1.1012	496	105000	833	1458	1771
1 3/8	1.38	1.2262	677	105000	1137	1989	2415
1 1/2	1.50	1.3512	896	105000	1506	2635	6199
1 5/8	1.63	1.4806	1166	105000	1958	3427	4162
1 3/4	1.75	1.6012	1468	105000	2467	4317	5242

Table 3-2 (22)

Bolt Torque for ASME SA-193 Grade B7 Material

Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Code Torque (ft-lb)	S_y (psi)	Torque for 40%S_y (ft-lb)	Torque for 70%S_y (ft-lb)	Torque for 85%S_y (ft-lb)
1 7/8	1.88	1.7262	1828	105000	3072	5375	6527
2	2.00	1.8512	2243	105000	3768	6594	8007
2 1/4	2.25	2.1012	3251	105000	5461	9557	11606
2 1/2	2.50	2.3512	4523	105000	7598	13297	16146
2 3/4	2.75	2.6012	5602	95000	9256	16197	11668
3	3.00	2.8512	7342	95000	12131	21229	25779
3 1/2	3.50	3.3512	11834	95000	19552	34216	41548

Table 3-3 (22)

Bolt Torque for ASME SA-453 Grade 660
Class A or B Material

Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Code Torque (ft\lb)	S_y (psi)	Torque for 40%S_y (ft-lb)	Torque for 70%S_y (ft-lb)	Torque for 85%S_y (ft-lb)
1/2	0.50	0.4084	23	85000	37	65	79
5/8	0.63	0.5168	46	85000	74	130	158
3/4	0.75	0.6309	83	85000	133	233	282
7/8	0.88	0.7427	134	85000	215	376	456
1	1.00	0.8512	201	85000	322	564	685
1 1/8	1.13	0.9792	299	85000	480	8408	1020
1 1/4	1.25	1.1012	421	85000	675	1181	1434
1 3/8	1.38	1.2262	574	85000	920	1610	1955

Table 3-3 (22)							
Bolt Torque for ASME SA-453 Grade 660 Class A or B Material							
Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Code Torque (ft\lb)	S_y (psi)	Torque for 40%S_y (ft-lb)	Torque for 70%S_y (ft-lb)	Torque for 85%S_y (ft-lb)
1 1/2	1.50	1.3512	760	85000	1219	2133	2590
1 5/8	1.63	1.4806	989	85000	1585	2774	3369
1 3/4	1.75	1.6012	1245	85000	1997	3494	4243
1 7/8	1.88	1.7262	1550	85000	2487	4352	5284
2	2.00	1.8512	1902	85000	3050	5338	6482
2 1/4	2.25	2.1012	2757	85000	4421	7737	9395
2 1/2	2.50	2.3512	3835	85000	6151	10764	13071
2 3/4	2.75	2.6012	5164	85000	8281	14492	17598
3	3.00	2.8512	6768	85000	10854	18995	23065
3 1/2	3.50	3.3512	10908	85000	17494	30614	37175

Table 3-4 (22)							
Bolt Torque for ASME SA-320 Grade L43 Material							
Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Code Torque (ft\lb)	S_y (psi)	Torque for 40%S_y (ft-lb)	Torque for 70%S_y (ft-lb)	Torque for 85%S_y (ft-lb)
1/2	0.50	0.4084	27	105000	46	80	97
5/8	0.63	0.5168	55	105000	92	161	195
3/4	0.75	0.6309	98	105000	164	287	349
7/8	0.88	0.7427	158	105000	265	464	564
1	1.00	0.8512	237	105000	398	697	846

Table 3-4 (22)

Bolt Torque for ASME SA-320 Grade L43 Material

Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Code Torque (ft\lb)	S_y (psi)	Torque for 40%S_y (ft-lb)	Torque for 70%S_y (ft-lb)	Torque for 85%S_y (ft-lb)
1 1/8	1.13	0.9792	353	105000	593	1038	1260
1 1/4	1.25	1.1012	496	105000	833	1458	1771
1 3/8	1.38	1.2262	677	105000	1137	1989	2415
1 1/2	1.50	1.3512	896	105000	1506	2635	3199
1 5/8	1.63	1.4806	1166	105000	1958	3427	4162
1 3/4	1.75	1.6012	1468	105000	2467	4317	5242
1 7/8	1.88	1.7262	1828	105000	3072	5375	6527
2	2.00	1.8512	2243	105000	3768	6594	8007
2 1/4	2.25	2.1012	3251	105000	5461	9557	11606
2 1/2	2.50	2.3512	4523	105000	7598	13297	16146
2 3/4	2.75	2.6012	6089	105000	10230	17902	21738
3	3.00	2.8512	7981	105000	13408	23464	28492
3 1/2	3.50	3.3512	12863	105000	21610	37818	45912

Table 3-5 (22)

Bolt Torque for ASME SA-307 Grade B Material

Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Maximum Bolt Load (lb)	Maximum Torque (ft-lb)
1/2	0.50	0.4084	917	8
5/8	0.63	0.5168	1468	15
3/4	0.75	0.6309	2188	27
7/8	0.88	0.7427	3033	44

Table 3-5 (22)				
Bolt Torque for ASME SA-307 Grade B Material				
Bolt Size (in)	Nominal Bolt Diameter (in)	Diameter at Root (in)	Maximum Bolt Load (lb)	Maximum Torque (ft-lb)
1	1.00	0.8512	3983	66
1 1/8	1.13	0.9792	5271	99
1 1/4	1.25	1.1012	6667	139
1 3/8	1.38	1.2262	8266	189
1 1/2	1.50	1.3512	10038	251
1 5/8	1.63	1.4806	12052	326
1 3/4	1.75	1.6012	14095	411
1 7/8	1.88	1.7262	16382	512
2	2.00	1.8512	18841	628
2 1/4	2.25	2.1012	27273	910
2 1/2	2.50	2.3512	30393	1266
2 3/4	2.75	2.6012	37199	1705
3	3.00	2.8512	44693	2235
3 1/2	3.50	3.3512	61743	3602

3.3.6 Long Form Torque/Preload Relationship

As an alternative to the short form torque/preload equation (Equation 3-1) discussed in Section 3.2.1, the relationship between torque and preload may be expressed by the long form torque/preload equation. As discussed below, it is really no improvement in accuracy over the short form equation but may prove useful when the thread and bearing surfaces are of different materials. The long form equation is given by:

$$\tau = F_p[(P / 2\pi) + (\mu_T R_T / \cos \beta) + \mu_N R_N] \qquad (3\text{-}3)$$

where,
- τ = Torque (inch-pounds)
- F_p = Preload (pounds)
- P = Thread pitch (inches/thread)
- μ_T = Coefficient of friction on thread flank
- R_T = Radius of thread flank (inches)
- μ_N = Coefficient of friction at bearing surface of turned element

R_N = Radius of nut bearing surface (inches)
β = Flank angle/2 (usually 30°) (degrees)

The relationship illustrated by Equation 3-3 is not as simple as that given by Equation 3-1. The long form equation accounts for friction effects and geometry associated with the thread (μ_T and R_T) and with the bearing area under the turned element (μ_N and R_N). In this equation, R_T is assumed to be the thread pitch radius, and R_N is assumed to be the mean radius of the bearing surface of the turned element. Table 3-6 lists coefficients of friction which can be used in Equation 3-3. Engineering handbooks provide additional information.

Table 3-6*

Coefficients of Static and Sliding Friction

Materials	Static Friction when Dry	Static Friction when Greasy	Sliding Friction when Dry	Sliding Friction when Greasy
Hard steel on hard steel	0.78	0.11(a)	0.42	0.029(k)
		0.23(b)		0.081(e)
		0.15(c)		0.080(i)
		0.11(d)		0.058(j)
		0.0075(p)		0.084(d)
		0.0052(h)		0.105(k)
				0.096(l)
				0.108(m)
				0.12(a)
Mild steel on mild steel	0.74		0.57	0.09(a)
				0.19(u)
Hard Steel on graphite	0.21	0.09(a)		
Hard steel on babbitt (ASTM No. 1)	0.70	0.23(b)	0.33	0.16(b)
		0.15(c)		0.06(c)
		0.08(d)		0.11(d)
		0.085(e)		

Table 3-6*

Coefficients of Static and Sliding Friction

Materials	Static Friction when Dry	Static Friction when Greasy	Sliding Friction when Dry	Sliding Friction when Greasy
Hard steel on babbitt (ASTM No. 8)	0.42	0.17(b)	0.35	0.14(b)
		0.11(c)		0.065(c)
		0.09(d)		0.07(d)
		0.08(e)		0.08(h)
Hard steel on babbitt (ASTM No. 10)		0.25(b)		0.13(b)
		0.12(c)		0.06(c)
		0.10(d)		0.055(d)
		0.11(e)		
Mild steel on cadmium silver				0.097(f)
Mild steel on phosphor bronze			0.34	0.173(f)
Mild steel on copper lead				0.145(f)
Mild steel on cast iron		0.183(c)	0.23	0.133(f)
Mild steel on lead	0.95	0.5(f)	0.95	0.3(f)
Nickel on mild steel			0.64	0.178(x)
Aluminum on mild steel	0.61		0.47	
Magnesium on mild steel			0.42	
Magnesium on magnesium	0.6	0.08(y)		
Cadmium on mild steel			0.46	
Copper on mild steel	0.53		0.36	0.18(a)
Nickel on nickel	1.10		0.53	0.12(w)
Brass on mild steel	0.51		0.44	
Brass on cast iron			0.30	

Table 3-6*				
Coefficients of Static and Sliding Friction				
Materials	Static Friction when Dry	Static Friction when Greasy	Sliding Friction when Dry	Sliding Friction when Greasy
Zinc on cast iron	0.85		0.21	
Copper on cast iron	1.05		0.29	
Tin on cast iron			0.32	
Aluminum on aluminum	1.05		1.4	
Bronze on cast iron			0.22	0.77(n)

(a) oleic acid; (b) Atlantic spindle oil (light mineral); (c) castor oil; (d) lard oil; (e) Atlantic spindle oil plus 2% oleic acid; (f) medium mineral oil; (g) medium mineral oil plus 0.5% oleic acid; (h) stearic acid; (i) grease (zinc oxide base); (j) graphite; (k) turbine oil plus 1% graphite; (l) turbine oil plus 1% stearic acid; (m) turbine oil (medium mineral); (n) olive oil; (p) palmitic acid; (q) ricinoleic acid; (r) dry soap; (s) lard; (t) water; (u) rape oil; (v) 3-in-1 oil; (w) octyl alcohol; (x) triolein; (y) 1% lauric acid in paraffin oil

**Marks Standard Handbook for Mechanical Engineers, 9th Edition, McGraw Hill Book Company.*

There are uncertainties associated with the friction and geometry terms in the long form equation. Factors which affect these terms include:

- Surface hardness and finish
- Material strength
- Lubricant
- Thread fit
- Perpendicularity of nut face to threads
- Stress levels.

Thus, the long form equation provides no improvement in accuracy over the short form equation. The long form equation may be applied in situations in which the thread and bearing surfaces are very different and thus require different coefficients of friction.

As an alternative to the long form equation, Table 3-7 (21) may be used to determine a nut factor (K) for different friction conditions at threads and bearing surfaces.

Table 3-7 (21)

Nut Factor (K)

Thread Friction (μ_T)	Friction on Bearing Surfaces (m_N)						
	0.08	0.10	0.125	0.14	0.16	0.20	0.25
0.08	0.118	0.131	0.148	0.158	0.171	0.198	0.231
0.10	0.128	0.142	0.159	0.169	0.182	0.209	0.242
0.125	0.142	0.155	0.172	0.182	0.195	0.222	0.255
0.14	0.150	0.163	0.180	0.190	0.203	0.230	0.263
0.16	0.160	0.173	0.190	0.200	0.214	0.240	0.274
0.20	0.181	0.195	0.211	0.221	0.235	0.261	0.295
0.25	0.208	0.221	0.238	0.248	0.261	0.288	0.321

3.3.7 Torque Tables for Desired Gasket Stress

Torque tables should specify flange facing, pressure rating, bolt material, and gasket type. These data are essential since gasket requirements primarily dictate the required preload. The required preload is different for each type of gasket.

Tables 3-8 through 3-11 were developed by Callaway Nuclear Plant using a procedure based on gasket stress which is illustrated in Figure 3-4. The assumptions for these tables are:

- Torque values
  - Bolt threads and all bearing surfaces lubricated with Fel-Pro N5000, nut factor is 0.16.
  - Calculated stress is based on the fastener root area

- Gasket stress is based on:
  - Full width of the gasket
  - Manufacturer's maximum and minimum gasket stress are noted for each pressure class.

- Even tightening versus specific tolerances is emphasized.

- Level 0 gives the bolt stress required to achieve 130% of the manufacturer's minimum recommended gasket stress for a range of ANSI B16.5 flange sizes.

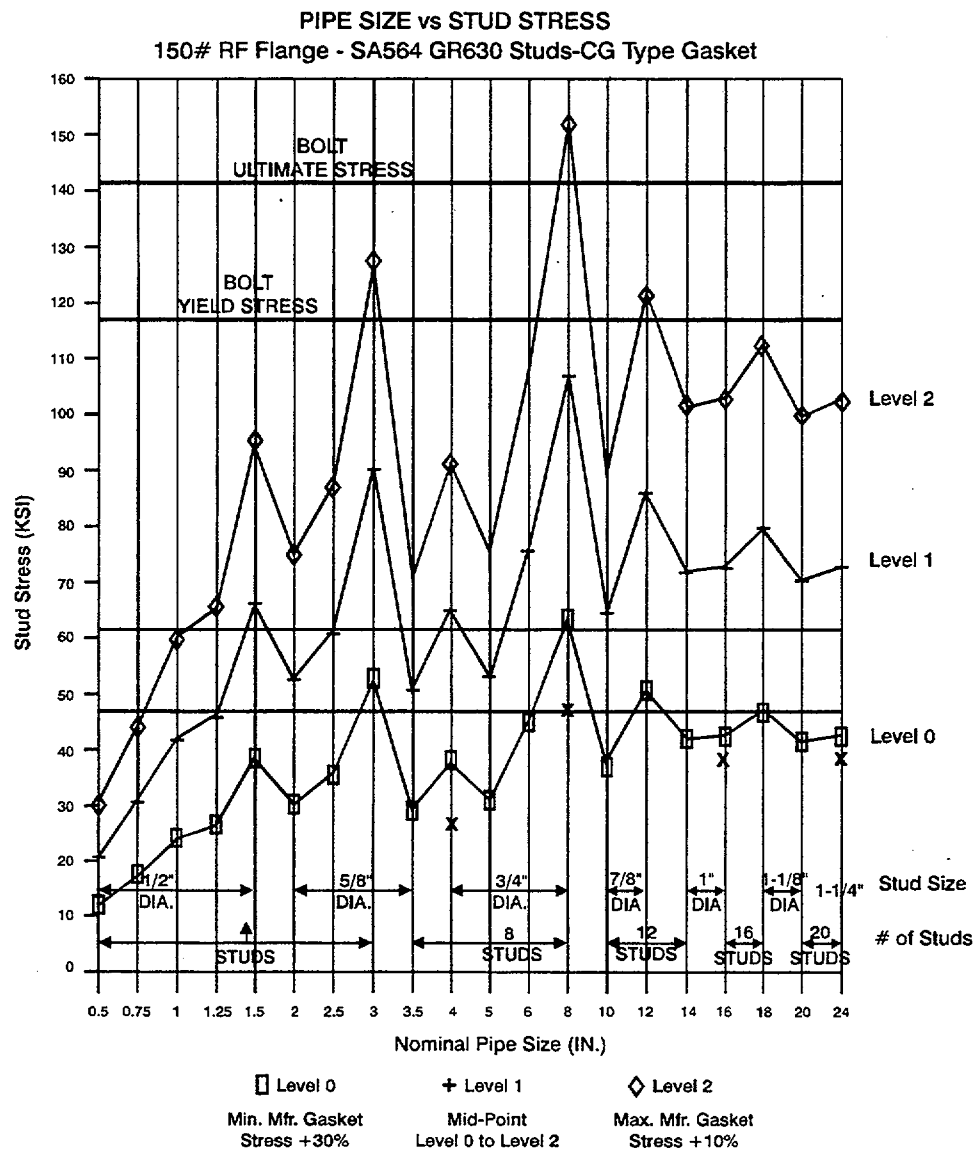


Figure 3-4 — Procedure for Developing Torque Tables for Gasketed Joint Assembly.

Reference Colin Pilkington, Callaway Nuclear Plant.

- Level 1 gives the bolt stress required to achieve a gasket stress at the mid-point between Levels 0 and 2 and is the normal recommended value for gasket assembly stress. Level 0 was considered too low due to the preload uncertainties developed using torque controlled assembly.

- Level 2 gives the bolt stress required to achieve 110% of the maximum recommended gasket stress for these flanges. It was chosen as the maximum assembly gasket stress allowed for maintenance use. Level 2 is to be used on a discretionary basis by maintenance with no engineering approval required when joint history necessitates a higher torque.

- Joints that do not seal using Level 2 gasket stress are referred to engineering for review and disposition.

Table 3-8*

150# Flange Raised Face Flanges; SA-193 GR.B7 or SA-564 GR.630 Fasteners, Spiral Wound Type CG Gasket

Nominal Pipe Size (in)	Raised Face Inside Diameter (in)	Raised Face Outside Diameter (in)	No. of Bolts	Size of Bolts (in)	Level 1 Torque (ft-lb)	Level 2 Torque (ft-lb)
1/2	0.84	1.38	4	0.50	50	65
3/4	1.06	1.69	4	0.50	50	65
1	1.31	2.00	4	0.50	55	65
1 1/4	1.66	2.50	4	0.50	55	75
1 1/2	1.91	2.88	4	0.50	60	80
2	2.38	3.62	4	0.63	100	140
2 1/2	2.88	4.12	4	0.63	115	145
3	3.50	5.00	4	0.63	150	175
3 1/2	4.00	5.50	8	0.63	95	120
4	4.50	6.19	8	0.63	120	160
5	5.56	7.31	8	0.75	175	250
6	6.62	8.50	8	0.75	250	300
8	8.62	10.62	8	0.75	260	310
10	10.75	12.75	12	0.88	335	450

Table 3-8*						
150# Flange Raised Face Flanges; SA-193 GR.B7 or SA-564 GR.630 Fasteners, Spiral Wound Type CG Gasket						
Nominal Pipe Size (in)	Raised Face Inside Diameter (in)	Raised Face Outside Diameter (in)	No. of Bolts	Size of Bolts (in)	Level 1 Torque (ft-lb)	Level 2 Torque (ft-lb)
12	12.75	15.00	12	0.88	440	500
14	14.00	16.25	12	1.00	570	700
16	16.00	18.50	16	1.00	575	710
18	18.00	21.00	16	1.13	930	1060
20	20.00	23.00	20	1.13	820	935
24	24.00	27.25	20	1.25	1180	1400

Manufacturer recommended gasket stress: minimum 7,000 psi, maximum 20,000 psi.

**Reference Colin Pilkington, Callaway Nuclear Plant.*

Table 3-9*						
300# Flange Raised Face Flanges; SA-193 GR.B7 or SA-564 GR.630, Fasteners Spiral Wound Type CG Gasket						
Nominal Pipe Size (in)	Raised Face Inside Diameter (in)	Raised Face Outside Diameter (in)	No. of Bolts	Size of Bolts (in)	Level 1 Torque (ft-lb)	Level 2 Torque (ft-lb)
1/2	0.84	1.38	4	0.50	50	65
3/4	1.06	1.69	4	0.50	80	100
1	1.31	2.00	4	0.50	85	105
1 1/4	1.66	2.50	4	0.50	85	105
1 1/2	1.91	2.88	4	0.50	140	175

Table 3-9*						
300# Flange Raised Face Flanges; SA-193 GR.B7 or SA-564 GR.630, Fasteners Spiral Wound Type CG Gasket						
Nominal Pipe Size (in)	Raised Face Inside Diameter (in)	Raised Face Outside Diameter (in)	No. of Bolts	Size of Bolts (in)	Level 1 Torque (ft-lb)	Level 2 Torque (ft-lb)
2	2.38	3.62	4	0.63	80	100
2 1/2	2.88	4.12	4	0.63	140	175
3	3.50	5.00	4	0.63	140	175
3 1/2	4.00	5.50	8	0.63	160	210
4	4.50	6.19	8	0.63	185	245
5	5.56	7.31	8	0.75	210	280
6	6.62	8.50	8	0.75	210	280
8	8.62	10.62	8	0.75	340	480
10	10.75	12.75	12	0.88	380	500
12	12.75	15.00	12	0.88	560	745
14	14.00	16.25	12	1.00	500	685
16	16.00	18.50	16	1.00	785	965
18	18.00	21.00	16	1.13	875	1135
20	20.00	23.00	20	1.13	960	1225
24	24.00	27.25	20	1.25	1565	1900

Manufacturer recommended gasket stress: minimum 12,000 psi, maximum 23,000 psi.

**Reference Colin Pilkington, Callaway Nuclear Plant*

Table 3-10*						
600# Flange Raised Face Flanges; SA-193 GR.B7 or SA-564 GR.630, Fasteners Spiral Wound Type CG Gasket						
Nominal Pipe Size (in)	Raised Face Inside Diameter (in)	Raised Face Outside Diameter (in)	No. of Bolts	Size of Bolts (in)	Level 1 Torque (ft-lb)	Level 2 Torque (ft-lb)
1/2	0.84	1.38	4	0.50	50	65
3/4	1.06	1.69	4	0.63	80	105
1	1.31	2.00	4	0.63	85	110
1 1/4	1.66	2.50	4	0.63	85	105
1 1/2	1.91	2.88	4	0.75	140	175
2	2.38	3.62	8	0.63	80	110
2 1/2	2.88	4.12	8	0.75	140	180
3	3.50	5.00	8	0.75	160	210
3 1/2	4.00	5.50	8	0.88	310	370
4	4.50	6.19	8	0.88	340	400
5	5.56	7.31	8	1.00	465	595
6	6.62	8.50	12	1.00	465	595
8	8.62	10.62	12	1.13	685	870
10	10.75	12.75	16	1.25	790	1050
12	12.75	15.00	16	1.25	875	1050
14	14.00	16.25	20	1.38	950	1190
16	16.00	18.50	20	1.50	1400	1725
18	18.00	21.00	24	1.63	1810	2220
20	20.00	23.00	24	1.63	1620	2020
24	24.00	27.25	24	1.88	2535	3170

Manufacturer recommended gasket stress: minimum 15,000 psi, maximum 27,000 psi.

**Reference Colin Pilkington, Callaway Nuclear Plant.*

Table 3-11*						
900# Flange Raised Face Flanges; SA-193 GR.B7 or SA-564 GR.630, Fasteners Spiral Wound Type CG Gasket						
Nominal Pipe Size (in)	Raised Face Inside Diameter (in)	Raised Face Outside Diameter (in)	No. of Bolts	Size of Bolts (in)	Level 1 Torque (ft-lb)	Level 2 Torque (ft-lb)
1/2	0.84	1.38	4	0.50	140	175
3/4	1.06	1.69	4	0.63	140	175
1	1.31	2.00	4	0.63	225	285
1 1/4	1.66	2.50	4	0.63	225	285
1 1/2	1.91	2.88	4	0.75	335	425
2	2.38	3.62	8	0.63	225	285
2 1/2	2.88	4.12	8	0.75	335	425
3	3.50	5.00	8	0.75	280	340
3 1/2	4.00	5.50	8	0.88	315	400
4	4.50	6.19	8	0.88	500	620
5	5.56	7.31	8	1.00	700	875
6	6.62	8.50	12	1.00	560	685
8	8.62	10.62	12	1.13	950	1185
10	10.75	12.75	16	1.25	950	1185
12	12.75	15.00	16	1.25	1065	1300
14	14.00	16.25	20	1.38	1250	1565
16	16.00	18.50	20	1.50	1615	2020
18	18.00	21.00	24	1.63	2535	3170
20	20.00	23.00	24	1.63	3100	3880
24	24.00	27.25	24	1.88	6220	7770

Manufacturer recommended gasket stress: minimum 17,000 psi, maximum 29,000 psi.

**Reference Colin Pilkington, Callaway Nuclear Plant.*

3.4 Hydraulic Tensioning

Hydraulic tensioners are widely used to preload large threaded fasteners (i.e., those generally greater than 1 1/2 inches diameter). It is assumed that tensioners provide near perfect preload control because the hydraulic ram exerts a controlled and accurate tensile force on the fastener during the assembly. The fastener, however, does not retain all of this load when the tensioner is removed. This loss of load is referred to as "tensioner efficiency". A review of the tensioning process identifies the factors which affect tensioner efficiency.

Figure 3-5 is a cross section of a typical hydraulic tensioner. The main features of the unit are the puller bar, annular hydraulic piston, nut rundown mechanism, and tensioner base. Figure 3-6 illustrates the tensioning process in the following steps:

1. Tensioner installation
   — The tensioner base is positioned over the stud and nut.
   — The puller bar is run down on the exposed stud threads.

2. Tensioner load application
   — Hydraulic pressure is applied to the tensioner.
   — An axial force is applied to the stud, causing it to stretch.
   — The joint reacts to the axial force by compressing under the tensioner base, under the lower nut, and at the joint interface. Note that the upper nut and the joint immediately below it are not stressed.

3. Nut run down
   — While the tensioner load is still applied, the upper nut is run down against the joint. This may cause the upper nut and the joint surface to be strained slightly, depending on the magnitude of the run down torque. This small compressive stress acts in parallel with the larger stress under the tensioner base.

4. Tensioner pressure release
   — The hydraulic pressure is released and the tensioner is removed. The upper nut and stud now carry the full load. Material is embedded in the thread surfaces and the nut bearing surface. This embedded material causes a loss of load.

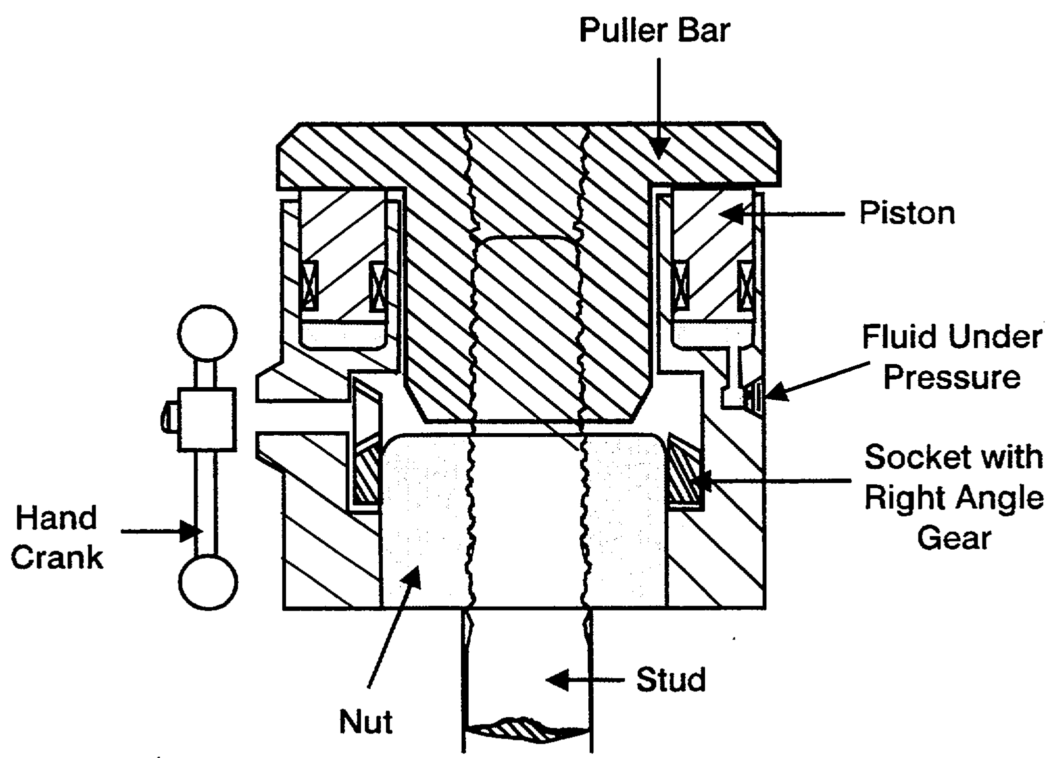


Figure 3-5 — Cross Section of a Hydraulic Tensioner.

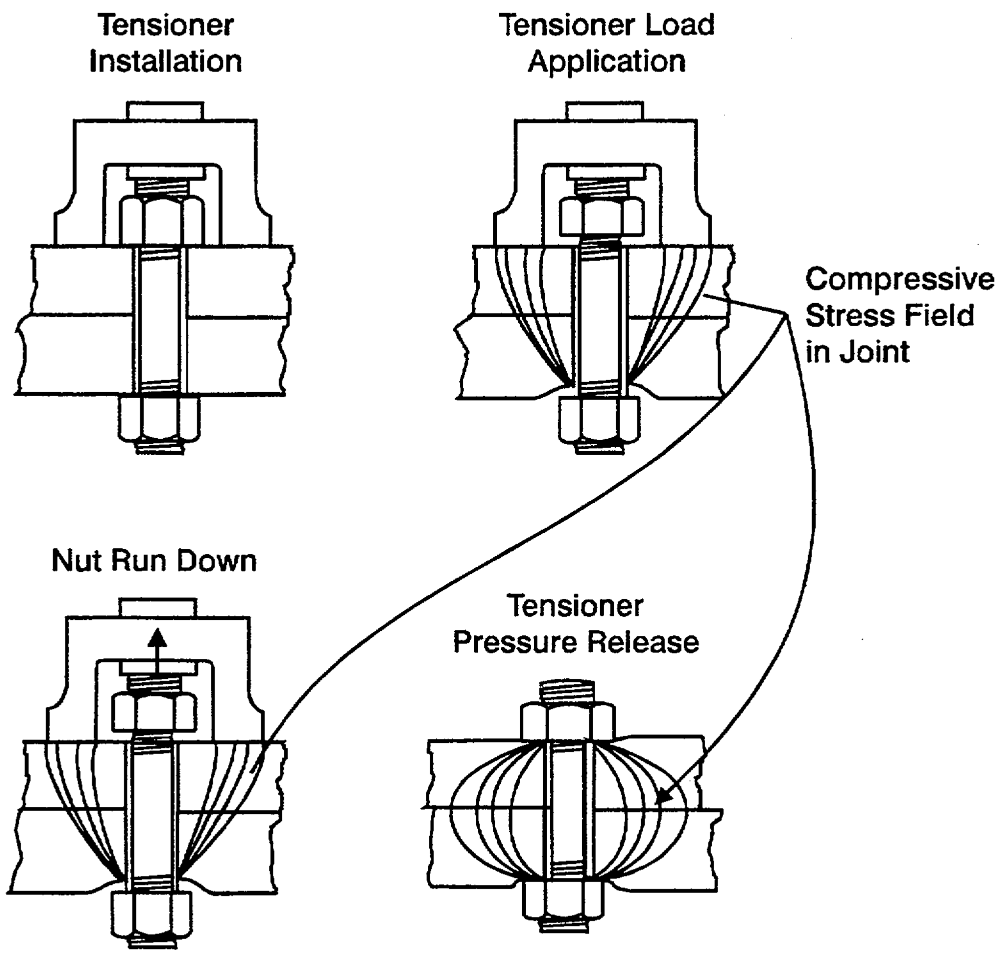


Figure 3-6 — Hydraulic Stud Tensioning Process.

3.4.1 Practical Considerations for Tensioning

The following should be considered before performing tensioning:

- Ensure that the tensioner has enough load capacity. Typically the tensioner load should be 25% to 30% higher than the preload desired in the stud.

- Ensure that the nut is run down firmly. This is the most important consideration in the tensioning process. If the nut is not run down firmly, zero preload can result. Nut rundown is adversely affected by the following:
  - A poorly constructed nut rundown mechanism. Right angle gear arrangements are preferable, and high and controlled rundown torque is desirable.
  - Avoid using fine stud threads. Fine stud threads can cause the nut to bind during rundown. Coarse threads are preferable.
  - A tensioner base that does not fit squarely on the joint surface. Check the base for signs of yielding or distortion. An ill fitting base can create interference with the nut and thus prevent nut rundown.
  - Studs that are not perpendicular to the joint surface. Non-perpendicularity results in stud bending and binding of the nut during rundown. Shimming the tensioner can correct for perpendicularity problems.

Figures 3-7 and 3-8 illustrate hydraulic tensioner efficiency. Figure 3-7 is a calibration curve of a long Inconel stud. The efficiency of the hydraulic tensioner is 75%. Figure 3-8 gives the relationship between the length to diameter ratio of the hydraulic tensioner and the tensioner efficiency.

There are two trends evident from Figure 3-8:

1. The longer length to diameter ratio studs have higher tension efficiency.

2. Coarse threads have higher tensioner efficiency than fine threads.

The longer studs are more efficient in tensioning since they experience greater stretch under action of the hydraulic tensioner. When the tensioner loads is released, the stud stretch is lost as the upper nut taper load is smaller on a percentage basis.

Coarse threads allow for better nut run down; there is less chance of binding between the nut and studs.

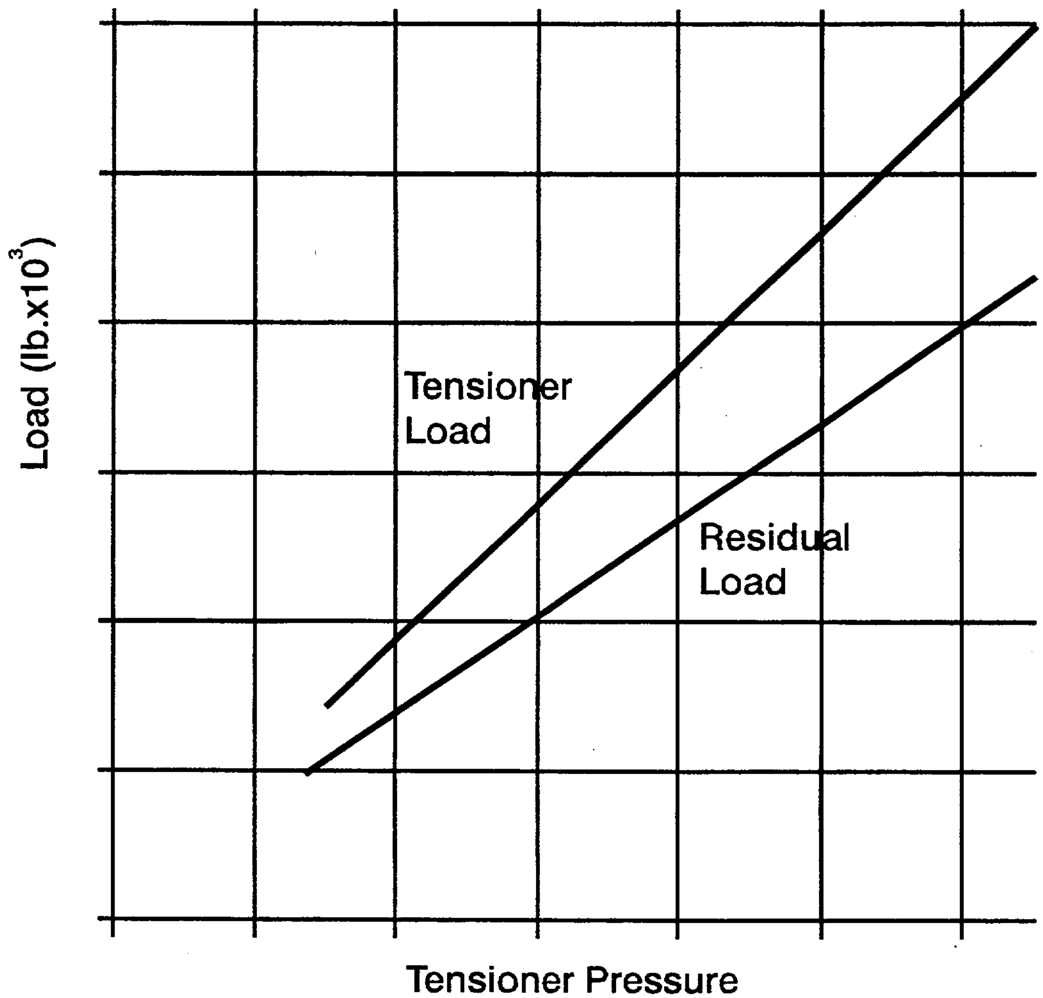


Figure 3-7 — Calibration Curve for an Inconel Stud.

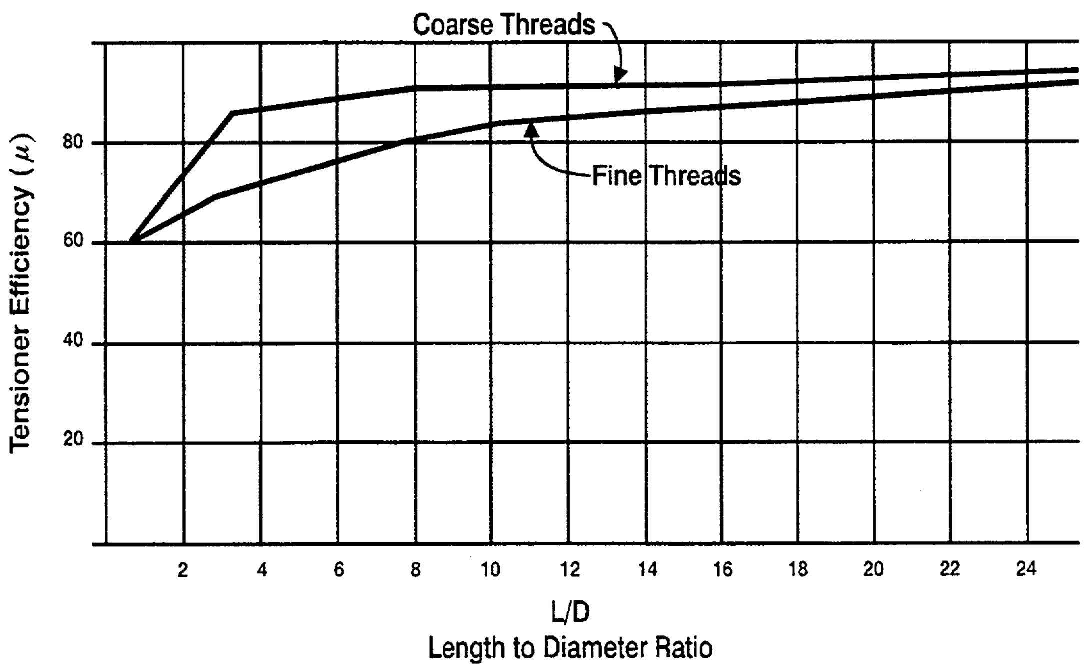


Figure 3-8 — Length/Diameter and Hydraulic Tensioner Efficiency.

3.4.2 Assembly Procedures for Tensioning

To effectively perform tensioning procedures, follow the tensioner manufacturer instructions and the following recommendations:

- Use multiple tensioners grouped together, if possible, to reduce elastic interactions.
- Perform the tensioning procedure <u>twice</u>. The second procedure compensates for any short term relaxation of the gasket. This step is extremely important for ring joints.
- Use uniform run down torque. Ensure that the nut turns.
- Verify that the specified hydraulic pressure is applied to the tensioner. This may be done by doing the following:
  - Check all hydraulic connections.
  - Check the pressure prior to running down the nut on each stud. When using multiple tensioners on one pump, running down one nut may reduce the system pressure.
  - Check that the nominal tensioner load is 20 lbs to 30 lbs higher than the desired stud preload. This over tension compensates for relaxation effects.
  - Perform a final check of the make up procedure after the specified tightening passes. This may be accomplished by applying the final tensioner pressure to each stud and attempting to run down the nut. If the nut moves, the residual preload is low and additional tightening passes are required.

3.5 Stretch of Fasteners

Measurement of bolt stretch is an accurate indicator of preload, provided it is measured with sufficient precision. Bolt stretch is used in many critical assemblies, including nuclear reactor pressure heads.

Elongation charts list the approximate stretch for various materials at different stress levels. An example elongation chart is given in Table 3-12.

Table 3-12					
Elongation Chart for Common Bolting Materials					
Bolting Material	Elongation at 20% of Yield*	Elongation at 40% of Yield*	Elongation at 60% of Yield*	Elongation at 80% of Yield*	Elongation at 100% of Yield*
ASTM A193 B8, B8M, B8C $E=28.5 \times 10^6$ psi $\sigma y=30$ ksi	0.2	0.4	0.6	0.8	1.0
Monel 40K $\sigma y=40$ ksi	0.3	0.5	0.8	1.1	1.3
SAE GR2 $\sigma y=55$ ksi	0.4	0.7	1.1	1.5	1.8
SAE GR3 $\sigma y=80$ ksi	0.5	1.1	1.6	2.1	2.7
SAE GR5, A325 $\sigma y=96$ ksi	0.6	1.3	1.9	2.6	3.2
ASTM A193 B7, B16 $\sigma y=105$ ksi	0.7	1.4	2.1	2.8	3.5
SAE GR8, A490 $\sigma y=120$ ksi	0.8	1.6	2.4	3.2	4.0
Inconel 718 $\sigma y=180$ ksi	1.2	2.4	3.6	4.9	6.1
4340 Steel RC47 $\sigma y=200$ ksi	1.3	2.7	4.0	5.3	6.6
Best Available High Strength Bolt Material $\sigma y=240$ ksi	1.6	3.2	4.8	6.4	8.0
Titanium 6AL4V $\sigma y=134$ ksi $E=17 \times 10^6$ psi	1.6	3.2	4.8	6.4	8.0

**Units for elongation are thousandths of an inch per inch of grip length. sy is yield strength. Modulus of Elasticity (E) is assumed to be 30 x 10^6 psi unless otherwise noted.*

Reference EPRI NP5067 "Good Bolting Practices".

The elongation for a particular material is found by multiplying the appropriate value given in Table 3-12 by the grip length in inches. For example, to determine the expected elongation for an SAE Grade 5 bolt stretched to 80% of yield with a 5 inches grip length, multiply 0.0026 inches/inches of grip length by 5 inches of grip length. This calculation gives an expected elongation of 0.013 inches

The data given in Table 3-12 are used to approximate elongation. More precise elongations are calculated using Hooke's Law:

$$\sigma_t = E\Delta L / L \tag{3-4}$$

where, σ_t = Tensile stress (psi)
E = Modulus of elasticity (psi)
ΔL = Stretch (inches)
L = Original unstrained length (inches)

Substituting F_p/A for the tensile stress in Equation 3-4 gives

$$F_p / A = E\Delta L / L \tag{3-5}$$

where, F_p = Axial load (lb)
A = Cross sectional area (inches^2)

A threaded fastener has non-uniform cross sectional areas, lengths, and stresses which must be considered when determining stretch. Figure 3-9 contrasts the actual stress distribution (Figure 3-9a) and an idealized stress distribution (Figure 3-9b) for a fastener. The threaded fastener acts like springs in series; the spring rate of each section must be considered. Figure 3-10 identifies the sections of a threaded fastener, the relationship of stretch and load for the configuration in Figure 3-10 is given by

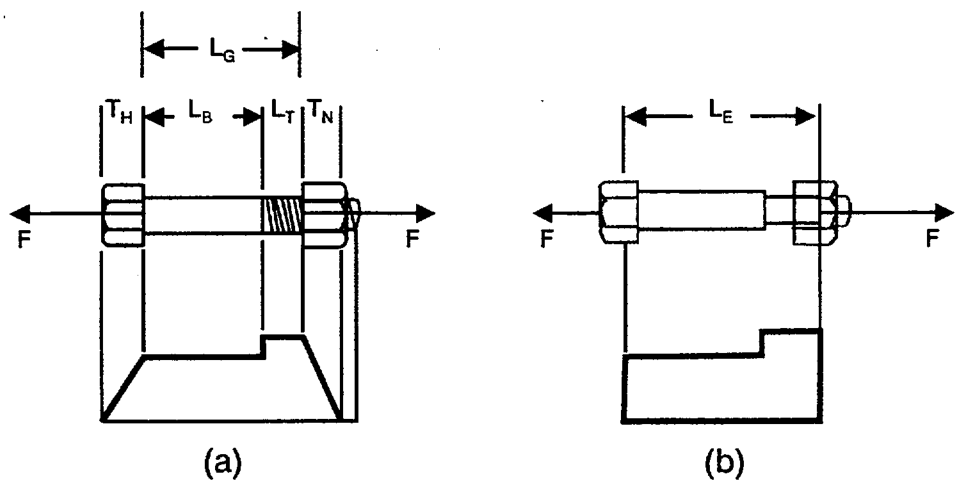
$$\Delta L = (F_p / E)(L_1 / A_1 + L_2 / A_2 + L_3 / A_3 + \ldots L_6 / A_6) \tag{3-6}$$

where L is the length and A is the cross sectional area for each section of the fastener as shown in Figure 3-10.

The stretch/load relationship for the more common nut and bolt configuration (see Figure 3-9) is given by:

$$\Delta L = (F_p / E)[(L_B / A_B) + (L_G - L_B) / As + (TH_1 + TN_2) / (2As)] \tag{3-7}$$

where, L_B = Length of bolt shank (inches)
A_B = Cross sectional area of bolt shank (inches)
L_G = Grip length (inches)
As = Thread tensile areas (inches^2)
TH_1 = Thickness of bolt heads (inches)
TN_2 = Thickness of nut (inches)



T_H	=	Head Thickness (inch)
T_N	=	Nut Thickness (inch)
L_T	=	Length of Threaded Region ($L_E = L_G + 1/2$) of Bolt (inch)
L_B	=	Effective Length of the Body of the Fastener (inch)
L_E	=	Effective Length of the Fastener
L_G	=	The Grip Length of the Fastener (inch)

Figure 3-9 — Comparison of Actual (a) and Idealized (b) Stress Distribution.

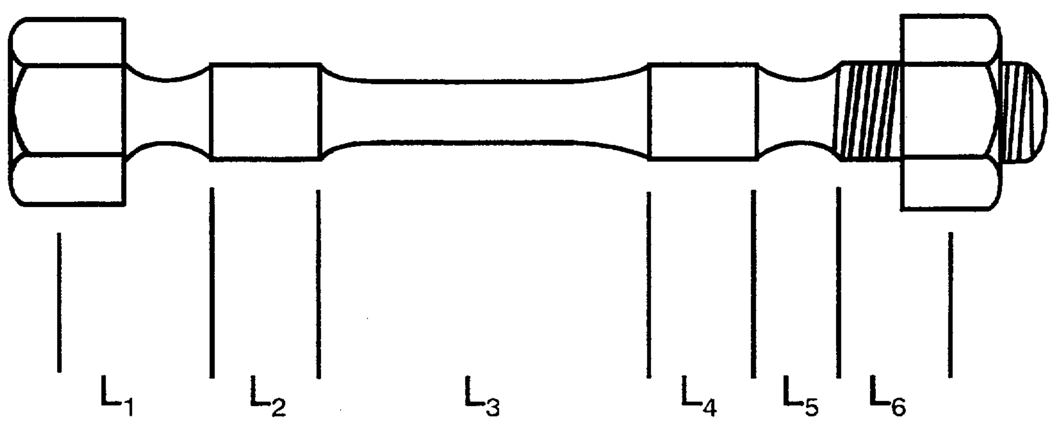


Figure 3-10 — Illustration of a Bolt Having Five Body Sections (L_1 - L_5) and One Thread Section (L_6).

Equation 3-7 is written with the assumption that one half the nut and one half the bolt head are included in the stressed length of the bolt.

Rule of Thumb: The stretch of a fastener can be estimated using the following rule of thumb: A fastener will stretch 0.001 inch per inch of effective length (grip length plus one nut height) per 30,000 psi stress. Applying this rule of thumb to SAE GR5 (5/8-inch diameter) loaded to 80% yield with a grip of 5 inches, for example, gives

$$\text{Stretch} = (0.001)[(0.8)(90{,}000) / 30{,}000 \times (5 + 0.625)] = 0.0135 \text{ inches}$$

3.5.1 Stretch Measurements

Micrometers, displacement gages, and ultrasonic extensometers are used to measure fastener stretch. A "C" type micrometer, illustrated in Figure 3-11, requires access to both ends of the fastener and a reasonably short fastener length.

A datum rod and depth micrometer are used to make stretch measurements if the fastener is drilled for the datum rod (see Figure 3-12). The stretch of the fastener relative to the datum rod is measured.

Displacement gages mounted on a reference frame are also used to measure the stretch of a fastener. Both ends of the fastener must be indicated.

Commercially available ultrasonic devices used for measuring bolt stretch, shown in Figure 3-13, are "time-of-flight" instruments. A pulse is introduced into one end of the fastener by a transducer. The ultrasound travels the length of the fastener, reflects off the far end, returns through the fastener, and is received by the transducer. The time required for a round trip is measured. Using various calibration factors, the system computes and displays the following:

- Initial length of bolt (i.e., before the load is added)
- Loaded length of bolt (i.e., after loading)
- Stretch.

Each of the three methods described above requires two measurements: (1) length of the unloaded fastener; and (2) length after the load is applied. The stretch is calculated from the difference in the two measurements. This requires accurate records.

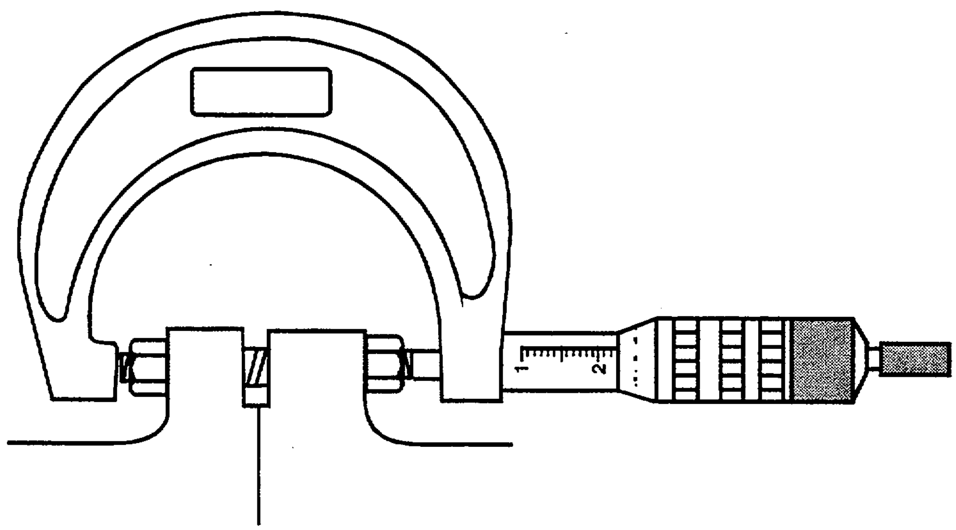


Figure 3-11 — "C" Micrometer Measurements.

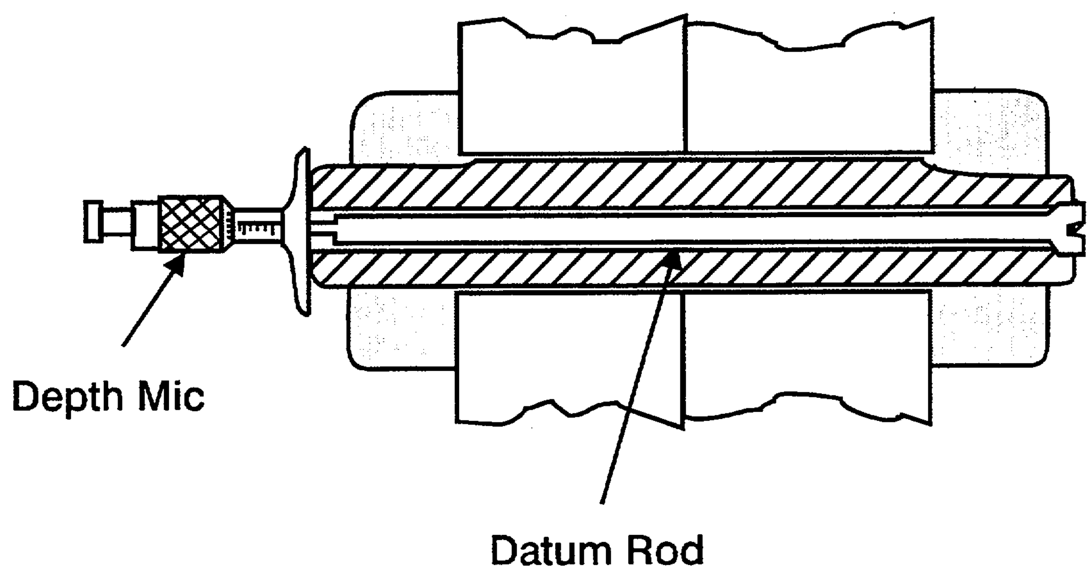


Figure 3-12 — Datum Rod and Depth Micrometer Can be Used to Measure the Stretch of the Bolt as Shown in the Illustration.

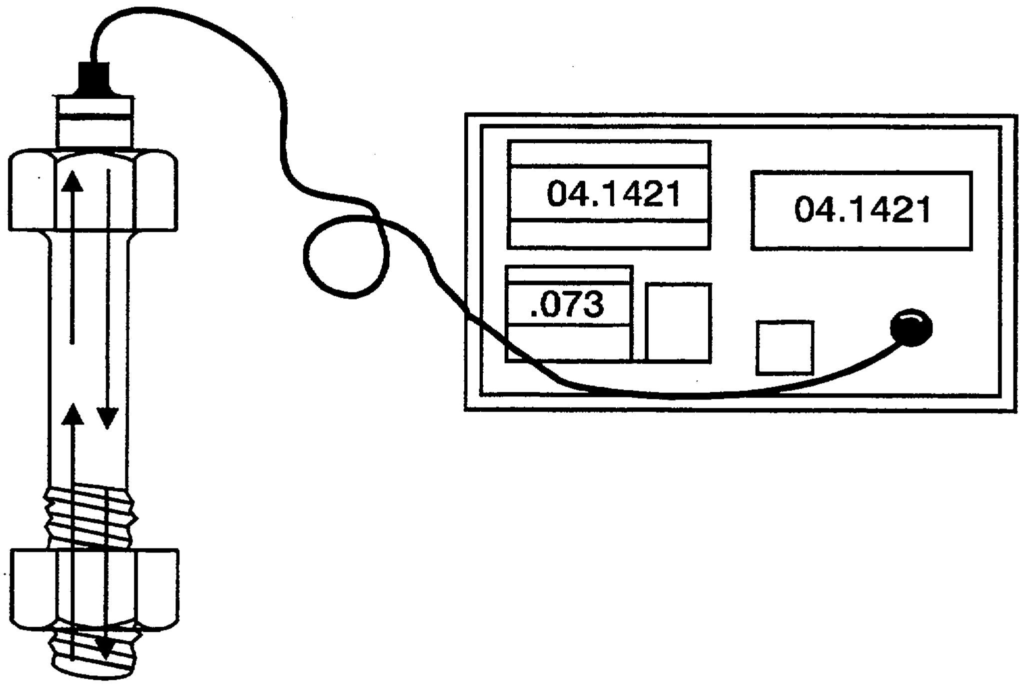


Figure 3-13 — Ultrasonic Device Used to Measure the Bolt Stretch.

3.6 Stretch Control Assembly Procedures

3.6.1 General Procedures

Parameters that affect the stiffness of a bolt should be monitored and controlled since stiffness determines the relationship between stretch and preload. The following factors should be controlled:

- Grip length of the assembly
- Body lengths and thread lengths of bolts.

Consider the following when making stretch measurements:

- Resolution of measuring device. The magnitude of a stretch measurement is usually thousandths of an inch. The resolution of the measuring device must be on the order of 0.0001 inches.
- Temperature of the fastener. Temperature affects both length and, for ultrasonic extensometers, the velocity of sound. Stretch readings must be compensated for changes in initial and final fastener temperatures. A change in temperature of 3°F for a steel fastener loaded to 50% of yield, for example, will produce a 1% error in the stretch measurement.
- Multiple measurements. Taking the average of several measurements is helpful when using a C-micrometer. An alternative is to measure over small balls pressed into the center of both ends of the fastener.
- Orientation of the measuring device. Always orient depth micrometers the same way when taking measurements. Also ensure that the datum rod seating configuration is clean and reliable.
- Reading and following manufacturer instructions. If using an ultrasonic extensometer to measure stretch, read and follow the manufacturer instructions carefully.

3.6.2 Specific Procedures

This section summarizes specific procedures to follow in taking fastener stretch measurements. The following suggestions improve the accuracy of the measurements:

- Machine the ends flat and parallel to each other with a surface finish of 125 μ-inches.
- For depth rod measurements, ensure that the datum rod seat is free of debris.

- Record the desired stretch and the recommended tool setting (i.e., torque or tensioner pressure).
- If the preload requirement is given as a load, convert it to a stretch using Equation 3-6 or 3-7.
- If the preload requirement is given as a torque, contact the vendor for the load specification. Convert this specification to a stretch using Equation 3-7.
- Measure and record the initial length of each fastener. Make at least two measurements of each fastener. The length measurements should be repeatable to within 1% to 2% of the desired stretch. Note that the stretch will be on the order of thousandths of an inch.

Stretch measurements are extremely valuable in the assembly of gasketed joints. They can aid the assembler by showing the level of preload existing in each bolt and the variability of load around the flange.

3.7 Gasket Compression Assembly Procedures

When using this assembly method, the stiffness of a gasket (i.e., stress and strain properties) becomes important. The stiffness of the gasket must be specified and controlled by the manufacturer of the gasket.

Measuring or controlling gasket deflection by use of compression stops is an effective method of assembling pressure retaining joints. The United States Navy has employed this method for many years.

If the gasket stiffness is controlled, gasket compression may provide a better indication of gasket loading than bolt torque. However, gasket compression is difficult to measure for most flanges. Vendors publish tables of the maximum recommended compression for their spiral wound gaskets. Portions of one vendor's table is reproduced in Table 3-13.

Table 3-13*			
Recommended Compression of Spiral Wound Gaskets			
Gasket Thickness (in)	Maximum Inside Dimension (in)	Recommended Flange Width (in)	Recommended Minimum/Maximum Compressed Thickness (in)
0.0625	Up to 6	3/8	0.050/0.055
0.100	10	1/2	0.075/0.080
0.125	Up to 20	1	0.090/0.100
0.175	Up to 40	1	0.125/0.135
0.250	90	1	0.180/0.200
0.285	185	1	0.200/0.220

**Reference Flexitallic*

3.7.1 Rubber Gasket Materials

Joints using rubber gaskets are generally found in service conditions which are not critical or severe (i.e., low temperature, low pressure, using water or a similar fluid). In most applications, the "skill-of-the-craft" method of assembly is suitable. In this method, the mechanic is instructed to tighten the bolt by manual wrenching with no torque measurement until the joint is tight. Uniformity of compression and tightness are important. The mechanic should be trained and cautioned not to over tighten, which could cause extrusion of the gasket material.

Measuring the gasket compression of a full faced rubber gasket is difficult since the gasket fills the space between the flanges and often protrudes beyond the outside diameter of the ring. In this case, deflection measurements are made with a caliper on the outside of the flange rings. The assembly compression of soft rubber type gaskets is usually 25% to 50% of the original thickness.

3.8 Torquing Assembly Procedures

The following is an example assembly procedure for torquing:

1. Ensure that all load bearing surfaces are in good condition. Check the thread flanks, the bearing surfaces of the nuts or bolt heads, the washers, and the flange surface.

2. Clean and lubricate all threads according to the governing plant procedure. Lubrication of all fastener and nut threads, as well as all bearing surfaces will increase the resultant preload achieved for the same amount of applied torque. Use the lubricant specified and apply it uniformly as directed.

3. Use hardened steel washers under the turned element to increase the resultant preload achieved for the same amount of applied torque.

4. Run the nuts or bolts down by hand. If a fastener does not run by hand, the thread is defective.

5. Use a multiple-pass, cross-bolting procedure:
   - If there is a gasket, ensure that it is compressed evenly. Use caliper measurements in four quadrants if necessary.
   - Torque as many studs as possible simultaneously. This will help pull the joint down evenly and reduce the loss of bolt load during the pass.
   - All torque wrenches should be of adequate capacity and have been recently calibrated.
   - All thread lubricants and anti-seize compounds should be approved. It is recommended that a limited number of types of lubricants be used. This helps avoid mistakes and provides a performance history for each lubricant.
   - Apply torque at a uniform rate. The final torque should be reached while the turned element is moving in the tightening direction, and not at the end of the tensioning movement.
   - Hold torque wrenches perpendicular to the axis of the bolt while torque is being applied.
   - If hydraulically powered torque wrenches are used, ensure that adequate reaction points are provided.

The following is the torquing assembly procedure for gasketed joints:

1. Tighten the joint with a minimum of four tensioning passes, using a cross-bolting sequence for each pass. Figure 3-14 gives typical flange cross-bolting patterns. For torque values up to 125 ft/lbs, use the sequence given below:
   - Pass 1: Bring all nuts up finger tight. Then tighten snugly and evenly.
   - Pass 2: Torque to a maximum of 30% of the final torque. Check that the flange is bearing uniformly on the gasket (i.e., uniform gap, parallel sealing surface).
   - Pass 3: Torque to a maximum of 60% of the final torque.
   - Pass 4: Torque to the final torque.

NOTE: It is extremely important to avoid using too high an initial torque, or too great an increase in applied torque between passes. Upon tightening the second fastener in the cross-bolting pattern, a 'prying' action will occur in the area of the first fastener tightened during the pass, resulting in overcompression of the gasket surface, creating the potential for subsequent leakage from that point. Maximum acceptable gasket compression induced is the governing factor used in calculating the torque level used during the initial tensioning pass, or the magnitude of incremental increases in torque in subsequent tensioning passes. To minimize this problem in joints over 125 ft.lbs, use good mechanics judgement to increase the number of additional tensioning passes as necessary in order to maintain the incremental increase in applied torque below that which might inadvertently overcompress the gasket.

2. After completing the required number of basic torquing passes, continue torquing the nuts in a clockwise manner until no further rotation of the nut is observed. This process may require an additional five to seven passes.

3. Torquing the fasteners of a joint in the reverse sequence in the final pass may improve preload uniformity. In critical situations, the preload achieved can be verified by making stretch measurements of the fastener.

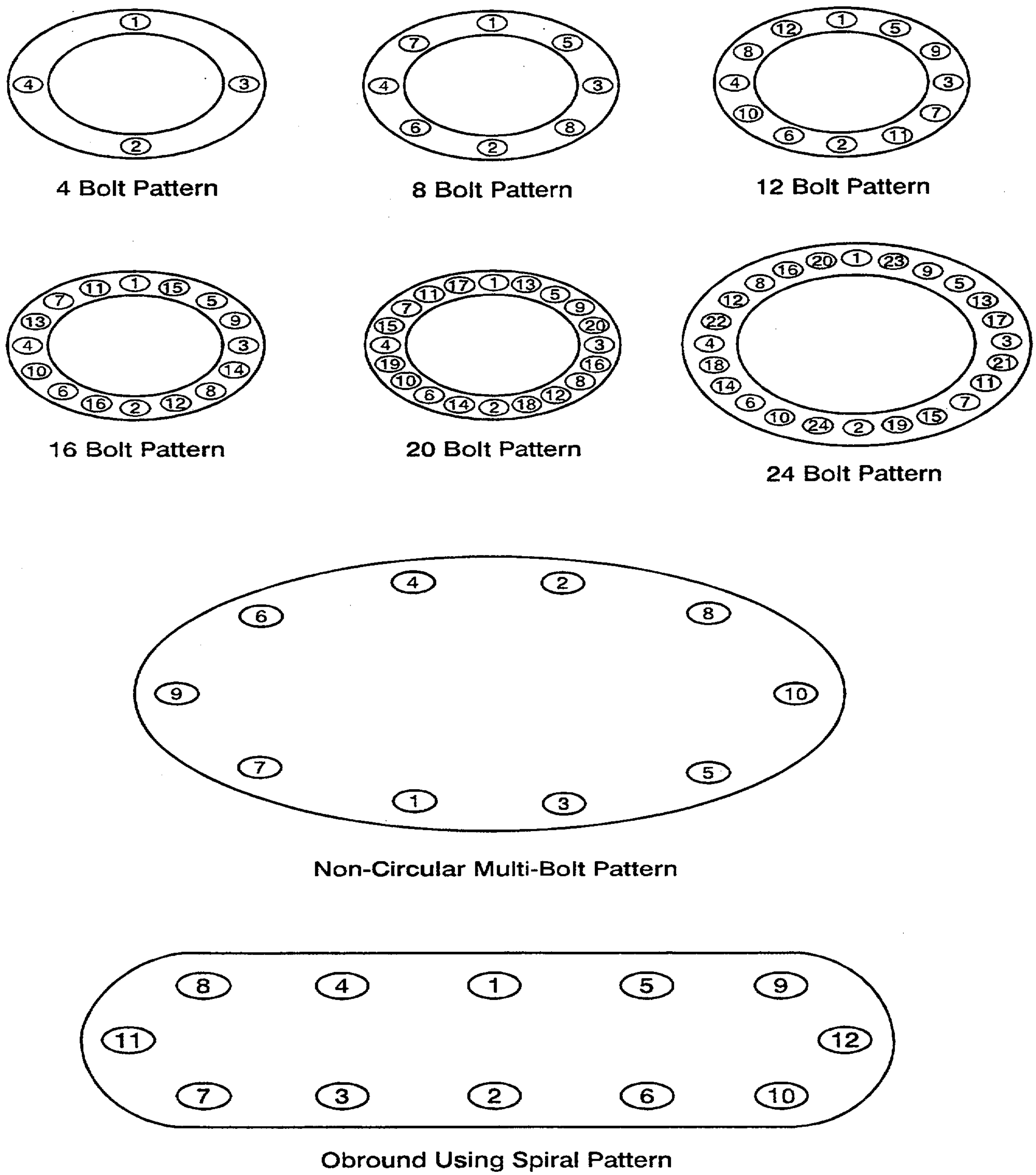


Figure 3-14 — Flange Bolting Patterns.

3.8.1 Elastic Interactions

Tightening one fastener will often partially loosen previously tightened fasteners near it. This "cross talk" between fasteners, during assembly or disassembly, is called elastic interaction.

Elastic interaction is one of the most common and most extreme forms of preload loss, especially in gasketed joints. It can cause a large scatter in the residual preload, as shown in Figure 3-15.

The plots (Figure 3-15) are the results of experiments designed to evaluate the preload achieved on a gasketed joint when the joint was torqued in a cross- bolting pattern:

3.8.1.1 Example of Elastic Interactions

Test Equipment:

Flange: 8-inch-600# ANSI B16.5, raised face, tapered nut weld neck flange bolted to a blind flange

Gasket: Spiral wound asbestos filled gasket Type CG

Fastener: 1-3/8-B7 studs, hardened washers

Lubricant: Fel-Pro N5000

Torque Wrench: Calibrated hydraulic torque wrench

Test Procedure:

The joint was assembled with a new gasket. The stud threads, washer, and nut bearing surfaces were lubricated. The joint was torqued in these passes using the torquing sequence shown in Figure 3-16.

Measurements:

Immediately after the application of torque to each fastener, the stretch was measured. These measurements are represented by "X" marked on the figure. After the complete pass was made at the given torque, the stretch of each stud was again measured (plotted as "0" on the figures).

The plots are the stud stretch (Y axis) versus the position of the studs relative to each other in a circumferential direction around the bolt circle (i.e., looking down on the flange stud). Starting at Stud 1, if we proceed clockwise around the flange stud 12 is next followed by 5 etc.

Figure 3-15 is typical load distributions resulting from torquing this 8-inch, 600 lb ANSI B16.5 raised face flange joint with a spiral wound gasket. Similar patterns have

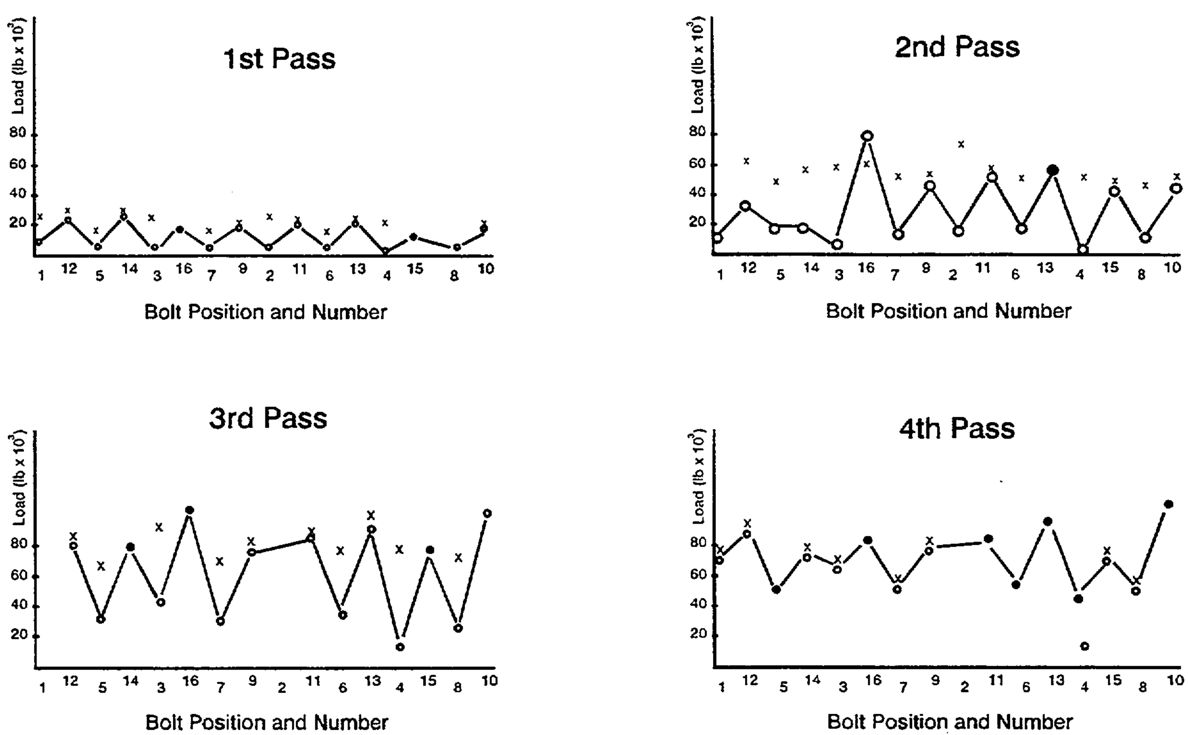
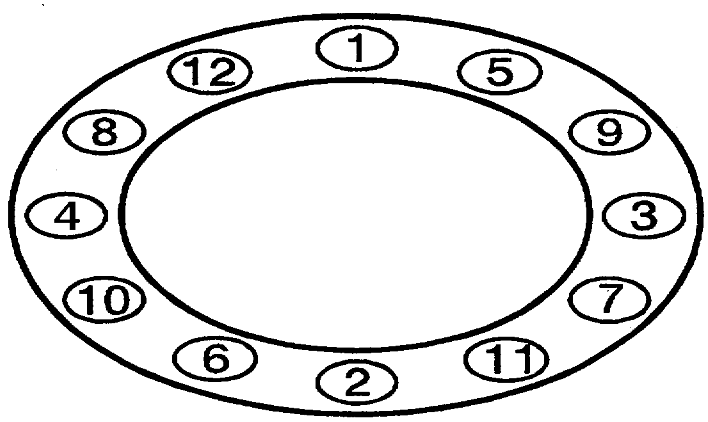


Figure 3-15 —Load Distribution in a Bolted Joint.



12 Bolt Pattern

Figure 3-16 — Bolt Tightening Sequence.

been found using stretch measurements on joints such as heat exchangers, manways, and pump housings.

The patterns after each pass indicate that the bolt loads fall into one of two groups. Bolts 1-8 have low residual loads, and bolts 9-16 have high residual loads. The loads in studs 1-8 at the time of torquing are approximately the same as those in studs 9-16. This indicates that the torque wrench and the nut factor are developing a preload with a scatter of less than ±30% of the target value. During each pass, the load introduced in bolts 1-8 is relieved when bolts 9-16 are tightened. Bolts 9-16 further compress the gasket and allow bolts 1-8 to relax. The average residual load in bolts 1-8 is 20% to 30% of the average of that in bolts 9-16.

The amount of scatter in residual preload is controlled by the stiffness of the gasket and the joint. A stiff joint and a flexible gasket give the greatest scatter. Because the gasket stiffens as the load is increased, preload scatter can be reduced as mentioned below.

3.8.1.2 Compensating for Elastic Interactions

In most cases, it is not necessary to compensate for elastic interactions. In a few cases, however, where flanges are relatively flexible or where serious problems with thermal effects exist, it is helpful to reduce the scatter residual preload. The following can minimize elastic interactions:

- Retighten the joint repeatedly. No sequential tensioning procedure exists to eliminate elastic interactions. To minimize scatter in the residual preload, after having reached the final torque using a cross-bolting tensioning pattern, continue tightening at the final torque using a sequential pattern around the flange until no further nut rotation is observed on any of the fasteners. Changing direction during the final pass may also reduce elastic interactions.

- Measure and control residual preload. Measuring and controlling the residual preload rather than the initial preload is the only infallible way to compensate for elastic interactions. Some type of stretch control (e.g., datum rods, strain gages, or load cells) or ultrasonic device must be employed. Relatively uniform residual preloads (±10% to 20%) can be achieved in the bolts of a joint. However, a different amount of torque may be required for each bolt.

Figure 3-17 gives a simplified model to explain joint interaction. Bolts 1 and 2 of this figure are loaded to same load (F_p). This load stretches the bolts and compresses the gasket an amount (Δq). The energy is stored in the gasket spring, which pushes on the bolts and holds the load in the bolts. The gasket elasticity, trying to return the gasket to the original thickness, keeps the preload on the bolts. If the gasket behaved like plastic, as does a lead washer, the gasket would flow or creep and the load on Bolts 1 and 2 would decrease.

In Figure 3-18, the same load (F_p) is applied to bolts 3 and 4, thus increasing the gasket compression. The load on Bolts 1 and 2 decreases since the gasket is compressed out from under them. For Bolts 1 and 2, the additional compression of the gasket caused by Bolts 3 and 4 is similar to the lead washer analogy in the previous paragraph.

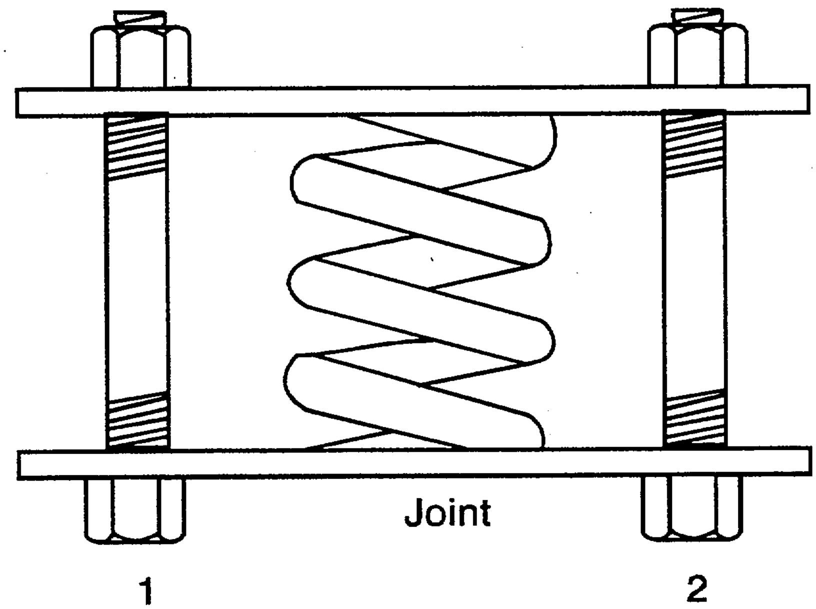


Figure 3-17 — Model of Joint Interaction for Two Bolts Loaded.

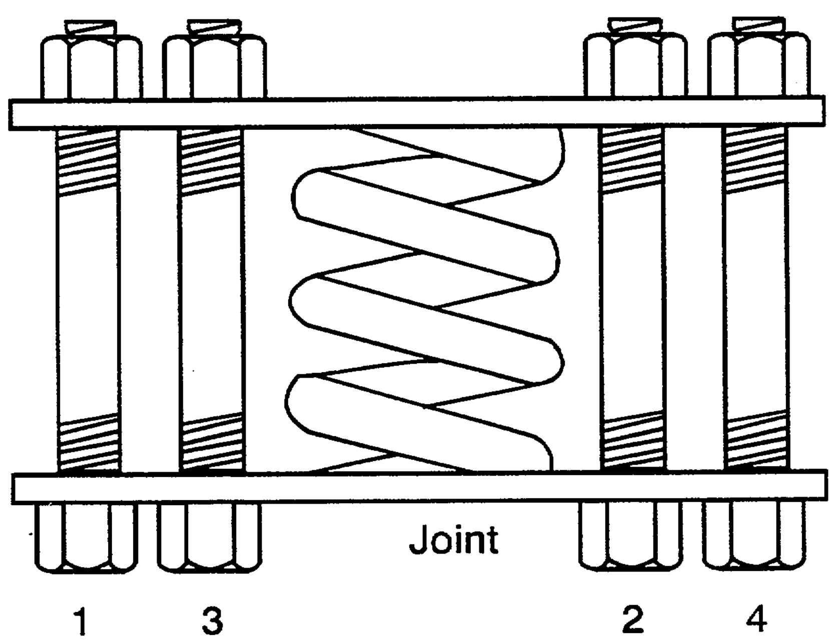


Figure 3-18 — Model of Joint Interaction for Four Bolts Loaded.

4.0 Inspection

4.1 Introduction

Inspections are performed several times during the life of a bolted joint and include receipt inspection and assembly and/or inservice inspection of the joint during operation. The purpose of receipt inspection is to assure that the joint materials are in accordance with specifications and will provide adequate service. Inspections performed during the operation of the system alert maintenance and operations to developing problems. When leakage is noted, detailed inspections of the troubled joint, both before and after disassembly, assist in determining the root cause of the leak. These inspections are essential in determining the proper corrective action required.

Receipt inspections are necessary in today's business environment. In the past, commercial grade hardware, such as bolts, nuts, gaskets, and flanges, could be ordered to recognized national specifications and be assured of quality. The distribution of counterfeit fasteners and flanges, however, has caused doubts in the reliability of certification or grade markings of parts. This situation has led many users of bolting products to verify the quality of the products by a receipt inspection program.

4.2 Receipt Inspection

This chapter will discuss the receipt inspection of gaskets and flanges. The receipt inspection of fasteners is covered in Chapter 16.

4.2.1 Gaskets

Receipt inspection of gaskets should be simple. A receipt inspection plan can help to determine the critical characteristics of the gasket and determine which inspections are required. Labels, dimensional checks, hardness, and general appearance are primary characteristics to be inspected. Gaskets packed on a backing board should remain on the board until they are installed. Dimensional checks of the inside and outside diameters of gaskets and the average gasket thickness should be performed. Note that some gasket materials swell slightly due to humidity in the air. Unless this swelling is deemed excessive, however, it should not be cause for rejection.

Spiral wound gaskets with a centering ring are required to be marked with the following information:

- Manufacturer name or trademark
- Flange size (NPS)
- Pressure class
- Winding metal abbreviation, if other than Type 304 stainless steel
- Filler material abbreviation

- Centering and inner ring metal abbreviation, if other than Type 304 stainless steel
- Flange identification, if other than ASME/ANSI B16.5.

An example of the markings for a NPS 12, Class 1500 ASME/ANSI B16.5 gasket having Inconel metal winding, PTFE (Teflon) filler material, and Inconel inner ring is shown below.

12-1500 INC 600-PTFE
INC 600 L.R.
(Manufacturer trademark)

In addition, spiral wound gaskets are color coded.

Gaskets for ring joints are marked on the outer surface with the manufacturer's name or trademark and the gasket number prefixed by the letters R, RX, or BX. Following the gasket number is the material identification code. Table 4-1 lists these identification codes.

Table 4-1 (22)

Ring Joint Markings

Ring Gasket Material	Identification Code	Marking Example (Preceded by Manufacturer's Marking)
Soft Iron	D	R51D
Low Carbon Steel	S	RX51S
4-6 Chrome 1/2 Moly	F5	BX51F5
Type 410 Stainless Steel	S410	R1S410
Type 304 Stainless Steel	S304	RX51S304
Type 316 Stainless Steel	S316	BX51S316
Type 347 Stainless Steel	S347	R51S347

Metal jacketed gaskets are required to be marked with the following:

- Manufacturer's name or trademark
- Flange size (NPS)
- Pressure class
- Jacket material abbreviation excluding soft carbon steel
- Filler material abbreviation

- Flange identification if other than ASME/ANSI B16.5.

If the size of the gasket does not allow sufficient room for the markings, a separate marking tag may be attached.

After verifying the gasket labeling and color coding, the inspector should verify the inside and outside dimensions as well as the dimensions of the centering and inner rings. The actual dimensions of the gasket can be compared to the values given in ASME/ANSI B16.20 or B16.21.

Elastomeric gaskets should be inspected for hardness. The Type A durometer, manufactured by the Shore Instrument Company, is the standard instrument used to measure the hardness of rubber compounds. A durometer's calibrated spring forces an indentor into the test specimen. The instrument's scale indicates the hardness of the rubber in increments of 5. A reading of 100 indicates that no penetration was made and, thus, the rubber is very hard. For hard elastomers (e.g., Type A elastomers with a hardness reading over 90), the Type D durometer may provide more accurate measurements.

Lastly, the general appearance of the gasket or gasket material should be determined with a visual inspection. The gasket surface should be inspected for any mechanical damage. The O-rings should be checked for cuts which may prevent sealing.

4.2.2 Flanges

In addition to gaskets, flange surface finishes should also be inspected in a receipt inspection. ASME/ANSI B16.5 Code lists permissible imperfections in flange facing finishes for raised face and large male and female flanges. Table 4-2 lists the maximum radial projection of surface finish imperfections based on the depth of the imperfection. Surface imperfections must be separated by at least four times the permissible radial projection. Protrusions above the serration are not allowed.

Table 4-2 (22)

Permissible Imperfections in Flange Facing Finishes for Raised Face and Large Male and Female Flanges

NPS	Maximum Radial Projection of Imperfections which are not Deeper than the Bottom of the Serration (in)	Maximum Depth and Radial Projection of Imperfections which are Deeper than the Bottom of the Serration (in)
1/2 - 2 1/2 inches	0.12	0.06
3 inches	0.18	0.06
3 1/2 - 6 inches	0.25	0.12
8 - 14 inches	0.31	0.18
16 inches	0.38	0.18
18 - 24 inches	0.50	0.25

ASME/ANSI B16.47 Code is scheduled to adopt a table similar to Table 4-2 for large diameter flanges. For tongue-and-groove, flat-faced, and small male and female flanges, no Code requirements currently exist. The recommendation for tongue-and-groove and small male and female flanges is that the radial projection of any imperfection be less than 25% of the width of the gasket. For flat faced-flanges, the table provided in ASME/ANSI B16.5 Code may be followed. For ring-joint flanges, no imperfections should extend more than 12.5% up a side wall or across the bottom of the groove. This standard is equivalent to that published in ASME/ANSI B16.5 for raised-face flanges (Table 4-2). Because ring joints have smaller seating areas than do raised-face flanges, the maximum recommended projection of imperfections for ring joints is one-half the value given for ring-joint flanges. One gasket vendor maintains that the tolerance of defects should be more restrictive.

4.3 Frequency of Assembly and/or Inservice Inspections

Inspections should be performed at the following stages in the service life of a joint:

- After assembly of the joint
- During the system pressure test
- When the system first reaches the operating pressure and temperature
- After a system or plant transient
- Just prior to system or plant shutdown.

The initial inspection of the joint should include a check of the bolt torque and the uniformity of the gasket compression by observing and/or measuring the gap between mating flanges. Subsequent inspections need only inspect for signs of leakage.

All personnel should be alert to wet process areas that may indicate a joint is leaking or has leaked. Any signs of current or previous leakage should be reported to maintenance or operations. A maintenance tag marking previously reported leaks will reduce duplicate reporting.

When joints are reported to be leaking, inspect them daily. If the leak rate does not increase, the inspection frequency may be decreased to weekly or biweekly. It may be useful to classify leaks as active or inactive. If the leak is inactive at the time of inspection, examine the operating conditions to determine what changes in the system status have occurred. Schedule subsequent inspections to coincide with the system conditions during which the leak occurred.

4.4 Inspection Criteria

Inspection criteria depend on the inspection being performed. For assembly inspections, the following should be included:

- Verification of proper clearances, including assurance that mating flanges are equally spaced for the full circumference of the joint; assurance that contact is made at all points around the joint for metal-to-metal connections
- Verification of proper gasket type and material, including assurance that the color coding on the edge of the gasket corresponds to the gasket material required
- Verification of bolt torque by observing the craftsmen tightening the bolts.

Pressure tests and inservice inspections are performed to discover leaks. Performing these inspections on uninsulated piping requires a hold time of ten minutes. For tests on insulated piping, a hold time of up to four hours may be required. The ASME Code specifies recommended hold times.

4.4.1 Leak Inspections

When signs of leakage are found, the source of the leak must be determined (22). Although the source is often obvious, sometimes what appears to be a leak is actually condensation dripping from a structure or from piping. For leaks from heat exchangers, note the location of the leak relative to the pass plates. If the leak is at or near a pass plate, the pass plate may be leaking into the outer gasket. For piping on which condensation forms, insulation is recommended. Although it does not contribute to leakage from the system, condensation represents a loss in cooling capacity of the water. It results in added corrosion to external components, increased cleaning costs, and increased operating costs.

After the source of the leak has been identified, examine the defective joint to determine the leakage rate. This information should be recorded for comparison with leakage rates in subsequent inspections to determine leakage growth rate. For dripping joints, the leakage rate is found by counting the drops leaked per minute or the time between drops. For steam leaks, however, the evaluation of leakage rates is qualitative. One person should perform all inspections of a given steam leak to determine whether the leak is increasing. If it is, steam may be cutting the flange surface. In this case, additional repair will be required when the joint is disassembled. If a leak is intermittent, the time of the leak should be noted and compared with system operating conditions. Pressure spikes during system start up and shut down or thermal cycling of the piping can cause joints to leak.

Before repairing a leaking joint, the effects of the leak should be considered. If a leak is not major and does not cause adverse effects such as corrosion or contamination, it is not mandatory to repair the leak immediately. However, all leaks have economic consequences. Pumping costs, loss of system efficiency, and cleaning costs add to the expense of a leak. Other factors which determine the urgency of repairing a leak include technical specification limits, administrative limits, and management and regulatory rules.

An examination of a leaking joint should be performed prior to disassembly to determine the cause of the leak. Disassembling the joint destroys some information that would otherwise guide the evaluator in detecting the cause of a leak. Items to be evaluated prior to disassembly include bolt torque, flange alignment, and gasket condition.

- Bolt torque is verified by retorquing each bolt to the original torque value and in the original torquing pattern. Note the amount that each bolt rotates. If any bolt turns more than one sixteenth of a revolution more than the others, the original torquing method is questionable. If all of the bolts require more than one half a revolution, the gasket or bolts may have relaxed. After tightening to the original torque value, the bolts may be torqued to the higher torque values shown in the torque tables given in Section 3, Section 3.2.7.

  Datum rods or ultrasonic devices measure the reduction in bolt length in the joint as the bolts are loosened. The joints are loosened fully, one at a time, then are carefully retightened to the original preload.

  If a random sampling of bolts in a joint suggests that the bolts are preloaded correctly, the there is no need to test the remaining bolts. If problems are apparent, additional bolts should be tested and/or all the bolts should be retightened to the desired preload in a cross bolting pattern. Both odd and even numbered bolts should be included in the sample.

- Flange alignment is determined by measuring the flange adjacent to bolts and at the midpoint between bolts. This measurement will indicate uneven or excessive gaps or flange bowing.

- The gasket is evaluated for signs of blow out and extrusion due to excessive gasket load. The location of the leak is also examined. For spiral wound gaskets, the gasket windings should not extend past the raised face portion of the flange.

The joint is not the sole focus of a leak inspection conducted prior to disassembly. The pipe loading near the joint should be evaluated as well. Pipe supports adjacent to the joint should be inspected for signs of binding. Even if no signs are found, a review of the piping stress analysis should be performed to ensure that the joints are not carrying excessive bending. In addition, several supports on either side of the joint should be examined for deformed members and galling or rubbing at the pipe support interface. The length of the pipe should be inspected to ensure that it has not contacted other structural members and created an unintentional support. A stress engineer can use the information gathered from these inspections to determine if unacceptable loading in the piping has been introduced.

The bolting should also be inspected when gasket compression occurs. Galling of the bolt threads, caused by insufficient or incorrect lubricant, may cause incorrect torque to be applied to the bolt during installation. Bolts that have an oxide coating were damaged during bolt up. Those that appear to be new at the galled locations were damaged during disassembly. In addition, the nuts should be run up on the studs to determine if any elongation of the bolts has occurred. If the nuts do not turn freely on the studs, bolt yielding may have occurred.

4.4.2 Inspections due to Questionable Bolt Quality

During assembly and subsequent operation, questions can and have arisen regarding material pedigree and physical condition. For some alloys (e.g., precipitation hardening alloys such as 17-4 PH) material property changes due to secondary aging effects can occur. In addition, degradations due to wastage or general corrosion and stress corrosion cracking are well characterized conditions meriting attention.

When the quality of the bolts in the joint is questionable, the following may be performed:

- A visual inspection for cracks, rust, damaged threads, etc.
- A hardness test to estimate tensile strength of the bolts. This test verifies that the bolts are made with the correct material.
- An ultrasonic test for possible cracks or wastage. Techniques for performing this test are developed and are described by S.N. Liu (23).

5.0 Troubleshooting

5.1 Troubleshooting Leaks

5.1.1 Introduction

The principal cause of bolted joint leaks is low fastener load resulting in insufficient gasket seating stress. Low fastener load can result from:

- A preload specified too low. A low specified preload can result if the fastener allowable stress is used to determine the assembly load without regard for other factors. See Section 2.5 for a discussion of assembly loads.
- Interaction effects during assembly which can result in preload scatter and a residual load across the joint approximately 50% of the design load.
- A flange design load that is converted to a torque value for assembly.
- Incorrect torque. The assumptions made concerning the torque to load relationship do not always adequately represent field assembly conditions. This results in assembly loads lower than desired levels.
- Relaxation of fastener gasket and flange stress which takes place throughout the life of the joint.
- Pressure thermal cycles.

Other causes of leaks include broken or corroded bolts, incorrect or damaged gaskets, etc. Detection and correction of these problems are covered elsewhere.

5.2 Leak Resolution

Leak resolution is divided into three steps: review of assembly procedures, review of preload requirements, and review of miscellaneous factors.

5.2.1 Review Assembly Procedures

The first step in resolving a leak is to review the assembly procedures for the joint. The procedures should be evaluated with the recommendations in Section 3. The following items should be addressed in the review:

- Determine the number of passes used to assemble the joint. One utility recommends remaking a leaking joint with a minimum of six to eight passes.
- Determine whether design preload is being achieved. Review the torque/preload relationship and elastic interactions discussed in Chapter 3. Changing torque,

torquing patterns, or developing an improved preload control method may be required.

5.2.2 Review Preload Requirements

If the assembly procedures are found to be satisfactory, review the preload specifications. Address the following items in the review:

- Determine if the specified preload is too high or too low. Appropriate preload levels are discussed in Chapter 2, Section 2.5.
- The assumptions made when the design preload was converted to an assembly value (i.e. torque). Examples of these assumptions include lubricant and nut factor accuracy, efficiency of tools and procedures, and relaxation of preload.

After considering these two aspects of preload, many utilities find that increasing the assembly torque is effective in eliminating the leak. Some companies develop three level torque tables (refer to Section 3.2.7). The three levels included in this torque table are the manufacturer's standard assembly torque, a higher level torque used routinely by maintenance, and a maximum level torque used on leaking joints on a discretionary basis by maintenance. Chronic leakers should be referred to the responsible engineering organization for additional consideration including a root cause investigation to determine corrective action.

5.2.3 Review of Miscellaneous Factors

5.2.3.1 Relaxation Effects

Following the review of preload specifications, address relaxation effects. Relaxation effects can cause leaks by causing a loss of preload in a previously tightened joint. To compensate for relaxation in the joint during service life, the preloaded bolt must have sufficient stretch. If the assembly preload produces a fastener stretch of less than 0.004 inch, installation of spacers or Belleville springs should be considered to increase the stretch of the fastener (refer to Figure 5-1).

Time, stress, cyclic loads, and temperature cause the components of a joint to relax. The gasket is usually the component most adversely affected by relaxation. Because it is subjected to thermal cycles, it is not unusual for the gasket to permanently set 0.001 to 0.003 inches. As relaxation progresses, the clamping load or sealing load across the joint is lost. When the clamping load degrades sufficiently, a leak occurs.

Roberts (24) reported a 0.001 inch set can occur in compressed asbestos gaskets subjected to 10,000 psi stress for 20 minutes at room temperature. A permanent gasket set of 0.001 inch will result in a loss of bolt stretch of 0.001 inch; for short fasteners with low preload, the loss of 0.001 inch stretch could result in a 25% loss in preload.

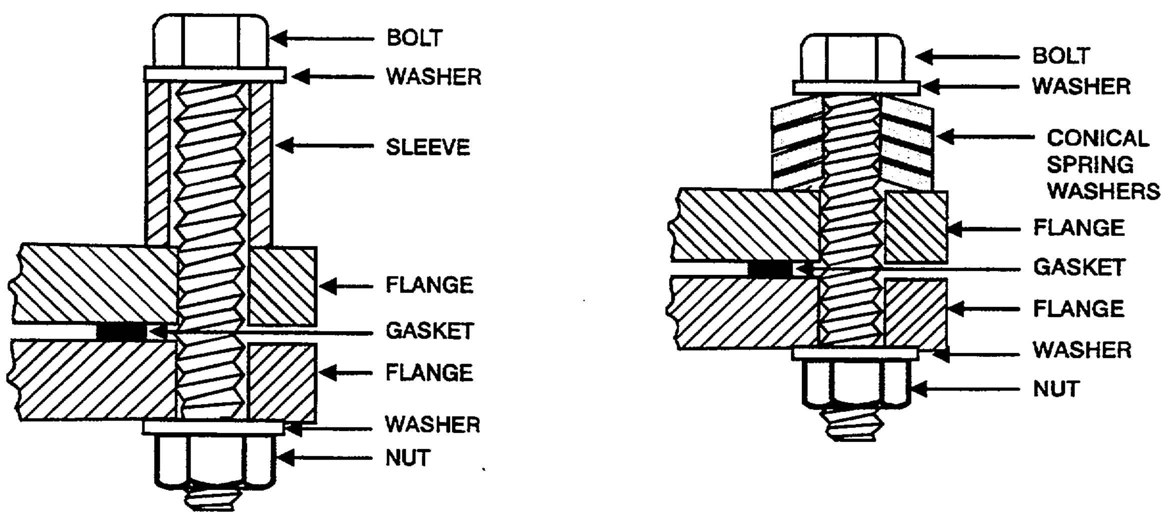


Figure 5-1 — Application of Spacers on Live Loaded Gasketed Joints.

Oak Ridge National Laboratory (25) studied the sealing of 1- to 2-inch diameter ring joints under cyclic thermal loads. The flanges of the joints were stainless steel with carbon steel bolts. Figure 5-2 shows the joint configuration.

In the study, installed spacers allowed the use of longer, more flexible bolts. This provides more initial preload stretch and reduces the thermal stress caused by the differing coefficients of thermal expansion. Thermal cycles affect both the bolt and ring stress. Figure 5-3 illustrates the effect of thermal cycles on bolt load. The figure indicates a decrease in stress with increasing load cycle. Bolt load becomes asymptotic (i.e., there is no further reduction in stress) at a bolt stress of 30,000 psi after 10 cycles. The conclusions of the study are:

- Spacers are required to reduce the effect of thermal stresses on bolt and gasket stress.
- The initial stud preload is 45,000 psi, approximately twice the allowable stress for the bolt material.
- Tight tolerance on ring and ring groove dimensions, surface finish, and hardness are required.

5.2.3.2 Joint Design

Following the review of relaxation effects, review the joint design. Analyze the type of gasket, bolt loads, thermal effects, choice of materials, and flange rotation. If the design is faulty, the problem may be corrected by altering the design, adding spacers or Belleville springs, changing materials, or changing the preload.

Many large diameter flanges (greater than 24 inches in diameter), designed to the ASME Boiler and Pressure Vessel Code, will exhibit large flange rotation. The design is based on allowable stress and does not limit flange rotation.

Flange rotation is important to the sealing performance of a flanged joint. Excessive flange rotation will reduce the sealing performance. Flange rotation is caused by the preload, by internal pressure, or by thermal stresses. Flange rotation causes the gasket to unload on the inside diameter while the compressive stress on the outside diameter increases. Flange rotation also results in a loss of fastener preload (refer to Figure 2-10).

Rodabaugh and Moore (7) reported a loss of preload of approximately 20% after pressurization of a 25-inch diameter vessel. Andreosso (13) reported a 20% loss in preload after pressurization of a pressurizer manway. In each of these cases, flange rotation caused the reduction in preload. Increasing the preload, increasing the confidence in the preload, or redesigning the flanges to reduce the rotation alleviates loss of preload.

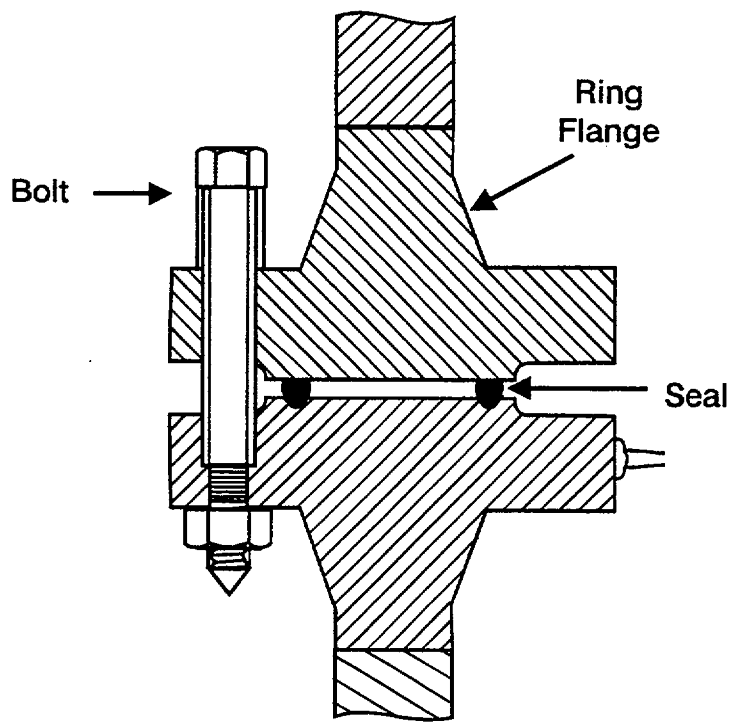


Figure 5-2 — Oak Ridge HRE-2 Ring Joint.

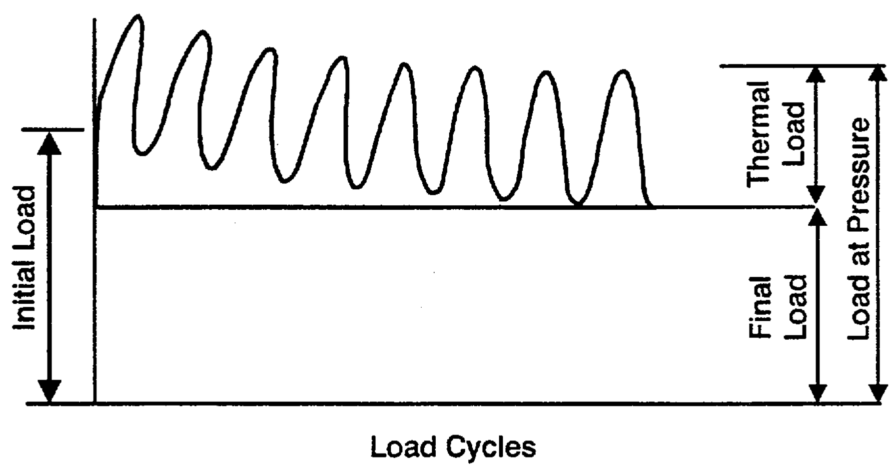


Figure 5-3 — Thermal Cycle Effect on HRE-2 Ring Joint.

5.3 Leak Minimization

The following sections suggest practices and products to minimize leaks.

5.3.1 Live Loading

Valve packages are sealed by manually adjusting the load on the packing by tightening the packing nut. Retightening is performed continually and compensates for wear and relaxation of the packing material. The use of Belleville springs to "live load" a valve packing alleviates the need for continual retightening. The stored energy and deflection in the spring stack are sufficient to provide a relatively constant sealing load to the relaxing packing.

Recently, Belleville springs have been shown to be effective in alleviating leaks in bolted gasketed joints. Figure 5-1 illustrates the application of Belleville washers and spacers in a flanged joint. These devices are useful:

- When the fastener preload results in low stretch (less than 0.004 inch) due to a small grip length or low preload.
- When the joint is subjected to cyclic thermal loads which cause permanent set in the gasket and a loss of preload.

5.3.2 Hot Torquing

The practice of retorquing fasteners at temperatures above the ambient assembly temperature is called hot torquing or hot bolting. The fasteners are retorqued as a leak preventative measure once the joint is brought to operating temperature and pressure. This practice reestablishes preload before a leak starts. Hot bolting is ineffective in sealing a leak once it has begun.

ASME Section VIII, Division 2, Article 3.5 discusses retightening of bolts. The article states that preloads may require renewing because of loss of load caused by operating conditions. The Code makes no mention of how to perform retightening or when it is permissible.

The Code also cautions about "stress ratchetting". If bolt loosening is caused by yielding of the fastener or flange, care must be taken in retightening the bolt. Repeated retightening may ultimately make the joint unserviceable.

Hot bolting may be performed using one of the following procedures:

- Retighten the fasteners while the joint is hot but without pressure. Some utility companies follow this process for steam generator manways. After the hot functional test, the manway studs are retorqued.

- Torque the fasteners while the joint is hot and pressurized. This is a dangerous practice and the consequences should be considered before use. This procedure has led to stud failure and increased leakage caused by hot bolting.

The question of how much to retorque the joint should be considered. Temperature may affect lubricants, lubricity of the joint, and the nut factor. Information on these effects may be obtained from the lubricant supplier or derived experimentally. Reapplication of the original preload should reestablish the original clamping force. Documentation of hot torquing procedures should be maintained in order to establish an experience base for future applications.

5.3.3 Sealants

A chronically leaking joint may be repaired with a sealant. Sealants are effective but should be used only until a long term solution is found (refer to EPRI Report NP-6523, On-Line Leak Sealing for guidance on sealing leaks).

5.3.4 Techniques for Minimizing Leaks

Step 1 *"Initial Procedures"*

Establish Initial Bolt Preload Stress
- For components — follow vendor recommendations
- For standard ANSI flanges — preload stress ≈ 50 ksi
- For Class 1 components — follow code requirements

Establish Basic Assembly Procedure
- Joint assembly procedure
- Bolt lubrication & procedure
- Bolt torquing procedure

↓

Step 2 *"Closer Examination"*

If Joint Leaks, Perform Engineering Review
- Measure torque during disassembly to check for preload loss
- Inspect joint during disassembly
- Visually inspect flange surfaces & alignment
- Inspect gasket for evidence of metal-to-metal contact or crushing
- Review joint application based on past experience
- Increase bolt preload stress to maximum using torque wrenches
- Measure uniformity of joint pull up during tensioning

↓

Step 3 *"Technical Assessment"*

If Joint Continues to Leak, Perform Engineering Assessment
- Quantify "cold spring" during joint disassembly
- Contact vendor for constructive suggestions
- Increase bolt preload to maximum permitted in joint
- Measure the actual preload produced in bolts (UT etc.)
- Consider using collars/springs to increase joint compliance
- Consider using hydraulic tensioners for more uniform tensioning

↓

Step 4 *"Engineering Analysis"*

If Joint Continues to Leak, Perform Engineering Analysis
- Flange rotation or distortion
- Bolt yielding
- Gasket crushing
- Effect of transient loading conditions
- Embedment
- Stress relaxation
- Gasket design/fabrication

Figure 5-4 — Suggested Four-step Approach to Bolted Joint Leak Reduction

5.4 Failure Evaluation

5.4.1 Flanged Joints

Table 5-1 is a guide for evaluating failure in flanged joints. The table lists failure modes, causes, and the recommended corrective actions.

Table 5-1

Failure Evaluation for Flanged Joints

Failure Mode	Cause	Corrective Actions
Gasket crush all around	• Over torque • Gasket is too soft; low modulus of elasticity • Gasket is too thick	• Install a new gasket
Gasket crush on one end only	• Excessive pipe moments • Pipe misalignment • Incorrect bolt up procedure	• Additional passes
Gasket crush on outer edge	• Flange rotation • Over torque • Gasket is too hard; high modulus of elasticity • Flange pressure rating is too low	• Crush control • Stretch control • Low stress gasket
Gasket disintegration	• Incorrect gasket material • Contamination of the flange and/or the gasket	Change gasket material
Localized corrosion	• Localized contamination of the flange and/or the gasket • Damaged surface	Renew sealing surface
Generalized corrosion	• Generalized contamination of the flange and/or the gasket • Incorrect gasket material	Change gasket material
No gasket compression (not seated for full circumference)	• Under torque • Gasket is too hard; high modulus of elasticity • Gasket is too thin	• Increase torque • Measure fastener strength • Softer gasket
Loose bolts on one end only	• Pipe misalignment • Excessive pipe moments • Incorrect bolt up procedure	• Align the pipe • Additional passes

Table 5-1

Failure Evaluation for Flanged Joints

Failure Mode	Cause	Corrective Actions
All bolts loose	• Bolt relaxation; bolt creep • Gasket relaxation; gasket creep • Under torque	• Reestablish the load • Install spacers or Belleville springs • Metal-to-metal contact with controlled gasket deflection
Random bolts loose	• Incorrect bolt up procedure • Bolt relaxation, if some low yield bolts have been mixed with the correct bolting	• Additional passes • Vary the stretch patterns
Gasket cut	• Installation error in the tools used to insert the gasket; mishandling the gasket during installation • Damaged gasket was not inspected at installation	Provide gasket handling instructions and fixtures
Flange surface scarred	• Installation error in the tools used to insert or remove the gasket • Damaged flange was not inspected during make up	Renew sealing surface
Gasket extrusion	• Gasket is too soft; low modulus of elasticity • Incorrect gasket material • Over torque • Gasket is too thick	• Stretch control • Gasket control
Gasket blowout	• Flange surface is too smooth • Incorrect gasket material • Gasket is too thick	• Machine
Bolts elongated	• Over torque • Incorrect bolting material • Gasket is too hard; high modulus of elasticity	• Revise assembly procedure • New gasket

Table 5-1		
Failure Evaluation for Flanged Joints		
Failure Mode	Cause	Corrective Actions
Flange bowed	• Over torque • Flange pressure rating is too low for service conditions • Flange material is not adequate for service conditions	Revise assembly procedure
Flange rotation	• Over torque • Flange pressure rating is too low for service conditions	• Revise assembly procedure • New flange

5.4.2 Other Joints

The above discussion centers on flanged joints but may be applied to most other equipment. Exceptions include O-ring joints. Both of these joints can suffer failures unique to their design.

The Parker O-Ring Handbook and SAE publication AIR1707, Patterns of O-Ring Failure (7), provide information on common O-ring failure modes. Table 5-2 is derived from these sources.

Table 5-2		
Failure Evaluation for O-Rings		
Failure Mode	**Cause**	**Corrective Actions**
• Extrusion or nibbling edges of the O-ring on the low pressure or downstream side of the gland appear chewed or chipped	• Excessive clearances • Pressure in excess of the system design or high pressure excursions • O-ring is too soft • Degradation of the O-ring material by system fluid (e.g. swelling, softening, shrinking, cracking, etc.) • Irregular clearance gaps due to excessive system pressure • Increase in clearance gaps due to excessive system pressure • Improper machining of the O-ring gland (sharp edges) • O-ring is too large, causing excessive filling of the groove	• Decrease the clearance by reducing the machining tolerances • Use a backup device • Check the O-ring compatibility with the system fluid • Increase the rigidity of the metal components • Replace with a harder O-ring • Break the sharp edges of the gland to a minimum radius of 0.002 inch • Ensure the proper size O-ring is installed
• Compression set O-ring is permanently deformed in a flat sided oval	• O-ring material has inherently poor compression set properties • Improper gland design • Excessive temperature causes the O-ring to harden and lose elastic properties • Volume swell of the O-ring due to system fluids • Excessive squeeze due to overtightening of the adjustable glands • Incomplete vulcanization during production of the O-ring • Introduction of fluid incompatible with the O-ring material	• Use a low set O-ring material whenever possible • Select an O-ring compatible with the intended service conditions • Reduce the system operating temperature • Inspect the incoming O-ring shipments for correct physical properties

Table 5-2

Failure Evaluation for O-Rings

Failure Mode	Cause	Corrective Actions
• Explosive decompression Small pits or blisters on the surface of the O-ring	• High pressure gas trapped within the internal structure of the O-ring. During rapid decreases in system pressure, the gas expands, causing blisters or ruptures on the seal surface.	• Increase the decompression time to allow trapped gas to work out of the material • Choose a seal material with good permeability characteristics
• Installation damage Skiving of the O-ring surface due to cutting by metal components; damage on the surface away from the bottom of the O-ring groove	• Sharp corners on mating metal parts, such as the O-ring gland or threads, over which the O-ring must pass • Insufficient lead-in chamfer • Blind grooves in the multiport valves • Oversized O-ring • Undersized O-ring • O-ring is twisted or pinched during installation • O-ring is not properly lubricated before installation • O-ring is dirty on installation • O-ring gland or other surface over which the O-ring must pass during assembly is contaminated with particles	• Break all sharp edges on the metal components • Provide a 20° lead-in chamfer. • Check all components for cleanliness before installation • Tape threads the O-ring will pass • Use an approved O-ring lubricant • Check the O-ring to ensure correct size and material • Be careful during installation not to damage the O-ring surfaces.

6.0 Design

6.1 General Discussion of Mechanical Joints

Joints are designated mechanical joints by the following common features:

- The joint interface is metal-to-metal; no gasket exists.
- The external load carried by the joint tends to be cyclic.
- The joints and fasteners are designed and analyzed for fatigue.
- The direction of external loads causes axial, shear, and bending loads on the fasteners.

Four types of mechanical or machine joints are illustrated in Figures 6-1 through 6-4.

6.2 Background History of Mechanical Joint Design

A bolted joint poses the following design challenges (6, 26 through 29):

- Sizing of the joint components to carry the design loads for the life of the assembly.
- Choosing the fastener configuration, size, and grade.
- Specifying the assembly method such as torque, turn of nut, etc.

Throughout the nineteenth century, the bolted joints were designed using a material stress method. This approach sized the components, in particular the fastener, according to the following equation:

$$\left(\frac{WorkingLoad}{FastenerCrossSectionalArea}\right) \times SafetyFactor \le FastenerStrength$$

In 1870, Wohler discovered that the strength of fastener parts was reduced when subjected to alternating loads. In 1900, Von Camerer introduced a concept for the analysis of bolted joints. His theory explained the manner in which bolted joints carried external loads. Von Camerer discovered that as external load is applied to a preloaded joint, the load is carried by the "joint system", including the fastener and joint. Not all of the external load is carried by the fastener. There is an increase in bolt load and a decrease in compressive load on the joint interface. These discoveries marked the beginning of the use of joint diagrams to analyze bolted connections.

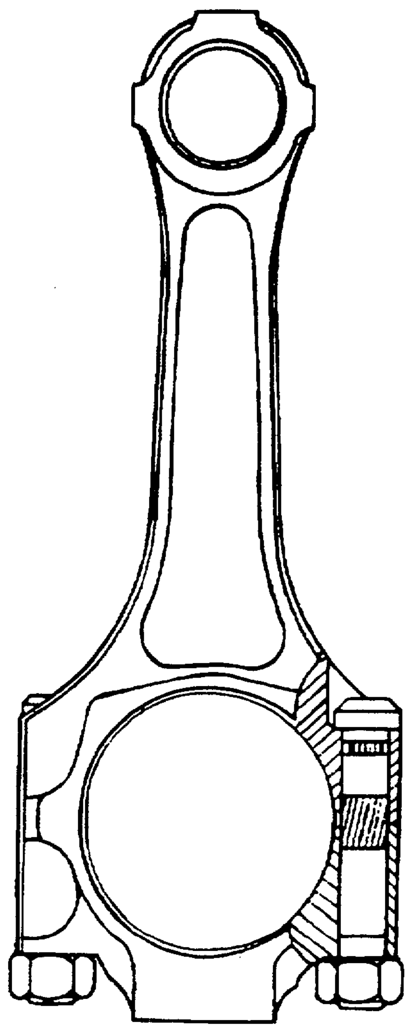


Figure 6-1 — Mechanical Joint 1.

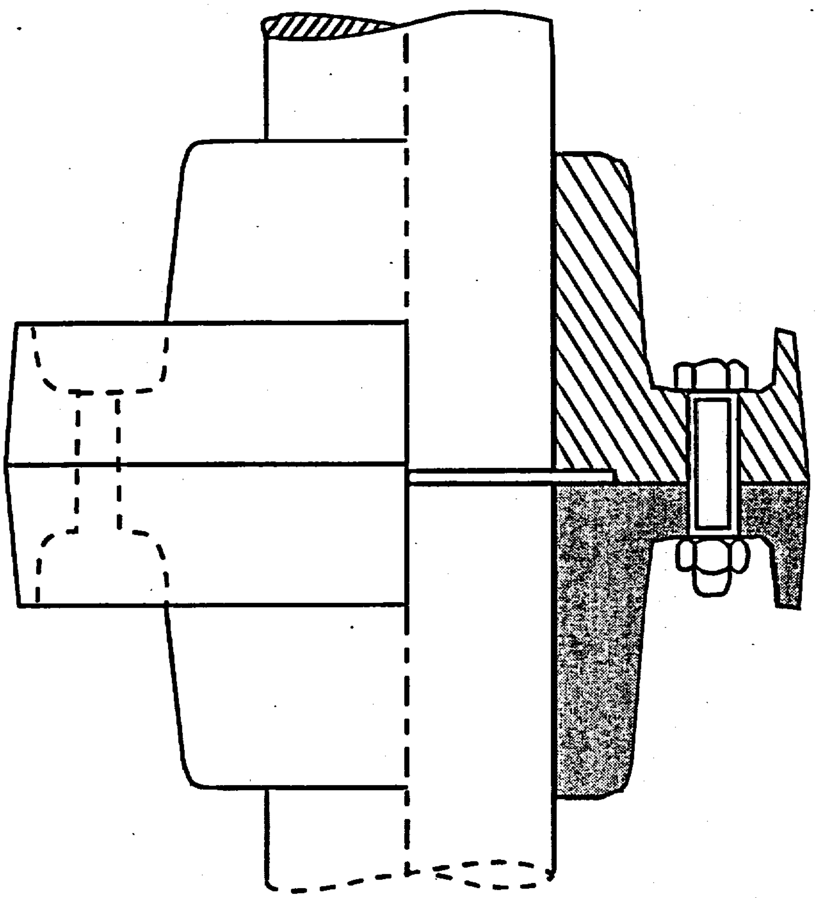


Figure 6-2 — Mechanical Joint 2.

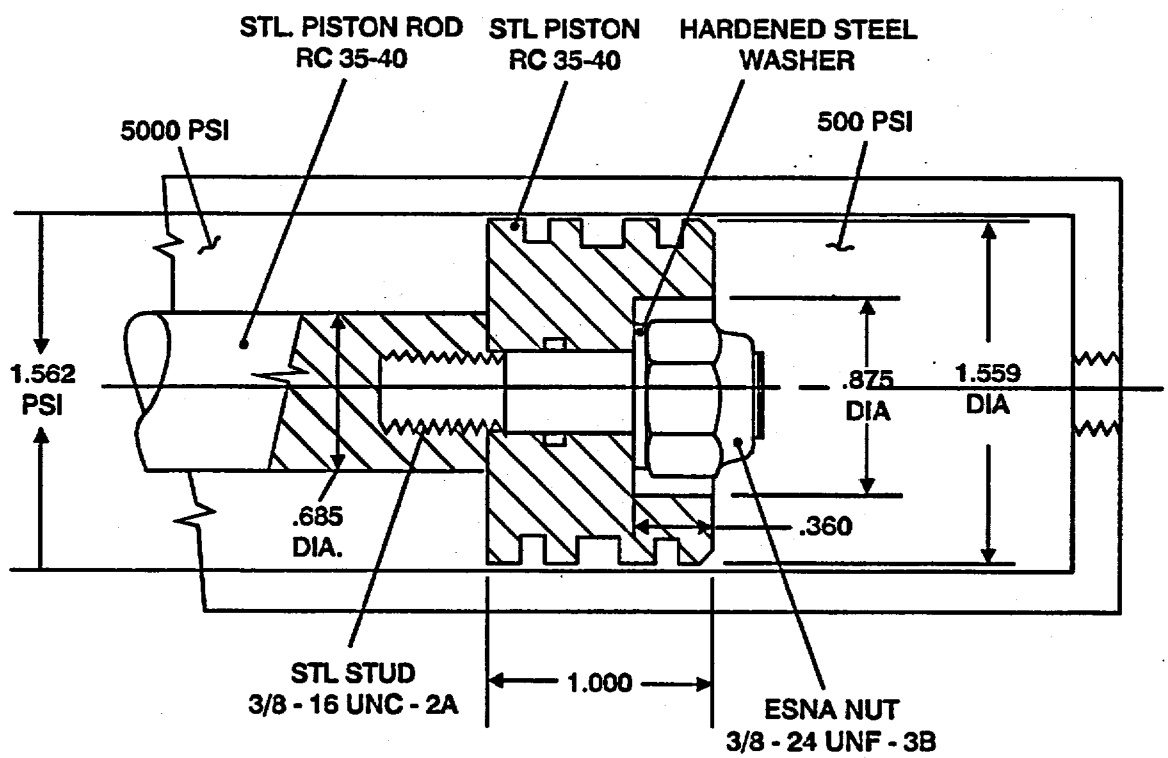


Figure 6-3 — Mechanical Joint 3.

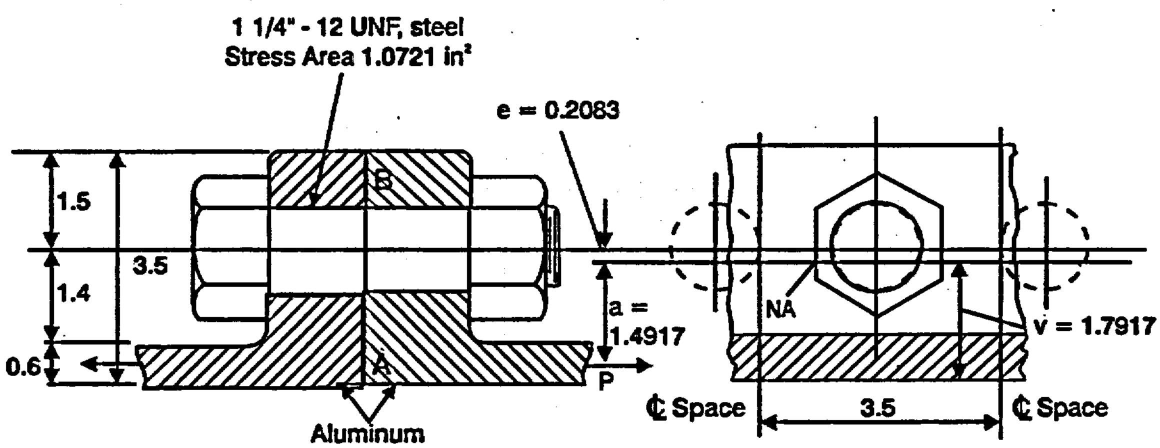


Figure 6-4 — Mechanical Joint 4

6.2.1 Joint Diagrams

In a joint diagram, the external loads on a joint are separated into partial forces. These partial forces include:

- Additional load on the fastener which adds to the initial preload developed at assembly
- Unloading of the joint interface.

Figures 6-5 and 6-6 are examples of joint diagrams. Figure 6-5 illustrates how a preloaded joint distributes the applied external load (F_A). The application of the external load (F_A) causes an increase in bolt load (F_{SA}) and a decrease in compressive load in the joint (F_{PA}). The magnitude of these two changes in load are proportional to the relative stiffness of the fastener and the joint members. Figure 6-6 illustrates the effect of a cyclic external load on the joint. Joint diagrams are discussed in detail in section 6.2.6 of this chapter.

6.2.2 Design Objectives

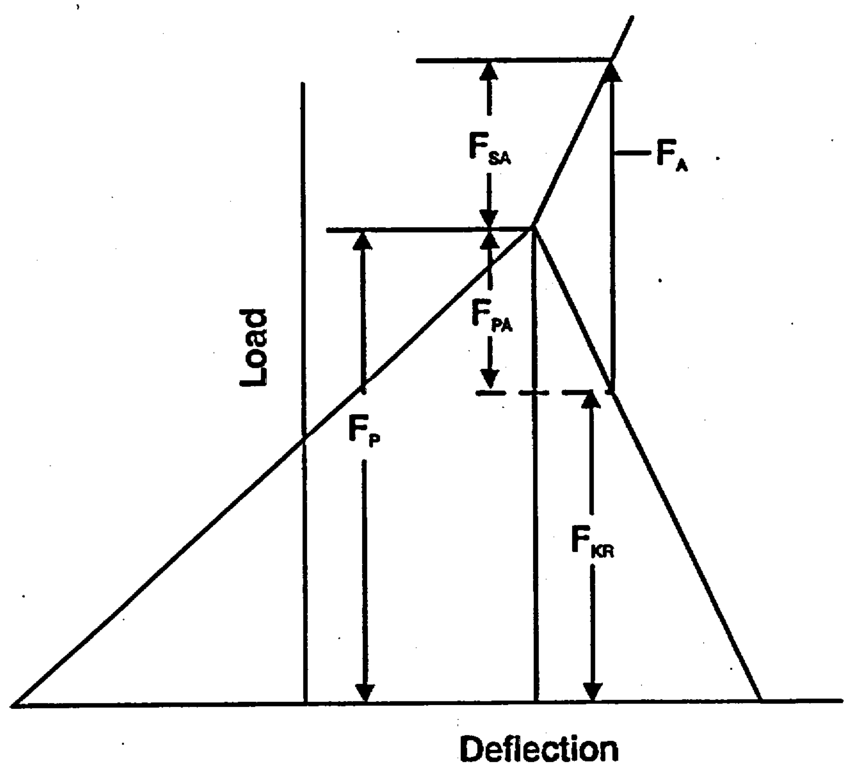
Joint Compression

The principal objective in designing a bolted joint is to ensure that the mating surface of the joint members remains in compression when subjected to external loads. Maintaining this compression is one criterion for judging the suitability of a design.

Figure 6-7 gives a cross section of a preloaded joint element loaded by an eccentric tensile load (F_A). The preloaded bolt produces a uniform compressive stress on the joint interface, see Figure 6-7a. The eccentric tensile load reduces the joint compressive stress and causes the joint section to bend. This distorts the stress distribution as shown in Figure 6-7b. If the compressive stress on the right side of the cross section at point "0" goes to zero, one sided lift off or "gaping" of the joint is said to have occurred.

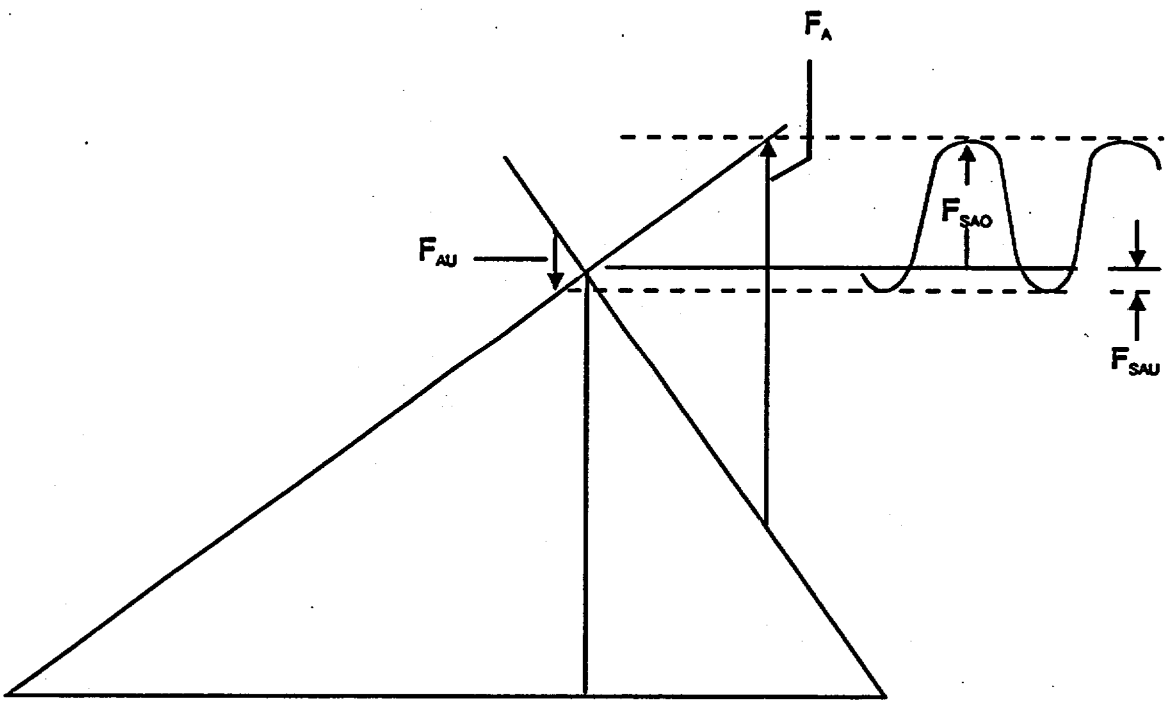
Maintaining the joint mating surfaces in a state of compression improves the joint performance in the following ways:

- The additional fastener loads caused by external loads are minimized.
- The fastener fatigue life is improved.
- Fretting and wear of the mating surfaces is reduced.
- Self loosening of the bolt is prevented.
- The joint is able to carry transverse shear loads.
- The bending stress in the fasteners is minimized.



F_A = External Tensile Load
F_{PA} = Decrease in Compressive Load in Joint
F_{KR} = Residual Compressive Load in Joint
F_P = Preload
F_{SA} = Increase in Bolt Load

Figure 6-5 — External Load Distribution.



F_A = External Tensile Load (lb.)
F_{AU} = External Compressive Load (lb.)
F_{SAO} = Increase in Bolt Tensile Load (lb.) due to External Load FA
F_{SAU} = Decrease in Bolt Load due to External Compressive Load FAU (lb.)

Figure 6-6 — Cyclic External Load.

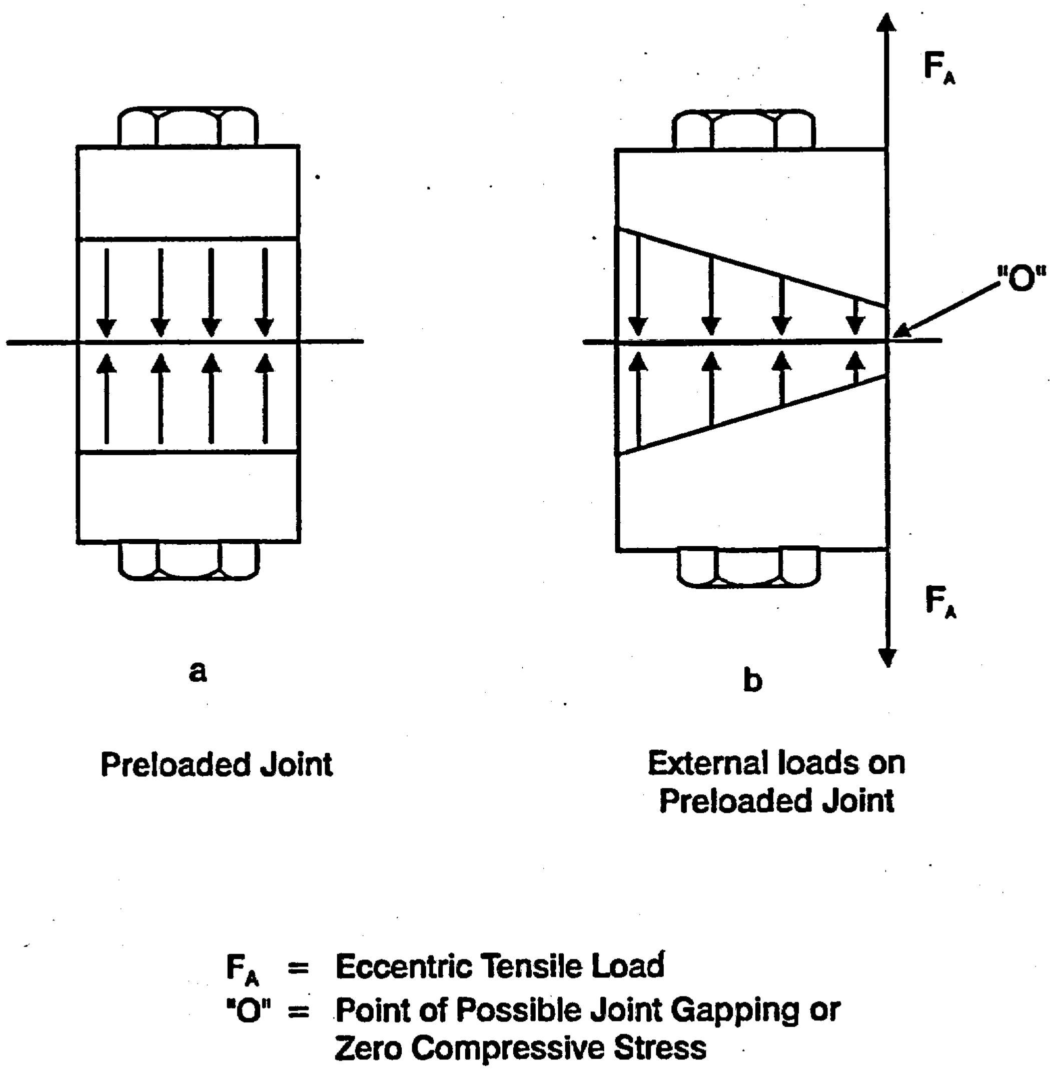


Figure 6-7 — Preloaded Joint Element with Eccentric Tensile Load.

Design Process

The second objective in designing a bolted joint is to ensure that neither the joint members nor the fasteners are over stressed. The design process includes:

- Modeling of the external loads and the joint.
- Calculation of the load required to maintain joint compression.
- Selection of fastener and assembly preloads to maintain joint compression.
- Determination of the safety factor by evaluating the fastener for static and cyclic loads, and evaluating the joint components for compressive stress and thread stripping.

6.2.3 Modeling Working Loads

Figure 6-8 illustrates the three possible load conditions for a mechanical joint. The most general load case, Figure 6-8a, has the following characteristics:

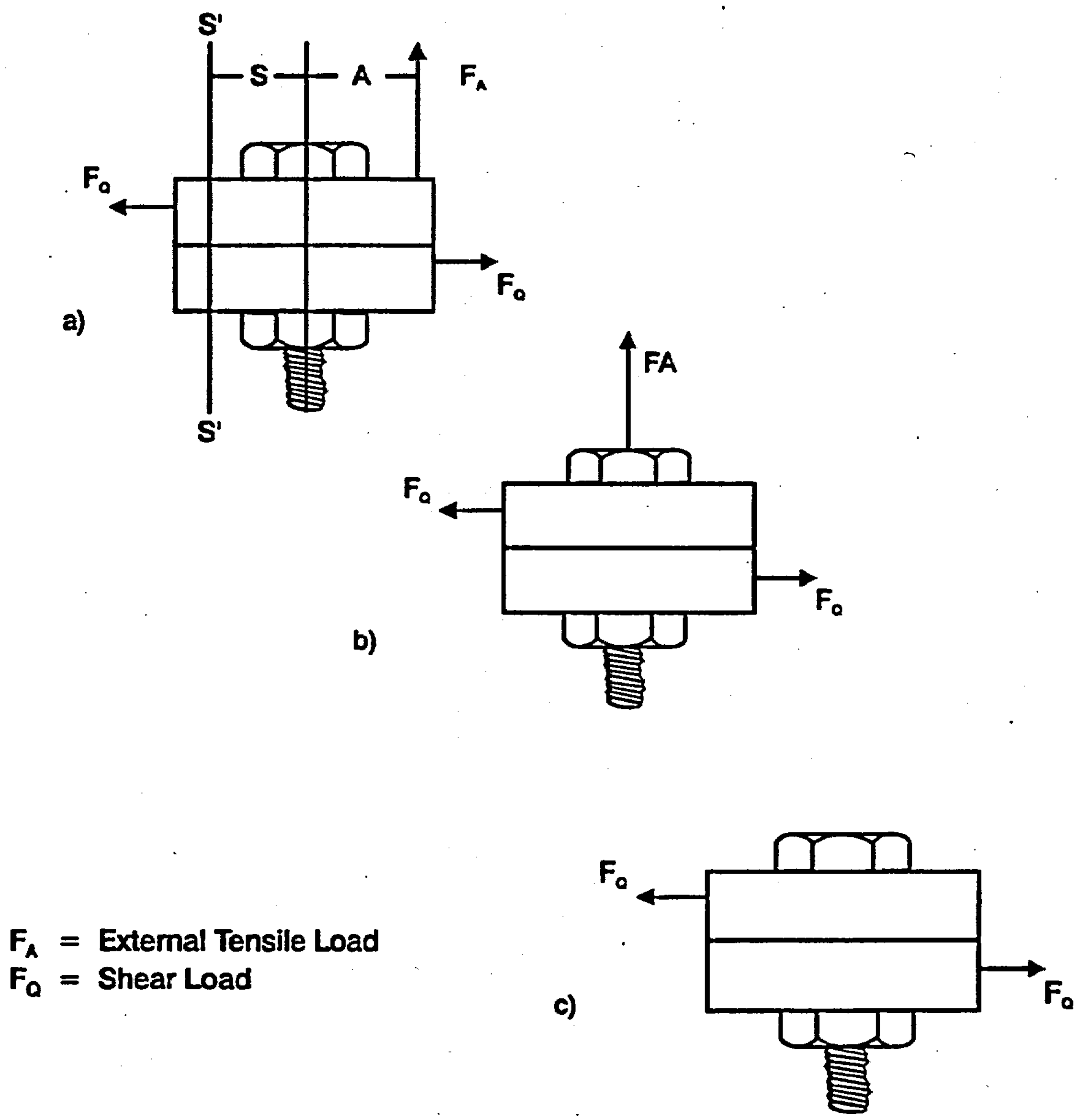
- An eccentric clamp. The bolt is displaced from the center of gravity of the joint by a distance (S).
- An eccentric axial load. The working load (F_A) acts at a distance (A) from the center of gravity. This axial load may be static or cyclic.
- Shear load (F_Q).

Figures 6-8b and 6-8c are other load conditions. The joint in Figure 6-8b is centrally clamped, centrally loaded, and subject to an external shear load. The joint in Figure 6-8c is centrally clamped but is subject to only an external shear load.

6.2.4 Modeling the Joint

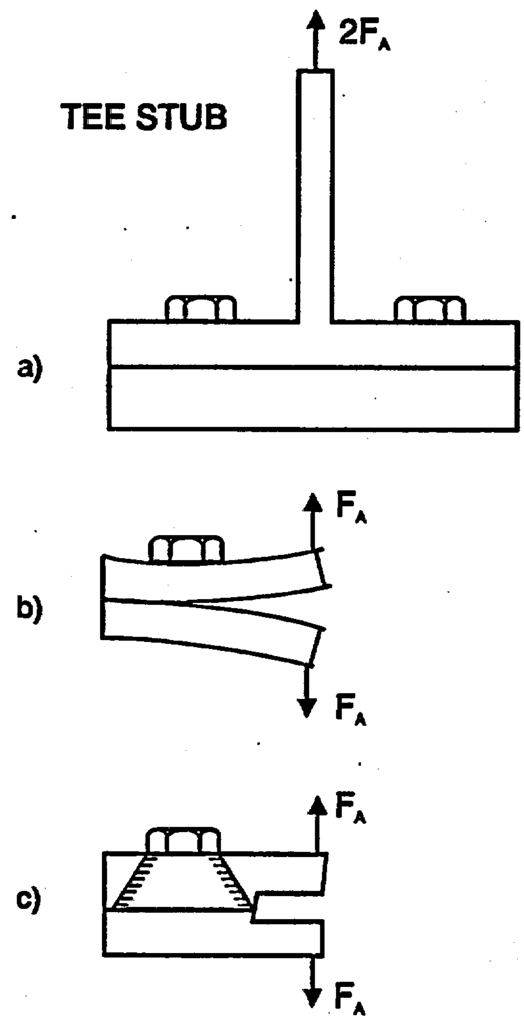
Figure 6-9 illustrates the process of developing a model for the analysis of a typical bolted connection. Figure 6-9a is a tee stub carrying a tensile load. Due to symmetry considerations, the tee stub may be simplified to the single-fastener system shown in Figure 6-9b. The single-fastener model is then further refined to properly represent the joint material which will be affected by the fastener preload and the external load. Figure 6-9c shows the compressive stress distribution in the joint members. The fastener influence on the tee stub is modeled as a truncated cone of limited size. In the model, the material outside the cone is ignored. These outside areas of the mating members do not significantly contribute to the load carrying capacity of the joint.

Figure 6-10 illustrates the model development for an automotive connecting rod. The joint material of the connecting rod is relatively long and slender. Consequently, the flexural rigidity of the assembly is important. The external load is a cyclic tensile load eccentric to the fastener. A variation in the stiffness of the joint from the rod side to the cap side produces a transverse load.



Loading Conditions

Figure 6-8 — Load Conditions for a Mechanical Joint.



F_A = External Tensile Load

Figure 6-9 — Model for Analysis of Bolted Connection.

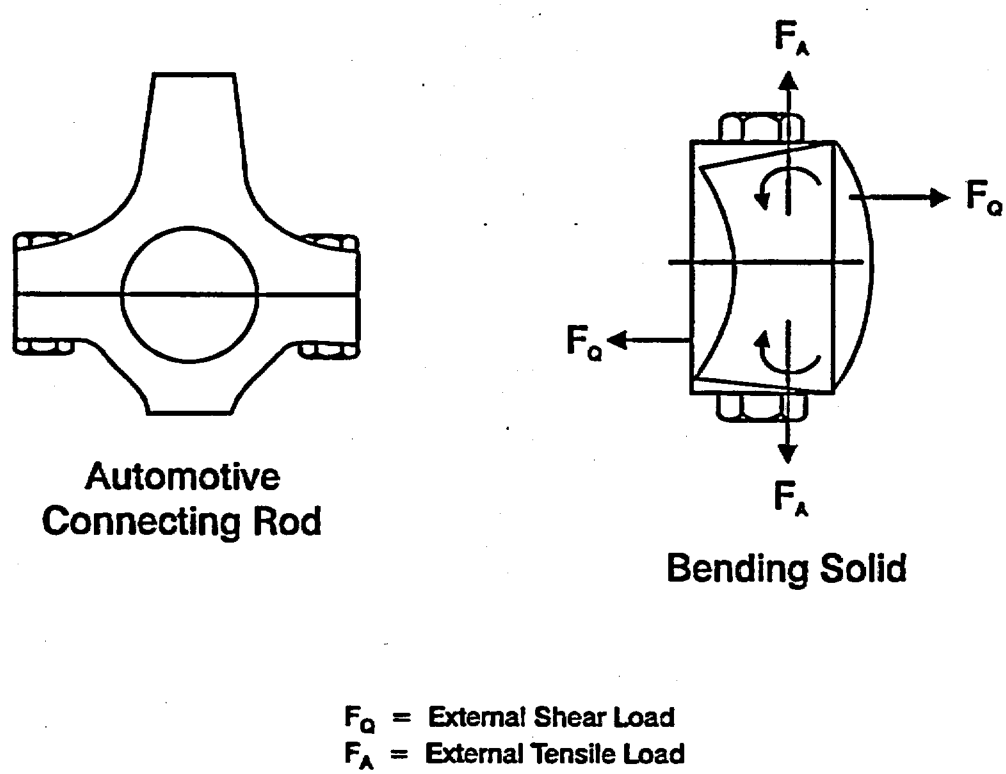


Figure 6-10 — Model for Analysis of Automotive Connecting Rod.

6.2.5 Joint Elasticity and Compressive Stress Field

An important step in the analysis of a machine joint is definition of the compressive stress field in the joint members beneath the bolt. Proper definition of the compressive stress field is important for the following reasons:

- Nonzero compression stress is a design criteria and an important analysis assumption.
- The analysis assumes that the bending solid shown in Figure 6-10 resists the external bending loads. The bending resistance of this solid depends on the contact surface of the two joint members remaining in compression.

The compressive stress field in joint members is a function of the joint geometry shown in Figure 6-11. The stress decreases rapidly with distance from the outside diameter of the bolt head. This stress field can be modeled as a truncated cone as shown in Figure 6-12. Care must be taken in defining the mating surface area of the joint. In Figure 6-12 the outer diameter of the parting surface area for cylindrical joint members is "g". The following relationship must hold:

$$g \leq D_X + L_{min} \tag{6-1}$$

where, D_X = Outside diameter of nut, bolt head, or washer
L_{min} = Minimum thickness of clamped parts

When the joint is rectangular as shown in Figure 6-13, the following conditions must be met:

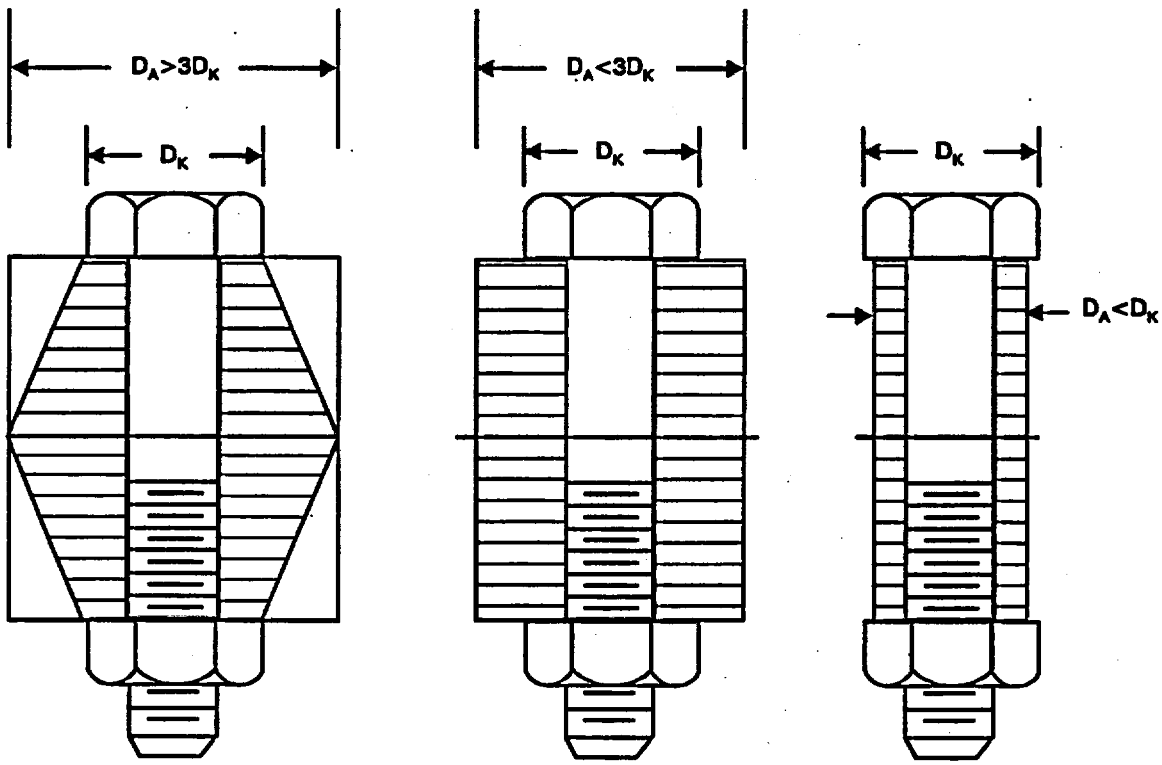
$$U + V \leq g \tag{6-2}$$

$$b < g$$

where, S"- S" = Center of area of the clamped parts with the bolt hole considered
U = Distance from center of area S" - S" to edge of joint away from external load
V = Distance from center of area S" - S" to edge of joint toward external load

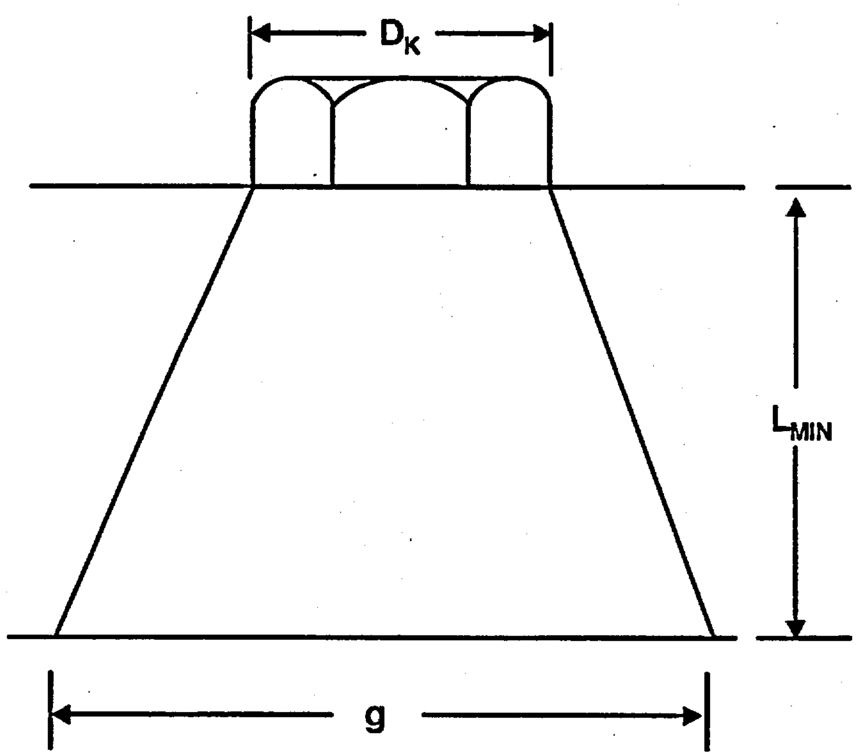
When the single bolt joint model is known, the design is evaluated by considering the following factors:

- Mechanical properties of the fastener
  - Yield strength
  - Fatigue strength
  - Size
- Joint members
  - Compressive strength



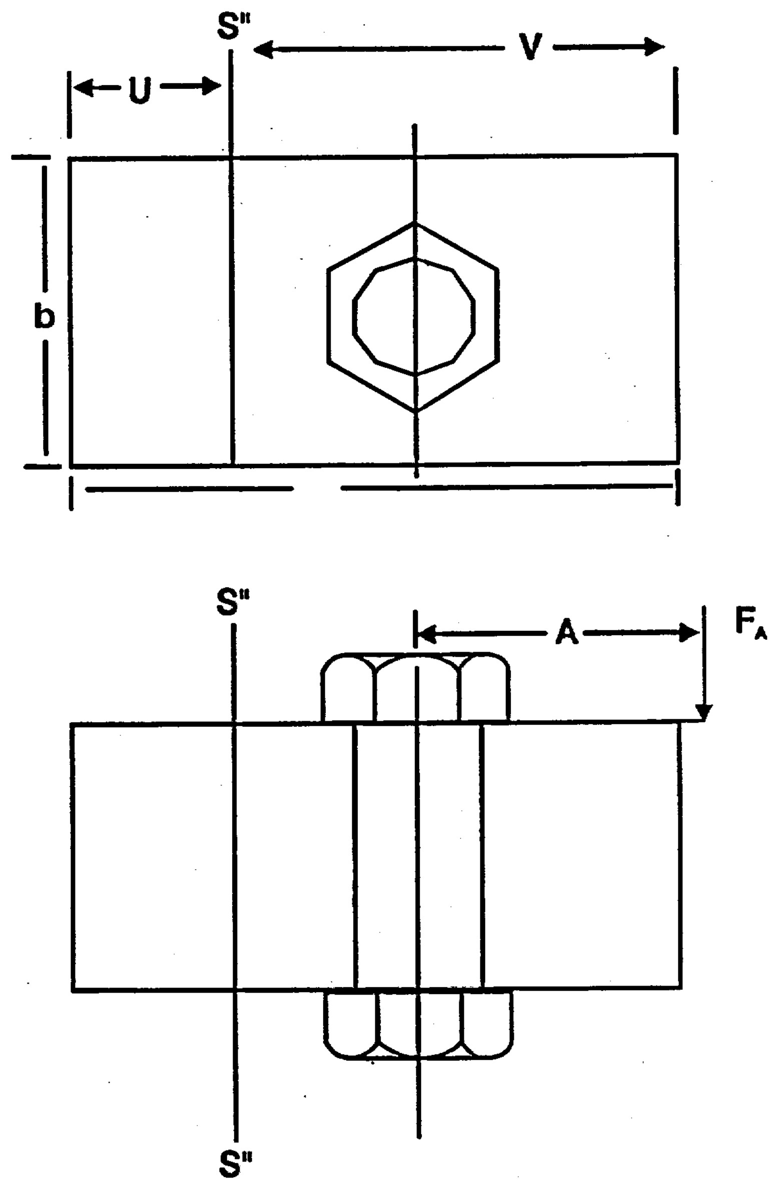
D_K = Outside Diameter of Nut, Bolt Head, or Washer
D_A = Outside Diameter of Joint Members

Figure 6-11 — Compressive Stress Field.



D_K = Outside Diameter of Nut, Bolt Head or Washer
L_{min} = Minimum Thickness of Clamped Parts
g = Outer Diameter of Mating Surface Area

Figure 6-12 — Model of a Compressive Stress Field/Cylindrical.



F_A = Tensile Load
U = Distance from Center of Area S"-S" to Edge of Joint Away from External Load
V = Distance from Center of Area S"-S" to Edge of Joint Toward External Load

Figure 6-13 — Model of a Compressive Stress Field/Rectangular.

  - Thread stripping
  - Volume of the joint material compressed by the fastener preload and unloaded by the working load

- Elasticity of the fastener and the joint
  - Flexibility of the fastener and the joint
  - Bending stiffness of the fastener and the joint

- External working loads
  - Axial
  - Shear
  - Bending
  - Cyclic
  - Combined loads of axial, shear, and bending
  - Loading plane: point of the introduction of the working load

- Assembly
  - Preload: assembly method, torque, friction, and elongation
  - Uncertainty of the preload
  - Combined torsion and tensile stress

- Relaxation
  - Reduction in the preload due to static and dynamic conditions.

6.2.6 Calculation Method

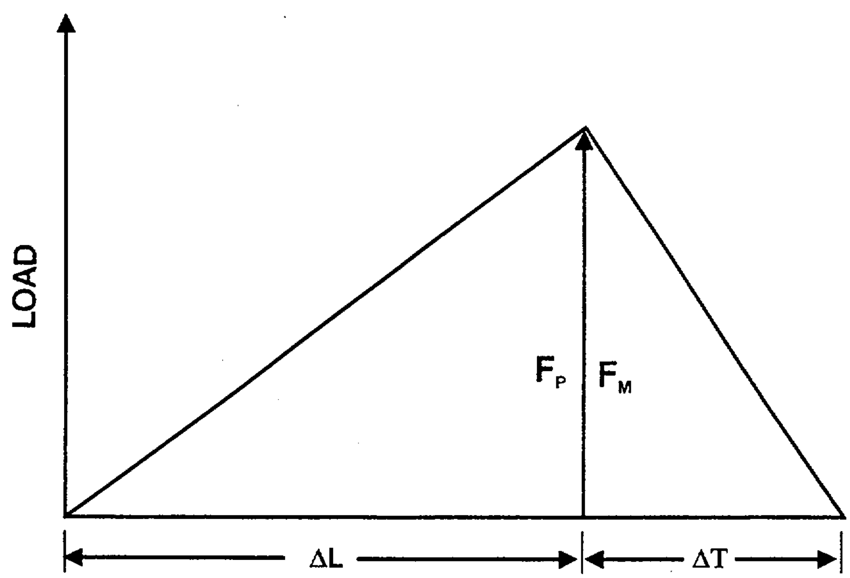
The analysis method for mechanical joints is best illustrated by using joint diagrams, such as those shown in Figures 6-14 and 6-15. A joint diagram illustrates the relationship between the forces and the deflections of the joint members and the fastener.

6.2.6.1 Preload Joint

Figure 6-14 is a snapshot of the forces and deflections existing in the fastener and joint members when the joint is preloaded, prior to the application of working loads. The preload in the fastener (F_P) is equal to the compressive load on the joint (F_M). The fastener is elongated by an amount (ΔL) due to the tensile load, and the joint is deflected by an amount (ΔT) due to the compressive load.

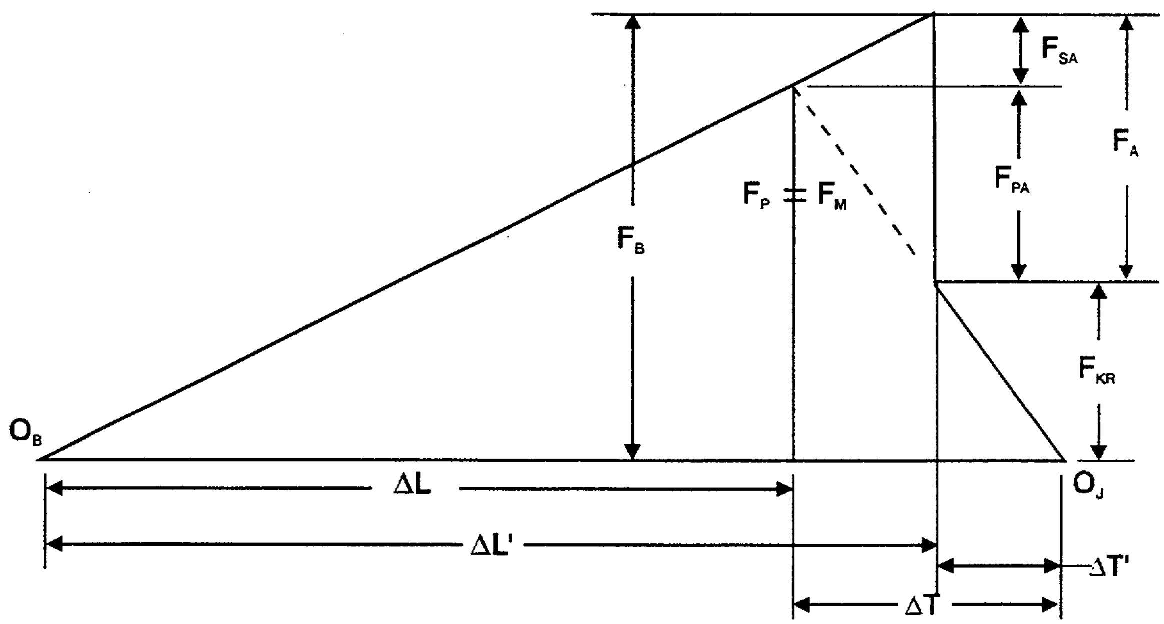
6.2.6.2 External Loads

In operation, the bolted joint is subjected to an axial tensile load (F_A). When this load is applied to the joint, the loads and deflections of the fastener and joint members change as shown in Figure 6-15. Generally, the load in the fastener is increased by an amount F_{SA}, while the compressive load in the joint is decreased by an amount F_{PA}.



F_p = Preload
F_M = Compressive Load in Joint
Δ_L = Elongation due to Tensile Load
Δ_T = Joint Deflection due to Compressive Load

Figure 6-14 — Preloaded Joint Prior to Application of Working Loads.



Joint Diagram with External Loads

F_P	=	Preload
F_M	=	Compressive Load
F_A	=	External Tension Load
F_{SA}	=	Increase in Fastener Load
F_{PA}	=	Reduction in Clamp Load due to Working Load
F_B	=	Bolt Load
ΔL	=	Elongation due to Tensile Load
ΔT	=	Deflection of Joint due to Compressive Load

Figure 6-15 — Externally Loaded Joint.

Figure 6-15 illustrates the following important concepts:

- The fastener stress is generally <u>increased</u> due to the application of external loads. The increase in fastener load (F_{SA}) is related to the external load (F_A) by the following equation:

  $$F_{SA} = \Phi F_A \tag{6-3}$$

  where Φ is a force ratio which depends on the relative stiffness of the fastener and the joint members.

  If the fastener is long and flexible and the joint is stiff, then the magnitude of Φ is low. A low value of Φ is desirable because it limits the increase in fastener load due to external loads. An expression for Φ is given by

  $$\Phi = K_B \;/\; (K_J + K_B) \tag{6-4}$$

  where the fastener stiffness (K_B) is

  $$K_B = \left(\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3} + \frac{0.5D}{A_1} + \frac{0.5D}{A_3}\right)\frac{1}{E} \tag{6-5}$$

where, A_i = Cross sectional area of fastener
L_i = Length of fastener section
0.5D = Stressed portion of the bolt head, thread engagement in tapped hole or nut
E = Modulus of elasticity of the fastener

  The subscripts in Equation 6-5 refer to the sections of the fastener shown in Figure 6-16. The half-diameter lengths are estimates of the effective stressed portion of the head and thread engagement, and are expressed as a percentage of the nominal fastener diameter (inside diameter).

  The joint stiffness (K_J) in Equation 6-4 may be estimated using a graph developed by Bickford (<u>30</u>), given in Figure 6-17. L_{eff}, the effective stressed length of the bolt, is the sum of the grip length and one half the engaged thread length, plus one half the engaged thread length or the bolt head thickness on the other end of the fasteners.

  If the external load is cyclic, then F_{SA} in Equation 6-3 is twice the amplitude of the cyclic load on the fastener. This cyclic amplitude is important in evaluating the fatigue performance of the fastener.

- The external load causes a reduction of the clamping load in the joint (F_{PA}). This is important since the principal objective of the joint design is to keep the

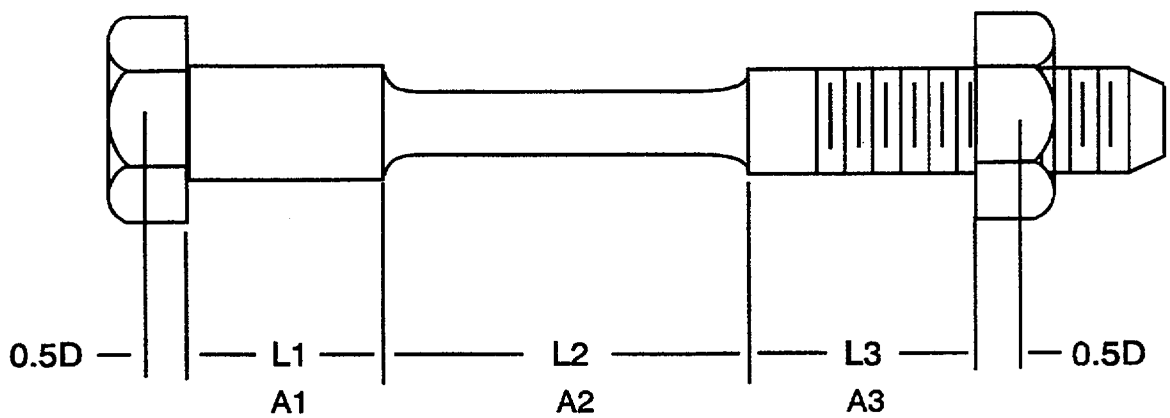
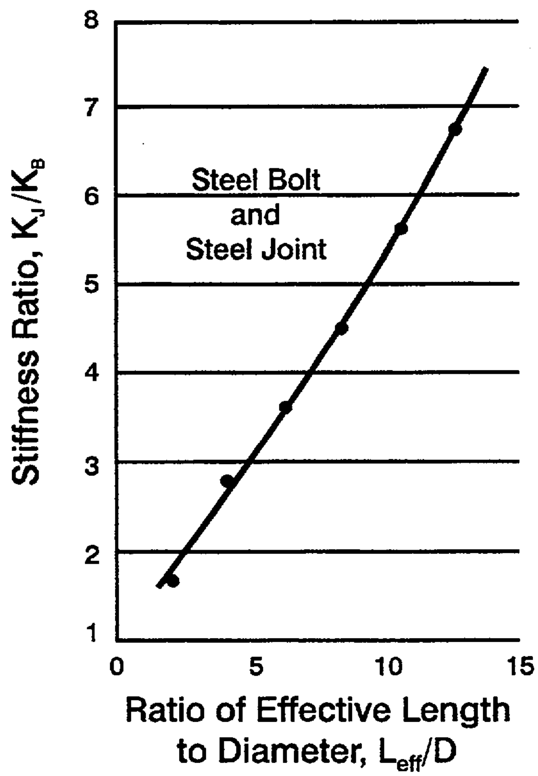


Figure 6-16 — Definition of Fastener Sections.



K_B	=	Bolt Stiffness $\left[\frac{lb}{in}\right]$
K_J	=	Joint Stiffness $\left[\frac{lb}{in}\right]$
L_{eff}	=	Effective Length of Bolt [in]
D	=	Diameter

Figure 6-17 — Bickford Graph of Joint Stiffness.

joint in compression. The residual compressive load in the joint (F_{KR}) may be expressed by

$$F_{KR} = F_P - F_{PA} \tag{6-6}$$

where, F_{AR} = Residual compressive load in the joint
F_P = Preload
F_{PA} = Reduction in compressive load due to the working load

The expression for the reduction in the compressive load (F_{PA}) is

$$F_{PA} = (1 - \Phi) F_A \tag{6-7}$$

Figure 6-15 illustrates these load parameters.

6.2.7 Preliminary Factors of Safety

Using Figure 6-15 and Equations 6-6 and 6-7, preliminary estimates of the suitability of a joint are possible. By comparing the preload (F_p) to the amount of joint unloading (F_{PA}), it can be determined whether sufficient preload exists for the joint to remain in compression. The preload should be greater than the joint unloading to achieve this goal.

The fatigue performance of the fastener may be estimated by comparing the cyclic stress amplitude ($F_{SA}/2$) to the fastener material endurance limit. The cyclic stress amplitude should be less than the endurance limit of the fastener.

6.2.7.1 Analysis Assumptions

The discussion thus far has been based on a number of simplifications. The following is a brief review of these simplifying assumptions:

- Linear elastic behavior. The fastener and joint members deform linearly with load. The straight lines on the joint diagrams relate the deformation to the load.

- External load. The working load is applied central to the joint members and in the axial direction. The loading plane for the introduction of the external load is from the bolt head to the nut. No bending, shear, torsion, or combined loads have been considered. For analysis of bending, refer to VDI 2230 (21).

- Preload. The preload introduced at assembly is exact. Neither assembly inaccuracy or relaxation of preload during service have been considered.

6.2.7.2 Preload Uncertainty and Relaxation

The joint compressive load (F_M) which is equal to the fastener preload (F_P) is developed in the joint at assembly. The magnitude of the compressive load on the joint, i.e., the preload, is the most important factor in determining the performance of the bolted joint. If the preload is too low, the external load will reduce the compressive load in the joint to zero, causing joint separation. Joint unloading is undesirable and is a violation of the first design objective, to maintain joint compression.

Figure 6-18 illustrates a method of evaluating the effects of preload inaccuracy and relaxation loss on joint performance. Figure 6-18a shows a theoretical estimate of preload required by the joint (F_{ma}). The magnitude of F_{ma} is sufficient to prevent joint unloading when an external load (F_A) is applied. The magnitude of the joint compression (F_{KRa}) is conservative and unloading does not occur. The joint compression (F_{KRa}) is the residual compressive load across the joint during operation.

In the assembly process, the actual preload achieved in the joint is less than the theoretical preload. Figure 6-18b illustrates the condition of the joint when less than the theoretical preload has been achieved. The bottom cross hatched area of the joint diagram represents the magnitude of preload which is uncertain due to inaccuracy of preloading techniques. Notice that the margin against unloading the joint has been reduced, that is F_{KRb} is smaller than F_{KRa}.

Figure 6-18c shows that preload relaxation further reduces the clamping load on the system. The margin against unloading is further reduced, that is F_{KRc} is smaller than F_{KRb} which is smaller than F_{KRa}.

The considerations necessary to determine the required joint clamp load are shown in Figure 6-19. The following summarizes these considerations:

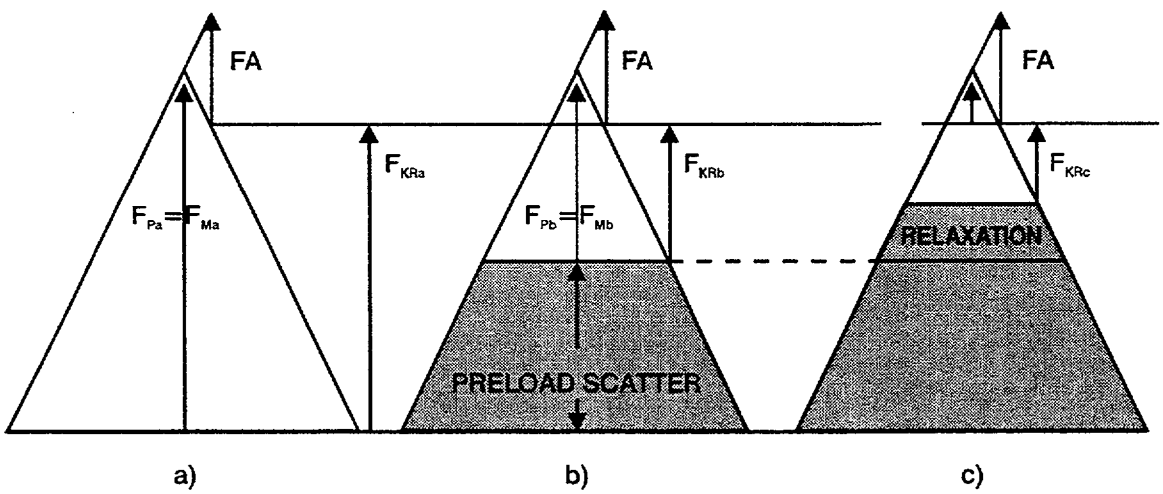
- F_M is the theoretical (nominal) joint compressive load which is equal to the fastener preload at assembly. This is the compressive load on the joint required to keep the joint from unloading during operation.

  F_{Mmin} is a revised estimate of the required compressive load which accounts for the expected relaxation of load (F_Z).

$$F_{M\min} = F_M + F_Z \qquad (6\text{-}8)$$

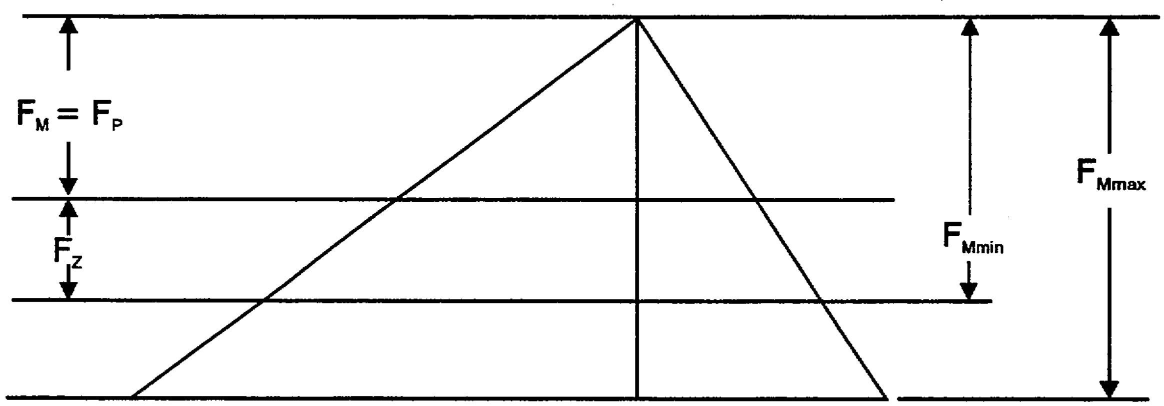
- The magnitude of joint compressive load is further adjusted to account for preload scatter, which depends on the assembly method. The preload scatter factor (a) is a term used to account for uncertainty and may be thought of as a design penalty for assembly method chosen. The most accurate tightening method is stretch control for a = 1.0. Torque is less accurate for a = 1.6. Table 6-1 (22) gives guidelines for choosing the tightening factor.

$$F_{Mmax} = \alpha \, F_{Mmin} \qquad (6\text{-}9)$$



F_A = Tension Load
F_M = Compressive Load
F_P = Clamp Load in Joint at

Figure 6-18 — Evaluating Effects of Preload Inaccuracy and Relaxation Loss.



Joint Diagram, Assembly

F_M = Joint Compressive Load
F_P = Preload
F_Z = Preload Relaxation

Figure 6-19 — Required Joint Clamp Load Considerations.

<table>
<tr><th colspan="4">Table 6-1</th></tr>
<tr><th colspan="4">Guidelines for the Tightening Factor (α_A)</th></tr>
<tr><th>Tightening Factor (α_A)</th><th>Tightening Method</th><th colspan="2">Remarks</th></tr>
<tr><td>1</td><td>Yield point controlled motorized tightening</td><td colspan="2" rowspan="2">The preload force fluctuation is determined mainly by the fluctuation of the yield point in the number of screws used. The screws are dimensioned here for F_{min}; the tightening factor α_A therefore does not apply for this type of tightening. The values in brackets serve for comparison of the tightening precision with the following methods.</td></tr>
<tr><td>1</td><td>Angle of rotation (turn-of-nut) controlled motorized or manual tightening</td></tr>
<tr><td>1.2</td><td>Tightening with elongation measurement of the calibrated screw</td><td colspan="2">Complicated method, very limited applications.</td></tr>
<tr><td>1.4 to 1.6</td><td>Torque control tightening with manual torque wrench or precision screwdriver with dynamic torque control. Experimental determination of the nominal tightening torque on original screw components (e.g. by elongation measurement of the screw)</td><td>Low values if a large number of setting or checking tests (e.g. 20). Low scatter of the resulting torque. Electronic torque limitation during assembly with precision screwdrivers.</td><td>Low values for:
• Small turning angle (e.g. relatively stiff joints.)
• Relatively soft counter piece.
• Counter-pieces which do not tend to tear (e.g. phosphated).</td></tr>
<tr><td>1.6 to 1.8</td><td>Torque controlled tightening with manual torque wrench or precision screwdriver with dynamic torque control. Determination of nominal torque by estimation of friction coefficient (surface and lubrication conditions).</td><td>Low values for:
• precise torque wrench (e.g. with clock gage)
• even tightening
• precision screwdriver (torque control).
High values for torque wrenches with signal- or release mechanism.</td><td>High values for:
• Large turning angle (e.g. relatively flexible joint and fine threads)
• High hardness of the counter piece, connected a with rough surface.
• Shape distortion.</td></tr>
</table>

Table 6-1		
Guidelines for the Tightening Factor (α_A)		
Tightening Factor (α_A)	Tightening Method	Remarks
1.7 to 2.5	Torque controlled tightening with screwdriver. Screwdriver with preset tightening torque, based on nominal tightening torque (for estimated friction conditions) with necessary adjustments.	Low values for: •Large number of tests (tightening torque) •Screwdriver with cut out coupling.
2.5 to 4	Impulse controlled tightening with impact wrench. Screwdriver with preset tightening torque, as above.	Low values for: •Large number of setting tests (tightening torque) •Impulse transmission with good fit (low wear).

The designer sizes the joint and selects the assembly preload based on F_{Mmax}. The designer must consider preload scatter and allow for it resulting in a more conservative design.

6.3 Safety Factors

6.3.1 Safety Factor for Preload Stress (S_{FP})

The safety factor for preload stress is defined by

$$S_{FP} = \frac{F_{SP}}{F_{M_{max}}} \tag{6-10}$$

where, F_{SP} = Fastener load capacity at the allowable stress (lb)
F_{Mmax} = Load required to keep the joint in compression (lb)

The preload stress factor of safety evaluates the preload capacity of the bolt (the numerator) against the load required by the joint (the denominator).

To increase the factor of safety, the numerator may be increased or the denominator may be decreased. To increase the numerator,

- Increase the fastener diameter.
- Choose higher strength fastener material.
- Allow the fastener to be preloaded to a higher stress.

To decrease the denominator,

- Reduce preload scatter by choosing a more precise assembly method.
- Reduce the embedment relaxation.
- Lower the force ratio (Φ) (refer to Section 6.2.6.2) by reducing the fastener stiffness relative to the joint stiffness (e.g. select a more flexible fastener and/or stiffer joint members).

6.3.2 Safety Factor for External Load (S_{EL})

Figure 6-20 illustrates that the fastener load is increased when a working load is applied to the joint. The working load factor of safety compares the fastener yield strength to the maximum load carried by the fastener and is given by:

$$S_{EL} = \frac{F_{YP}}{F_p + F_{SA}} \tag{6-11}$$

where, F_{YP} = Yield strength of fastener (lb)
F_P = Fastener preload (lb)
F_{SA} = Increased in fastener load due to external load (lb)

The working load factor of safety (S_{EL}) may be increased by:

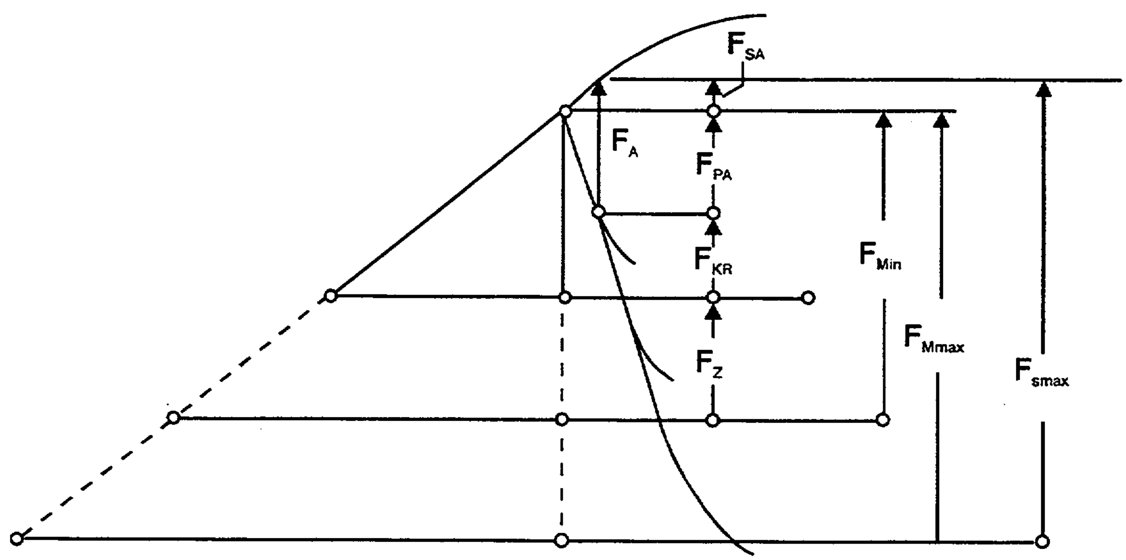
- Choosing high strength fasteners or utilizing a higher percentage of the fastener strength by allowing higher fastener stress.
- Reducing the portion of the working load carried by the fastener (F_{SA}) by modifying the joint stiffness ratio (F).

6.3.3 Safety Factor Against Fatigue Failure (S_f)

The fatigue safety factor, illustrated in Figure 6-21 (22), is a comparison of the fastener allowable cyclic stress to the mean stress, the bending stress, the preload, and the cyclic stress in the fastener. The fatigue safety factor is given by

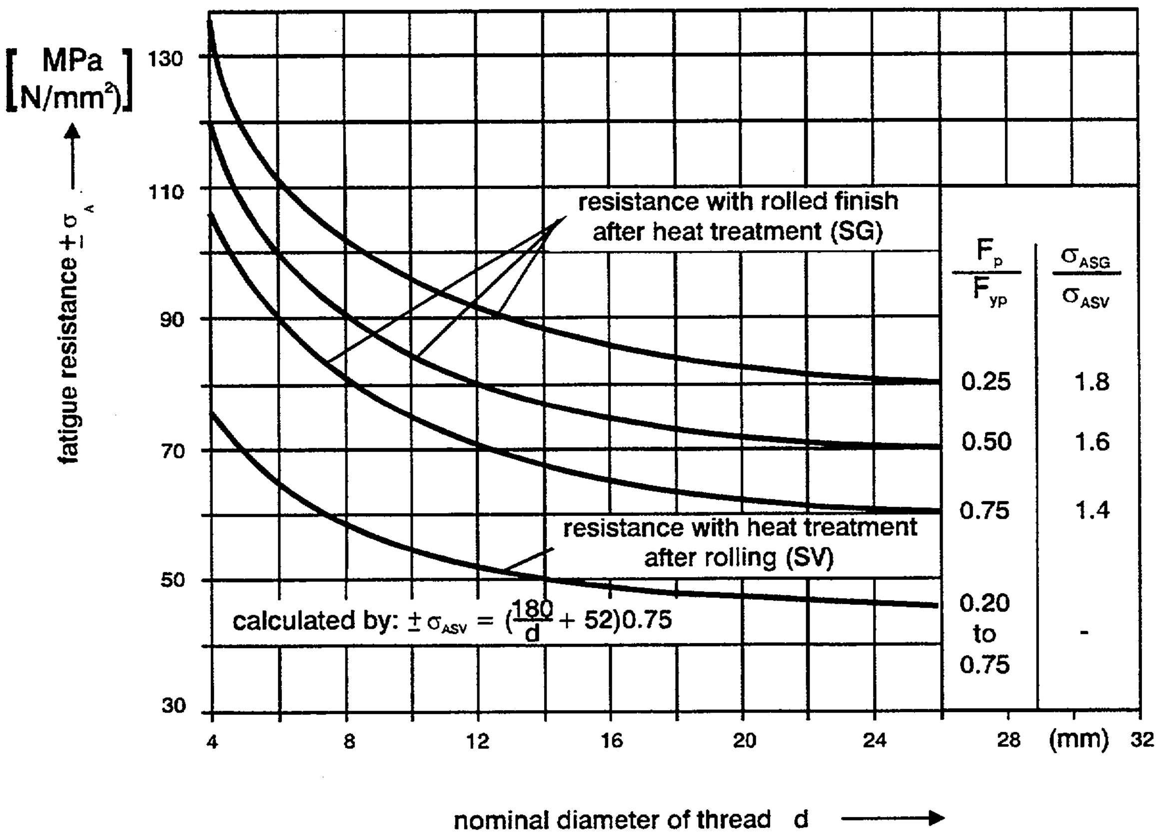
$$S_f = \frac{\sigma_{ASG}}{\sigma_{ASV} \ / \ 2} \tag{6-12}$$

where, s_{ASG} = Allowable cyclic stress amplitude for the fastener (See Figure 6-21) (psi)
s_{ASV} = Magnitude of cyclic stress in the fastener caused by the additional load F_{SA} and the bending stress (psi)



F_A = External Tension Load
F_M = Joint Compressive Load
F_S = Maximum Operating Fastener Load
F_{PA} = Reduction in Clamp Load
F_{KR} = Residual Compressive Load
F_Z = Load Relaxation
F_{SA} = Increase in Fastener Load

Figure 6-20 — Working Load Applied to a Joint.



F_P	=	Preload
F_{YP}	=	Yield Strength of Fastener
σ_{ASG}	=	Allowable Cyclic Stress Amplitute for the Fastener
σ_{ASV}	=	Magnitude of Cyclic Stress in the Fastener Caused by the Additional Load F_{SA} and the Bending Stress

Figure 6-21 — Determination of Fatigue Safety Factor (S_f).

The fatigue safety factor may be increased by:

- Using fatigue resistant fasteners
- Decreasing the cyclic stress in the fastener
- Decreasing the bending stress in the fastener.

6.3.4 Safety Factor for Surface Pressure (S_{SP})

The maximum bearing pressure in the joint under the bolt head or nut is calculated and compared to the allowable bearing pressure for the joint material.

6.3.5 Safety Factor for Shear Loads (S_Q)

The compressive force on the joint multiplied by the mating surface friction is the shear load capacity of the joint. Figure 6-22 illustrates these principles. The safety factor for shear loads results from a comparison of the shear load capacity to the applied shear load. The shear loads safety factor is given by:

$$S_Q = \frac{(F_P - F_Z - F_{PA})\mu}{F_Q} \tag{6-13}$$

where,
- μ = Friction coefficient at the joint parting surface
- F_P = Fastener preload (lb)
- F_Z = Relaxation of preload (lb)
- F_{PA} = Reduction in joint compressional load due to external load (lb)
- F_Q = Shear load (lb)

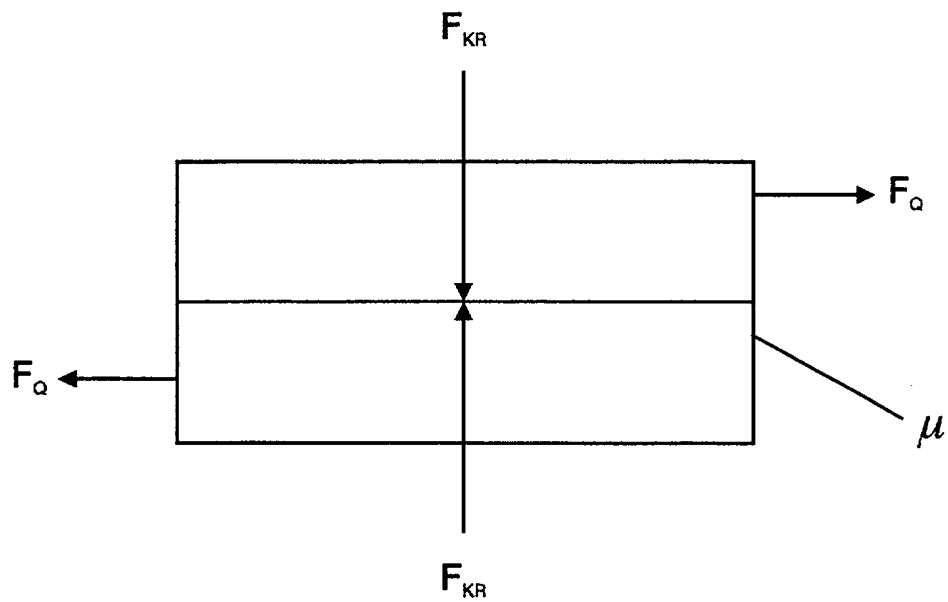
6.3.6 Safety Factor for Loosening (S_L)

Loosening is evaluated by determining if the joint unloads. The safety factor for loosening results from a comparison of the minimum required compressive load in the joint at assembly (F_{Mmin}) and the amount of joint unloading at the working load (F_{PA}). This is given by:

$$S_L = \frac{F_{M_{min}}}{F_{PA}} \tag{6-14}$$

6.4 Design Considerations for Miniature Fasteners

The quality of an assembly depends on the quality of the parts, particularly for miniature fasteners. In some cases, the design rules for large diameter fasteners must be modified when designing with small fasteners. It is extremely important to properly estimate the loads acting on parts joined with small fasteners.



μ = Friction Coefficient
F_Q = Shear Load
F_{KR} = Residual Compressive

Figure 6-22 — Safety Factor (S_Q) for Shear Loads.

Since no strength grading system exists for small screws, the amount of clamp load required for the particular assembly should be identified. Table 6-2 gives recommendations for thread size, hardness, and corresponding yield strength for optimum assembly. Although tensile strength values are more readily available, yield strength values are utilized in the design of miniature fasteners.

<table>
<tr><th colspan="3">Table 6-2*</th></tr>
<tr><th colspan="3">Calculating Required Fastener Strength</th></tr>
<tr><th>Thread Size</th><th>Tensile Stress Area (inches²)</th><th>Strength of Materials</th></tr>
<tr><td>00-90</td><td>0.00100</td><td>Base calculations on yield strength</td></tr>
<tr><td>0-80</td><td>0.00180</td><td>Carbon content and hardness primarily determine yield strength</td></tr>
<tr><td>1-64</td><td>0.00263</td><td></td></tr>
<tr><td>1-72</td><td>0.00278</td><td rowspan="9">These Rockwell "C" hardness values and corresponding yield strengths represent the standard range of through hardened parts:
• RC 28 = 75,000 psi yield strength
• RC 32 = 94,000 psi yield strength
• RC 38 = 130,000 psi yield strength</td></tr>
<tr><td>2-56</td><td>0.00370</td></tr>
<tr><td>2-64</td><td>0.00394</td></tr>
<tr><td>3-48</td><td>0.00487</td></tr>
<tr><td>3-56</td><td>0.00523</td></tr>
<tr><td>4-40</td><td>0.00604</td></tr>
<tr><td>4-48</td><td>0.00661</td></tr>
<tr><td>5-40</td><td>0.00796</td></tr>
<tr><td>5-44</td><td>0.00830</td></tr>
</table>

Normally a 4 point Rockwell "C" scale is used for heat treating tolerance. Carburized steel is a duplex material having a high carbon case and a low carbon core with a gradual transition from one to the other. Core hardness should be used as a conservative estimate of load carrying ability.

**Speck, J., "Fastening with Miniatures - Design Considerations," Fastening Technology, October 1984.*

The following example details the process used to calculate the required fastener strength:

A service load of 100 lbs is to be held by a No. 0 diameter screw. What minimum yield strength is required? The safety factor, designated by the designer, is 2.

Load (P) = 100 lbs x 2 (safety factor) = 200 lbs

Area (A) = 0.00180 in^2

Stress = P / A = 200 lbs / 0.00180 in^2 = 111,111 psi

Therefore, a minimum yield strength of 112,000 psi is required. A screw with a hardness of RC 38 should be chosen.

Torsional strength is critical for small diameter screws since it is easy to "twist off" (i.e., cause a torsional failure of a small screw). Table 6-3 gives an approximation of torsional and tensile strengths for one manufacturer's alloy steel thread forming screws and 18-8 stainless steel machine screws.

Table 6-3*

Miniature Screw Performance Data

			Maximum Torsional strength (inch-pound)		Maximum Tensile Strength (pounds)		
Nominal Size/Basic Diameter	Drill Size*	Hole Size*	Alloy Steel Thread Forming: Ferrous and Nonferrous Castings and Fabrications**	Machine Screws, Coarse Threads 18-8 Stainless	Alloy Steel Thread Forming: Ferrous and Nonferrous Castings and Fabrications**	Machine Screws, Coarse Threads 18-8 Stainless	Alloy Steel Thread Forming: Screws for Forming into Plastics*,**
0-80/0.0600	1/64	0.0510	4.0	1.8	340	150	200
2-56/0.0860	#48	0.0760	12.0	5.0	700	315	550
4-40/0.1120	#41	0.0960	25.0	12.0	1150	515	1000
6-32/0.1380	#31	0.1200	48.0	22.0	1730	775	1500

* *Drill and hole sizes conform to ANSI B18.6.4 latest requirements for Type C coarse thread series.*

** *Values relate to thread forming screws having round body, back tapered, screw thread, shank length geometry. Torsional and tensile strengths should be adjusted for other thread forming screws. Torsional strengths were derived under test conditions specified in ANSI B18.6.4, Paragraph 2.7.1.2.*

**Speck, J., "Fastening with Miniatures - Design Considerations," Fastening Technology, October 1984.*

Corrosion protection of miniature fasteners is usually accomplished by electroplating. Although the average plating thickness is less than 1% of the total pitch diameter for a 1/4-20 screw, the plating thickness is almost 4% of the total pitch diameter for a 0-80 screw.

Cadmium, zinc, and nickel are common electroplating materials. Cadmium is a poor choice due to the toxicity and expense. Cadmium is superior to zinc only under marine conditions where salt is prevalent. In industrial environments where sulphur from combustion processes is present, the useful life of cadmium coatings is often less than that of comparable thickness zinc/chromate coatings. Stainless steel Types 302 or 410 are suitable material choices for corrosion resistant miniature fasteners.

6.5 Thermal Stress

The thermal stress in a joint results from the difference in the coefficients of thermal expansion of the fastener and joint material as well as from temperature differences. Timoshenko (30) developed a solution to the following example:

A cold rolled steel bolt of length (l) = 12 inches passes through a hard drawn copper tube of the same length, as shown in Figure 6-23a. The entire assembly is at a temperature (T_1) of 70°F and is tightened snugly. Subsequently, the nut is tightened 1/4 turn, and the entire assembly is raised to a temperature (T_2) of 140°F. What stresses will exist in the bolt and the tube under these conditions?

The cross sectional area of the steel bolt (A_s) is 1/2 in^2; the modulus of elasticity (E_s) is 30 x 10^6 psi; the coefficient of thermal expansion (α_s) is 6.5 x 10^{-6} inches/inches/°F; and the thread pitch (p) is 1/8 inch. For the copper tube, A_c = 3/4 in^2; E_c = 16 x 10^6 psi; and α_c = 9.3 x 10^{-6} inch/inch/°F.

Preload stress is developed in the assembly due to the 1/4 turn of the nut. Based on static equilibrium, the compressive force in the copper tube (S_c) must balance the tensile force in the steel bolt (S_a), refer to Figure 6-23b. This relationship is given by:

$$S_s = S_c \qquad (6\text{-}15)$$

Furthermore, since the final length of the bolt and the tube must be equal, it follows that the amount that the tube is shortened plus the amount the bolt is extended must be equal to the thread displacement of the nut along the bolt. Expressed algebraically, this relationship is given by

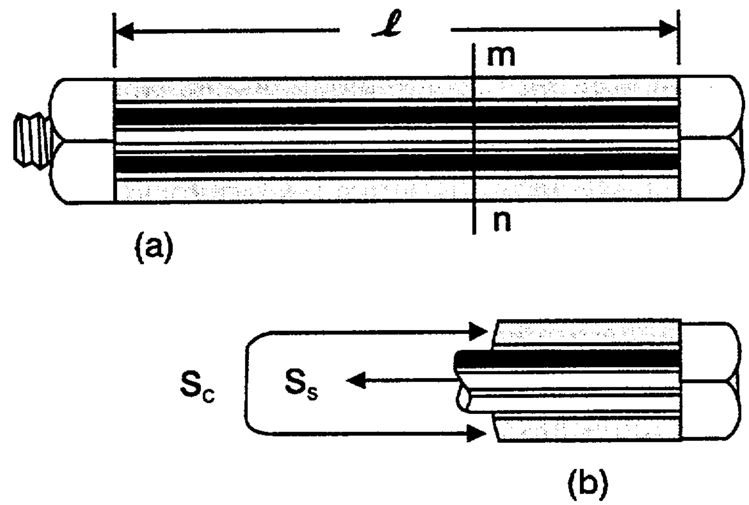
$$\delta_s + \delta_c = pn \qquad (6\text{-}16)$$

where,
- p = Thread pitch
- n = Fraction of one complete turn used to tighten (in this example, n = 1/4 turn).
- δ_s = Extension of steel bolt (in)
- δ_c = Compression of copper tube (in)

The total extension of the steel bolt is

$$\delta_s = S_s l \ / \ A_s E_s + \alpha_s l \Delta T \qquad (6\text{-}17)$$

where, ΔT = Net increase in temperature.



ℓ	=	Length
S_C	=	Compressive Force in the Copper Tube
S_S	=	Shear Stress
m-n	=	Section Line Which Gives Free Body 'b'

Figure 6-23 — Cold Rolled Steel Bolt in Hard Drawn Copper Tube.

The total shortening of the copper tube is

$$\delta_c = S_c l \,/\, A_c E_c - \alpha_c l \Delta T \tag{6-18}$$

Substituting these expressions for δ_s and δ_c into Equation 6-16 and noting from Equation 6-15 that $S_s = S_c = S$, we obtain

$$S(1 \,/\, A_s E_s + 1 \,/\, A_c E_c) + (\alpha_s - \alpha_c)\Delta T = pn \,/\, l \tag{6-19}$$

From the given numerical data,

$$S_s = S_c = S = 18{,}670 \text{ lbs}$$

The corresponding stresses are σ_s = 37,340 psi tension and σ_c = 24,900 psi compression.

6.6 Modulus of Elasticity

The modulus of elasticity, also known as Young's modulus, is the ratio of the stress on a body and the resulting strain. Figure 6-24 gives the modulus of elasticity for various bolting materials at temperatures from 70°F to 800°F.

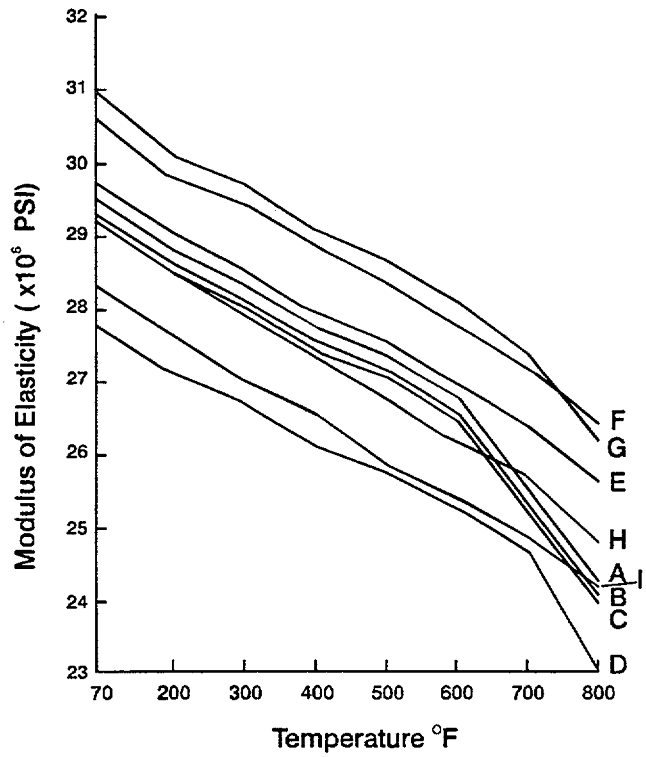
6.7 Coefficient of Thermal Expansion

Figure 6-25 gives the coefficient of thermal expansion and Young's Modulus for various bolting materials at ambient conditions. Table 6-4 gives the coefficient of expansion for joint materials.

Table 6-4

Thermal Expansion Coefficients for Bolting Materials

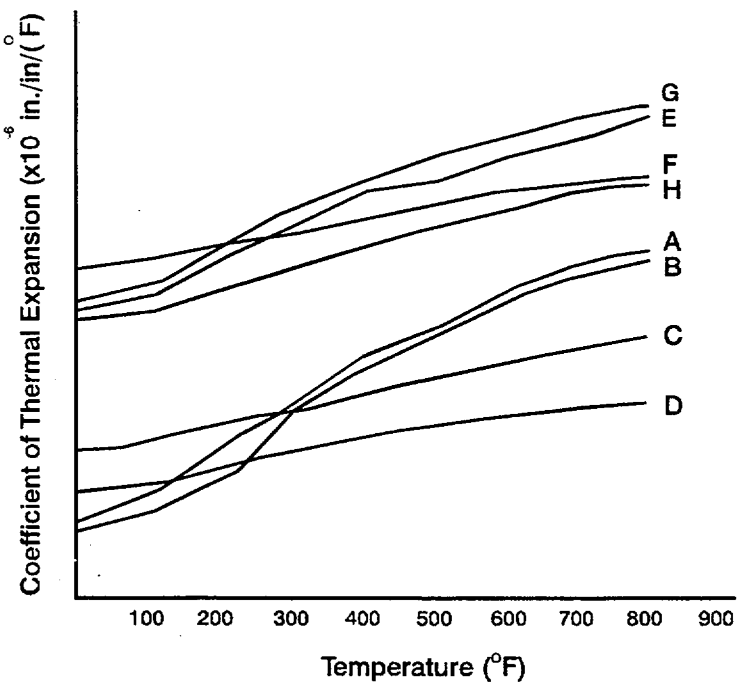
Bolt Material	Thermal Expansion Coefficients (in/in/°F)	Young's Modulus (psi)
Aluminum	12.8×10^{-6}	10×10^{6}
Copper	9.3×10^{-6}	17.5×10^{6}
Carbon Steel	6.5×10^{-6}	29.5×10^{6}
Low Alloy Steel	5.5×10^{-6}	29.7×10^{6}
High Alloy Steel	8.9×10^{-6}	28.3×10^{6}
High Nickel Alloy	6.8×10^{-6}	30.0×10^{6}
Titanium (Unalloyed)	4.6×10^{-6}	15.5×10^{6}



A = Carbon Steel (A307, A36, Carbon = (0.30%)
B = Carbon Steel (Carbon) = 0.30%
C = Carbon Molybdenum Steel
D = Nickel Steel (B8,B8M)
E = Chrome Molybdenum Steel (1/2 to 2 Cr)
F = Chrome Molybdenum Steel (2 1/4 to 3 Cr)
G = Chrome Molybdenum Steel (5 to 9 Cr)
H = Straight Chromium Steel
I = Austenitic, High Alloy Steel

Figure 6-24 — Modulus of Elasticity for Bolting Materials at Temperatures from 70° to 800°F

Reference EPRI NP5067 "Good Bolting Practices".



A	=	A193	B16, A540 B21
B	=	A193	B7, A320 L7,L7M,L7B,L43
C	=	A193	B5
D	=	A193	B6
E	=	A193	B8,B8A
F	=	A193	B8T
G	=	A193	B8C
H	=	A193	B8R,B8RA

Figure 6-25 — Coefficient of Thermal Expansion for Bolting Materials.

Reference EPRI NP5067 "Good Bolting Practices".

6.8 Set Screws

A set screw is a fastener used to secure a collar, sleeve, or gear to a shaft (31). In contrast to other fastening devices, the set screw is primarily a compression device. The forces developed by the screw point during tightening produce a strong clamping action which resists the relative motion between the assembled parts.

Failures, such as collars spinning on shafts are affected by the selection, design, and application of set screws. The proper set screw for a particular application is selected based on the head style, size, and point style which will provide the required holding power.

Set screws are categorized by the head and point style. Figure 6-26 illustrates basic head forms and point styles for set screws. Selection is based on factors, such as the type of driver used, dimensional and clearance constraints, weight savings, safety, and appearance.

6.8.1 Set Screw Size Selection

Screw size selection is based on the holding power required for a particular application. Figure 6-27 is a typical shaft and collar assembly in which the force (F) developed by the cup face on the shaft due to tightening produces an equal reactive force (F_1). This clamping action results in two frictional forces, one between the shaft and the collar (F_2) and the other between the shaft and the point (F_3). These frictional forces provide most of the resistance to relative axial and torsional movement of the parts.

The magnitude of the force is the principal factor in determining the axial holding power of the set screw or the resistance of the assembly to relative movement along the longitudinal axis of the shaft. Some additional load resistance is developed by the penetration of core points into the shaft. No spotting hole is required for cup point and cone point set screws. Because of the small face area, these screws penetrate the shaft more than oval point or flat point set screws. A set screw of a given size can be used on shafts of different sizes. The torsional holding power of the set screw is determined by multiplying the axial holding power by the shaft radius.

Engineers often use a rule of thumb which states that the diameter of a hollow set screw is equal to approximately half the diameter of the shaft. The range of usefulness of this rule is limited. Table 6-5, developed from experimental data for cup point set screws, provides a guide for the proper sizing of set screws. While it was developed for a specific set screw form and point style, the values given in the table can be modified to provide design data for many other forms and point styles of set screws.

The holding resistance values given in Table 6-5 are ultimate strengths and should be considered with safety factors appropriate to the given application and load conditions. A safety factor of 1.5 to 2.0 under static load conditions and 4.0 to 8.0 under dynamic conditions should provide adequate results.

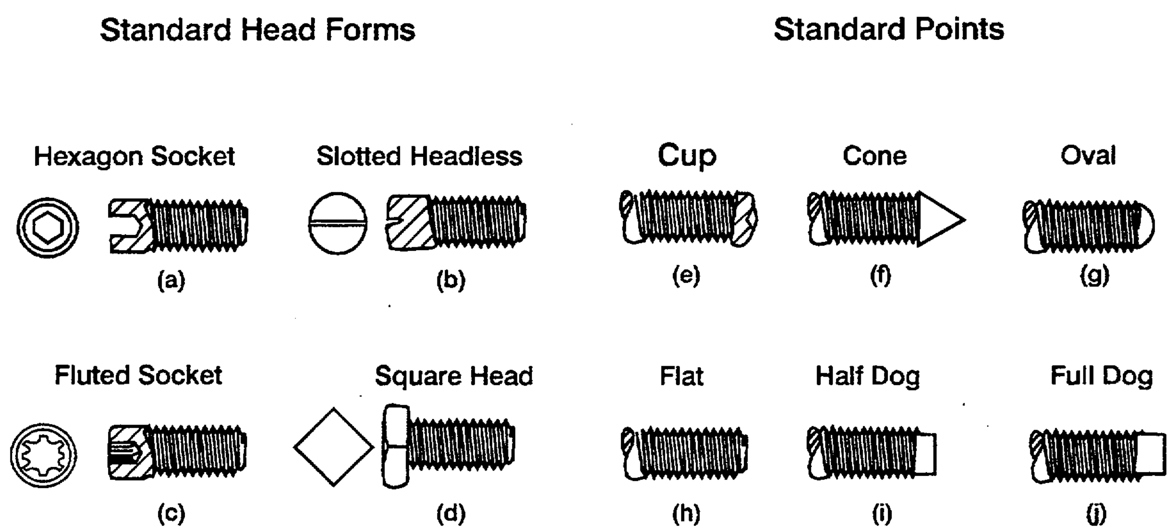
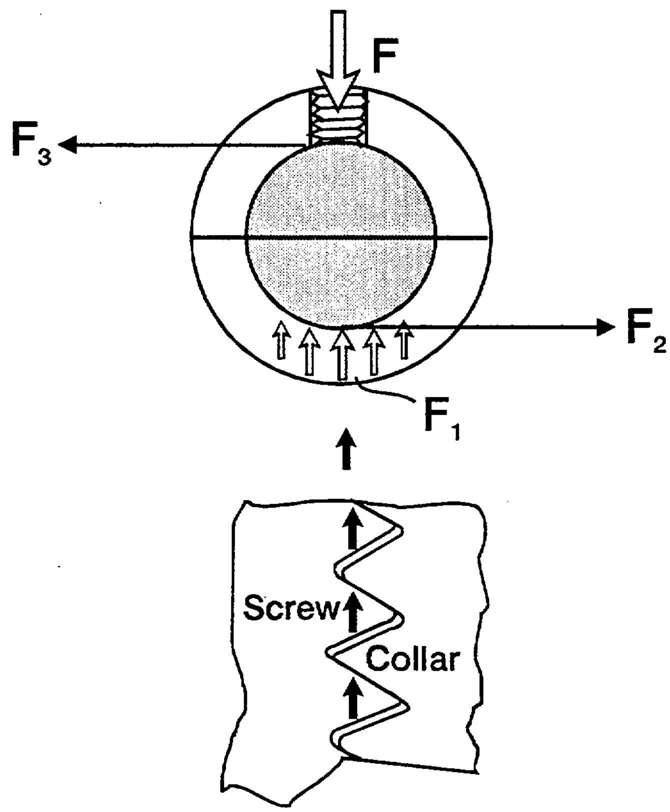


Figure 6-26 — Set Screw Basic Forms and Point Styles.



F = Force Developed by Cup Face on the Shaft due to Tightening
F_1 = Reactive Force
F_2 = Frictional Force Between Shaft and Collar
F_3 = Frictional Force Between Shaft and Point

Figure 6-27 — Shaft and Collar Assembly

6.8.2 Additional Factors in Selecting Set Screws

Additional factors should be considered when selecting the optimum set screw for an application. These factors include the seating torque, the point style, the relative hardness, the length of the thread engagement, the thread type, the type of driver, the number of set screws, and the type of plating. These factors are discussed in the following paragraphs.

6.8.2.1 Seating Torque

Experiments have shown that torsional resistance power is almost directly proportional to the seating torques for cup, flat, and oval point set screws. Figure 6-28 illustrates torsional holding power as a function of seating torque for a 5/16 inch diameter knurled, cup point screw seated against a 1-inch diameter shaft with a hardness of RC15. A 50% increase in the seating torque will increase the holding power of a set screw by 50%, within the strength limits of the assembly.

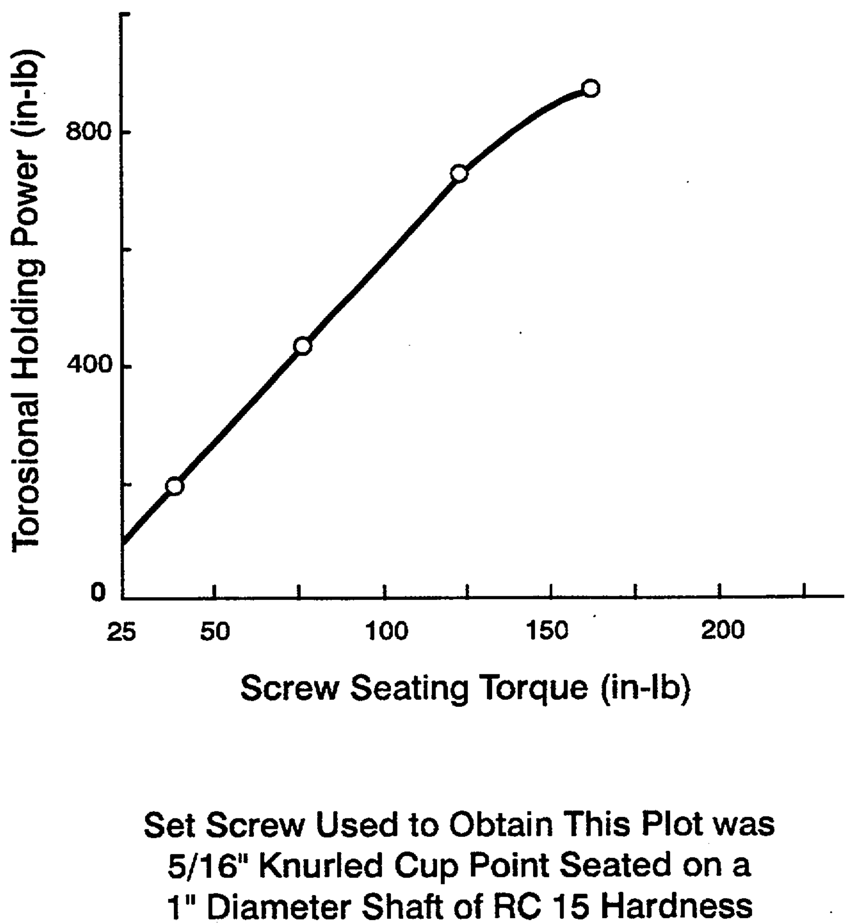


Figure 6-28 — Screw Seating Torque/Torsional Holding Power Curve (31).

Table 6-5 (31)

Torsional Holding Power for Cup Point Set Screws

Shaft Diam.	Nominal Size																		
	# 0	# 1	# 2	# 3	# 4	# 5	# 6	# 8	# 10	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	3/4"	7/8"	1"
	Seating Torque (inch-pounds)																		
	1	1.8	1.8	5	5	10	10	20	30	87	165	280	430	820	820	1325	2400	5200	7200
	Axial Holding Power (pounds)																		
	50	65	85	120	160	200	250	385	540	1000	1500	2000	2500	3000	3500	4000	5000	6000	7000
1/16"	1.5	2	2.6	3.2															
3/32"	2.3	3	4	5.6	7.5														
1/8"	3.1	4	5.3	7.5	10	12.5													
5/32"	3.9	5	6.6	9.3	12.5	15.6	19	30											
3/16"	4.7	6.1	8	11.3	15	18.7	23	38	51										
7/32"	5.4	7.1	9.3	13	17.5	21.8	27	42	59										
1/4"	6.2	8.1	10.6	15	20	25	31	48	68	125									
5/16"		10	13.2	18.7	25	31.2	39	60	84	156	234								
3/8"			16	22.5	30	37.5	47	72	101	187	280	375							
7/16"				28.3	35	43.7	55	96	118	218	327	437	545						
1/2"					40	50	62	108	135	250	375	500	625	750					
9/16"						56.2	70	120	152	281	421	562	702	843	965				
5/8"						62	72	144	169	312	468	625	730	937	1060	1250			
3/4"							94	168	202	375	562	750	937	1128	1310	1500	1875		
7/8"							109	192	238	437	656	875	1086	1310	1530	1750	2190	2820	
1"									270	500	750	1000	1260	1500	1750	2000	2500	3000	3500
1 1/4"									338	625	937	1250	1580	1875	2190	2500	3125	3750	4375
1 1/2"										750	1125	1500	1875	2250	2620	3000	3750	4500	5250
1 3/4"											1310	1750	2210	2820	3030	3500	4375	5250	6150

Table 6-5 (31)

Torsional Holding Power for Cup Point Set Screws

Shaft Diam.	Nominal Size																		
	# 0	# 1	# 2	# 3	# 4	# 5	# 6	# 8	# 10	1/4"	5/16"	3/8"	7/16"	1/2"	9/16"	5/8"	3/4"	7/8"	1"
	Seating Torque (inch-pounds)																		
	1	1.8	1.8	5	5	10	10	20	30	87	165	280	430	820	820	1325	2400	5200	7200
	Axial Holding Power (pounds)																		
	50	65	85	120	160	200	250	385	540	1000	1500	2000	2500	3000	3500	4000	5000	6000	7000
2"											1500	2000	2500	3000	3500	4000	5000	6000	7000
2 1/2"													3125	3750	4370	5000	6250	7500	8750
3"														4500	5260	6000	7500	9000	9500
3 1/2"															6120	7000	8750	10500	11250
4"																8000	10000	12000	14000

6.8.2.2 Point Style

The penetration of a hollow point set screw provides as much as 15% of the total holding power of the screw. The increased holding power of a cone point set screw, which contains neither a spotting hole or a predrilled hole in the shaft, is due to the point's deep penetration. The oval point set screw has less contact area and thus gives a smaller increase in holding power.

The axial holding power data given in Table 6-5 can be applied to different set screw point styles by multiplying the values given for cup point set screws by the following:

- Cone point set screws: 1.07
- Dog point set screws: 0.92
- Flap point set screws: 0.92
- Oval point set screws: 0.90.

The values given in Table 6-5 were obtained based on the assumption that the point of the screw is not specially reset into the shaft and that the penetration results solely from tightening. For example, a dog point seated in a hole drilled in a shaft acts only as a pin. Thus, the holding power is not obtained from Table 6-5, but is determined by the shear strength of the screw material.

6.8.2.3 Relative Hardness

Figure 6-29 shows holding resistance as a function of shaft hardness. Figure 6-29 uses a hard (RC50), knurled cup point set screw seated at 165 in-lb. Notice that the holding resistance decreases as the shaft hardness approaches the screw hardness.

6.8.2.4 Length of Thread Engagement

If sufficient engagement is available to prevent stripping in the tightening process, the length of the thread engagement has no noticeable effect on axial and torsional holding power. The length of engagement depends on such factors as the amount of applied load, the type of material, the type of thread, and the screw diameter. In most cases, the minimum recommended length of engagement is equal to the diameter of the set screw. This length will normally permit the development of recommended seating torques without the danger of thread stripping. The tabulated values of seating torque in Table 6-5 were developed under the assumption that the engagement length was sufficient to prevent stripping.

6.8.2.5 Thread Type

Experimental results indicate that coarse and fine threads of the same class of fit perform identically. Consequently, the values given in Table 6-5 apply to either thread type.

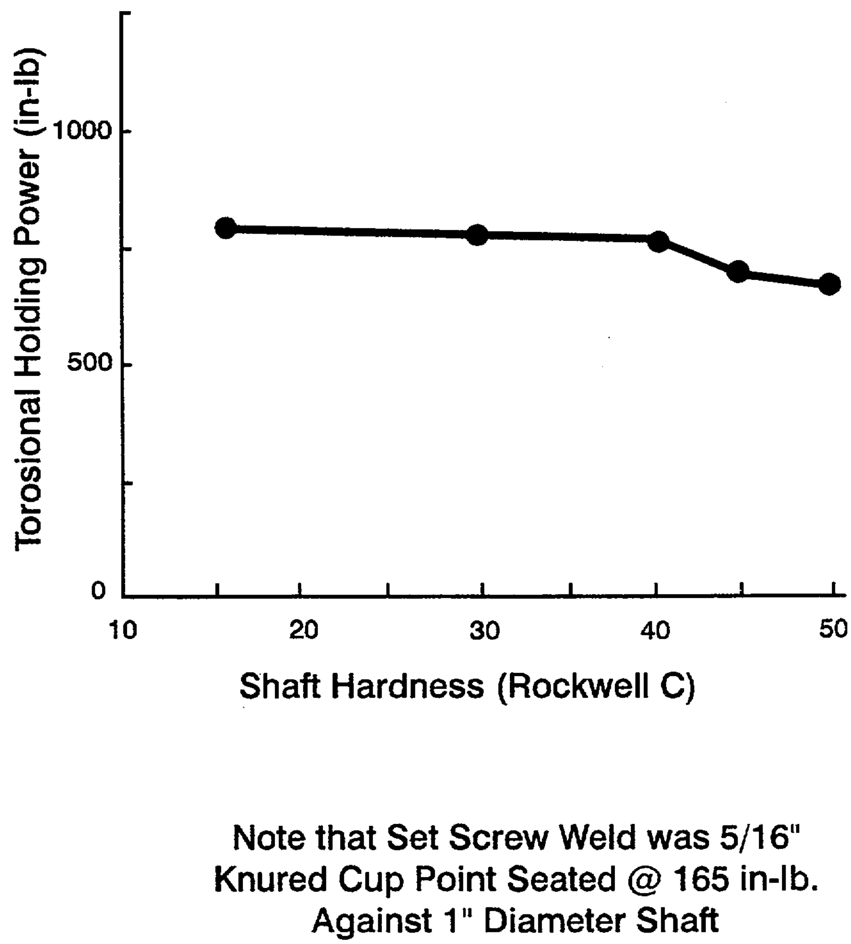


Figure 6-29 — Shaft Hardness/Torsional Holding Power Curve (31).

6.8.2.6 Type of Driver

The torsional holding power values given in Table 6-5 apply to socket type set screws. However, the values also apply to slotted and square head set screws, provided the indicated seating torque is developed. Although the type of the screwdriver has no direct bearing on the holding power, it does have an effect on the amount of seating torque which can be attained.

For the slotted set screw, the maximum seating torque is that which can be developed by a screwdriver. Deformation of the screw slot occurs at a torque much less than that which would strip the threads.

The maximum torque which can be applied to socket or spline head set screws is also lower than that which would strip the threads. However, it is higher than that which can be developed by a screwdriver. Consequently, the torque which can be applied is a function of the driver. Square head set screws can be tightened with a wrench until the threads strip or until the screw fails in torsional shear. Table 6-6 lists typical recommended installation torques for square head set screws.

Table 6-6 (31)

Recommended Tightening Torque for Square-Head Set Screws

Screw Size (in)	Recommended Tightening Torques (in-lb)
1/4	212
5/16	240
3/8	828
7/15 (7/16)	1344
1/12	2100
5/8	4248
3/4	7704

6.8.2.7 Number of Set Screws

The torsional and axial holding powers are increased by approximately 30% when a second set screw is installed diametrically opposed to the first screw. However, the holding powers are approximately doubled when the second screw is installed in an axial line with the first screw. Figure 6-30 illustrates the relationship of the angle between the two screws and the total holding power. The optimum displacement for two screws installed on the same circumferential line is 60°. This displacement is recommended as a compromise between maximum holding power and minimum metal between tapped holes. A 60° displacement gives 1.75 times the holding power of a single screw.

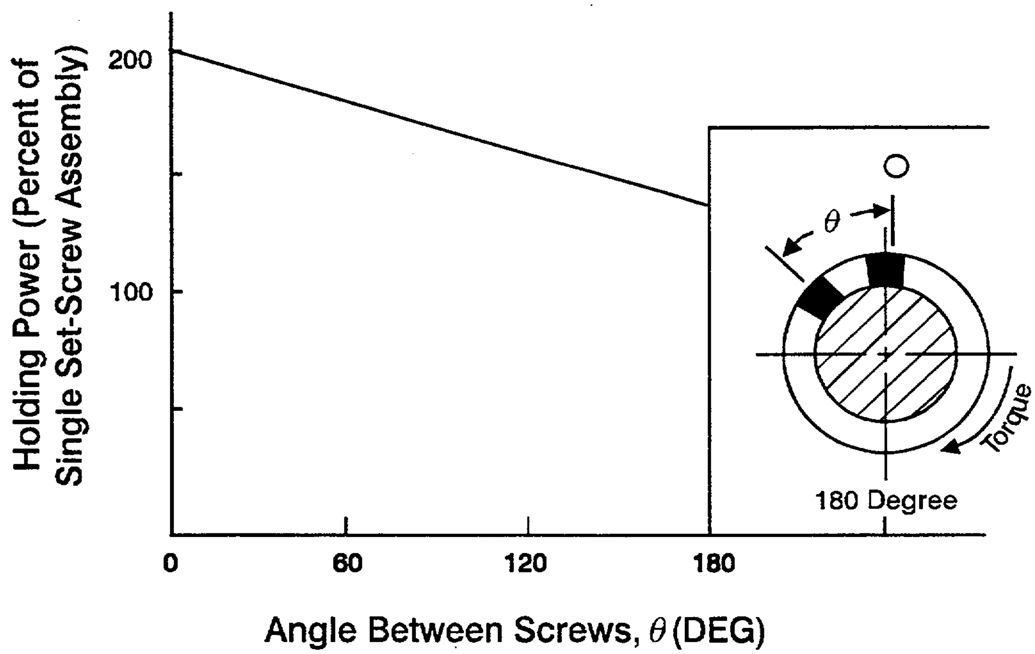


Figure 6-30 — Angle Between Two Screws/Total Holding Power Curve (31).

6.8.2.8 Type of Plating

A soft plating, such as cadmium or zinc, increases the holding power by 5% to 10% for a given tightening torque. The plating acts as a lubricant, resulting in an increase in preload. A comparable increase can be achieved by using a thread lubricant. Set screws may also be plated to inhibit corrosion.

6.9 Locking Devices

Threaded fasteners often include locking devices which maintain preload and/or retain lightly preloaded fasteners. Threads can be locked by preload, lock nuts, jam nuts, lock washers, Belleville washers, adhesives, tack welding, lock wire, staking, and cotter pins.

6.9.1 Preload as a Locking Device

Loosening of a fastener can be prevented by providing sufficient preload. The frictional forces between the load bearing threads resist the loosening forces. These frictional forces depend on the coefficient of friction between the mating threads, the magnitude of the load, the thread pitch, and the lubricity of the surfaces. The following conclusions have been made:

- A minimum preload of 20% of yield is required if preload is to be relied on for locking.
- Fine threads offer more resistance to loosening than coarse threads.

6.9.2 Prevailing Torque Lock Nuts

A prevailing torque nut, is a nut which provides frictional resistant to rotation because of special features of the threads. The prevailing torque is defined as the torque required to run a vibration resistant nut down to the joint surface. This resistance to free turning of the threads will provide the resistance to loosening and is accomplished by a nylon collar or distorted thread form present in either the male or the female threads. The prevailing torque must be added to the required clamping torque when determining the assembly torque for a prevailing torque lock nut.

6.9.3 Jam Nuts

A jam nut is a nut used in combination with another nut to prevent the fastener from loosening. The process is sometimes called double nutting. The jam nut can be thinner or of the same thickness as the first nut. Some jam nuts have special features that require placement next to the joint and preloaded by the other nut. The compressive load produced by the upper nut distorts the threads causing the interference that creates the lock.

6.9.4 Lock Washers

The ASME Code states that lock washers should not be used on fasteners greater than 1/4 inch. On fasteners less than 1/4 inch, however, two styles of lock washers are common:

spring action helical washers and toothed washers. Belleville washers, a type of spring-action helical washer, are discussed in Section 6.9.5.

6.9.4.1 Spring Action Helical Washers

ANSI/ASME Section B18.21.1 lists the materials from which spring action helical washers can be made. Most of these washers are fabricated of a hardened carbon steel. Because the washers are heat treated, they are placed under the element that is turned and compressed elastically during tightening. Spring action helical washers exert a spring-back reaction force which helps retain fastener tension and resistance to loosening. A disadvantage of spring action helical washers is that the spring flattens at relatively low loads compared to preload. These washers are available in different thicknesses (i.e. regular, heavy, extra duty, and hi-collar). The hi-collar washers are used only with the 1960 series of socket head cap screws.

6.9.4.2 Toothed Washers

Toothed washers are made from hardened carbon steel. The teeth on the washer dig into the fastener bearing surface, thus developing a greater resistance to loosening. However, the teeth can damage a bearing surface that has been treated with a protective coating. Figure 6-31 illustrates two types of internal toothed lock washers. These washers are ineffective without the development of a reasonable preload. An external toothed lock washer is also available. However, the internal toothed washer is recommended for standardization.

6.9.5 Belleville Washers

Belleville spring or conical washers are available for a variety of applications. The washers may be stacked to increase the load for a given deflection. Conversely, the washers may be used in series to increase the deflection for a given load.

6.9.6 Adhesives

The most frequently used adhesive is a mixture of an epoxy resin and a hardening agent. The resulting adhesive bond provides an effective locking mechanism. The use of adhesives makes assembly difficult. Adhesives are only recommended for permanent, one time applications.

6.9.7 Tack Welding

Tack welding can be used for carbon steel fasteners, but it is not recommended for hardened alloy materials.

6.9.8 Lock Wire

Lock wire is useful in multi-fastener applications. The wire is fed through holes in the heads of the bolts and screws, thus tying the fasteners together. The key to the effectiveness of lock wire is routing the wire through the fastener heads. The wire should be routed so that if a fastener moves in the direction of removal, the tensioned wire

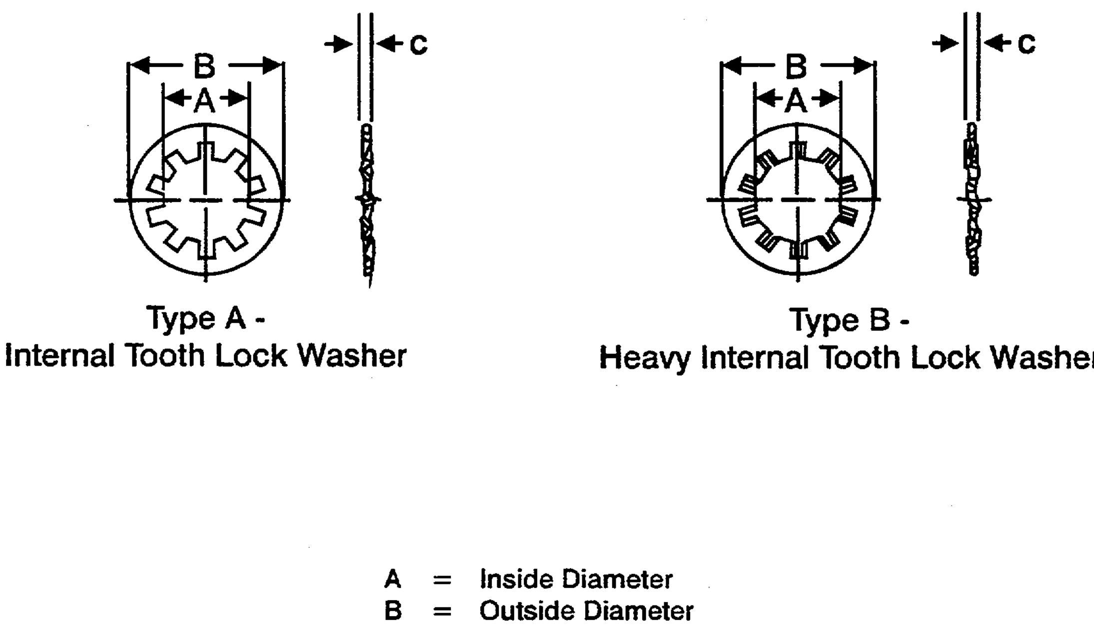


Figure 6-31 — Internal Toothed Lock Washers.

tightens the adjacent fasteners. Using lock wire to prevent loosening requires increased installation time, drilled head fasteners, and proper routing of the wire.

6.9.9 Staking of Threads

Staking threads with a center punch is another effective locking method. The major disadvantage of this technique is that the threads become damaged, making disassembly more difficult.

6.9.10 Cotter Pins

Cotter pins are used with castellated nuts whose strength is approximately 80% of that of full thickness nuts. To install cotter pins, align the nut slots with the holes drilled through the shank of an externally threaded fastener. This installation and the resulting deformation are critical to the performance of cotter pins. Difficulties in using cotter pins can result from locating the hole in the bolt shank and from a reduction in the nut strength due to slotting.

6.10 Plain Washers

Hard washers are recommended in several bolting standards. These washers benefit the bolted joint in the following ways:

- By distributing the fastener load over a larger area of the joint, washers increase the ratio of the joint stiffness to the bolt stiffness. This increase in stiffness ratio diminishes bolt fatigue problems.
- Washers maintain uniform interface forces between joint members. This uniformity improves gasket performance.
- Washers facilitate the assembly of poorly mated parts by bridging slotted or oversized holes. Washers are mandatory in many applications for this reason.
- Washers significantly reduce the friction at the bearing surface of the turned element. This reduction results an improvement in the accuracy and repeatability of torquing operations.
- Washers prevent damage to soft joint surfaces.
- Washers reduce the amount of embedment between nut, bolt, and joint members, thus reducing relaxation after tightening.

6.11 Hardened Washers

Hardened washers used under the bolt head and the nut distribute bearing stresses, increase grip length, improve joint to bolt stiffness ratios, and provide a hard, smooth surface for torquing during fastener tightening. Hardened washers are generally used when the bearing surface is softer than the fastener material.

Structural codes require the use of heavy hardened washers when using A325 or A490 fasteners and torque control as the assembly method. Heavy hardened washers are also required if oversized holes are used in the joint members.

6.12 Electrical Connections

An electrical connection must be designed to remain tight and maintain good conductivity through a large temperature range. Meeting this design requirement is difficult if the materials specified for the bolt and the conductor are different and have different rates of thermal expansion. For example, copper and aluminum bus materials expand faster than most bolting materials. If thermal stress is added to stresses inherent at assembly, the joint members or fasteners can yield. If plastic deformation occurs during thermal loading (i.e., heatup) when the connection cools, the joint will be loose. The following sections provide material recommendations and installation techniques for copper and aluminum bus installations.

6.12.1 Copper Bus Bars

Copper bus is typically hard temper (ASTM B187, Standard Temper HO4) and has a minimum tensile strength of 40 ksi. Because hard copper has no detectable yield point, the tensile strength is used in all calculations. The coefficient of thermal expansion for copper bus is 9.3×10^{-6} inches/inches/°F and Young's modulus is 17.5×10^{6} psi.

6.12.2 Aluminum Bus Bars

Aluminum bus is typically medium hard temper (ASTM B2317 6061-T61). The material has a tensile strength of 20 ksi and a yield strength of 15.0 ksi. Softer tempers are seldom used because they have low yield strength (8 ksi), which would permit excessive cold flow. The coefficient of thermal expansion for aluminum bus is 12.8×10^{-6} inches/inches/°F and Young's Modulus is 10×10^{6} psi.

Special cleaning and coating requirements for aluminum prevent the formation of a highly resistant aluminum oxide film on the exposed metal.

6.12.3 Aluminum Bolts

Aluminum bolts are recommended for use with aluminum bus in order to achieve the same coefficients of thermal expansion, 12.8×10^{-6} inches/inches/°F. The tensile strength (62 ksi) and yield strength (40 ksi) of aluminum bolts (2024-T4) are comparable to those of Grade 2 low carbon steel bolts of the same size. Aluminum bolts are often lubricated to prevent galling during assembly.

6.12.4 Bronze Bolts

Bronze bolts (ASTM F-486 No. 651) are ideal for use with copper bus. The two materials have nearly the same coefficients of thermal expansion. The tensile strength of bronze bolts is 75 ksi, and the yield strength is 35 ksi.

6.12.5 Low Carbon Steel Bolts

Low carbon steel bolts (ASTM A-307 Grade A) are not recommended for power connections. However, the bolts may be acceptable for low power or instrument connections in which no significant thermal heating is produced. Low carbon steel bolts have a tensile strength of 60 ksi and a yield strength of 36 ksi. During installation, these bolts could be stressed to the yield point, with additional thermal stress resulting in the plastic deformation.

6.12.6 High Strength Steel Bolts

High strength steel bolts (ASTM A-325, ASTM A-490, and SAE Grade 5) are recommended for use with copper bus or aluminum bus because of the bolts' low cost. Coating these bolts with zinc reduces corrosion. Because high strength steel bolts have a relatively low coefficient of thermal expansion (6.5×10^{-6} inches/inches/°F), Belleville washers may be used in conjunction with the bolts. The tensile strength of these bolts is 120 ksi, and the yield strength is between 80 ksi and 92 ksi.

6.12.7 Stainless Steel Bolts

The coefficient of thermal expansion for austenitic stainless steel bolts (ASTM A-193 Classes 1 and 2) is comparable to that of copper bus (9.3×10^{-6} inches/inches/°F). These bolts are suitable for corrosive environments. Class 1 bolts have a yield strength of 30 ksi and a tensile strength of 75 ksi. Thus, the bolts will approach or enter the plastic region due to installation stress. Class 1 bolts are not recommended for use with aluminum bus. Class 2 bolts have a yield strength of 80 ksi and a tensile strength of 125 ksi. These bolts are appropriate for any application. However, Class 2 bolts are more expensive than Class 1 bolts.

7.0 Assembly

7.1 Cleanliness and Surface Preparation for Electrical Joints

7.1.1 Silver Plated and Tin Plated Conductors

Little preparation is required for aluminum and copper conductors plated with silver or tin. Cleaning the conductor with a solvent to remove any dust or other foreign material is sufficient. Abrading or cleaning with a wire brush removes the plating and should be avoided. An insert sealing compound can be used to prevent air and moisture from entering the joint.

7.1.2 Aluminum Conductors

In preparing an aluminum conductor for assembly, first clean the conductor of dirt or foreign material. Apply a chemically active sealing compound and abrade the surface using:

- Fiberglass brush
- Emery cloth or emery paper
- Wire brush
- Draw file
- Steel wool

After the surface has been abraded, the contact surfaces can be bolted.

7.1.3 Copper Conductors

Copper conductors should be abraded using the same method described in Section 7.1.2. Assemble copper conductors by using an inert sealing compound.

Sealing compounds containing suspended metallic particles are intended for use with aluminum cable to aid in penetration of oxide film. Appropriate sealing compounds provide significantly lower joint resistance in copper conductors.

7.2 Electrical Bolted Joint Assembly

In assembling bolted electrical connections, large, flat washers should be used on both sides of the bus. In special cases or in cases in which reliability is critical, Belleville washers may be used in place of the large, flat washers. Figures 7-1 and 7-2 are assembly diagrams of bolted electrical connections employing large, flat washers and Belleville washers.

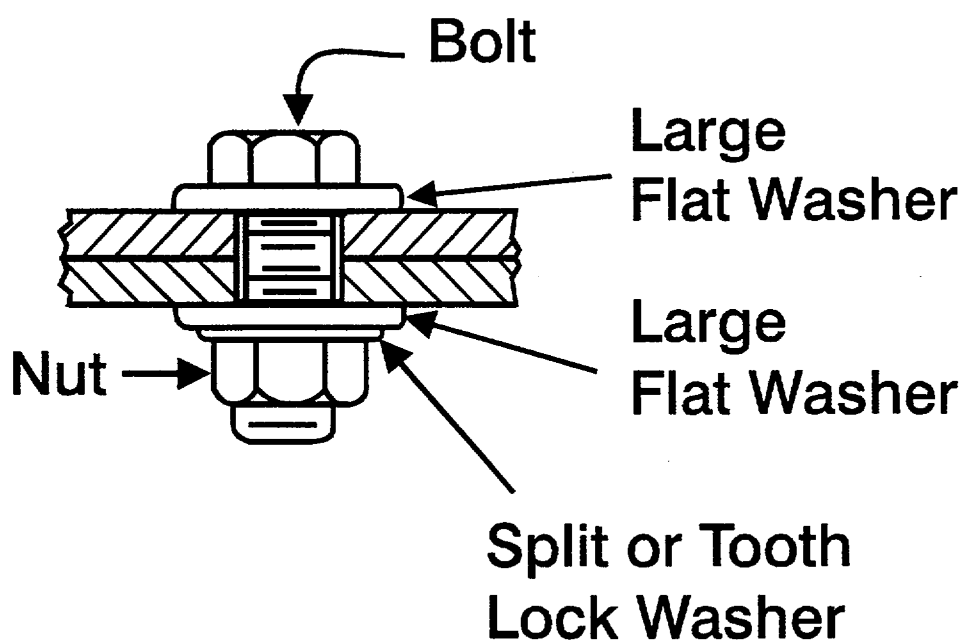


Figure 7-1 — Bolted Electrical Hard Drawn Bus Joint.

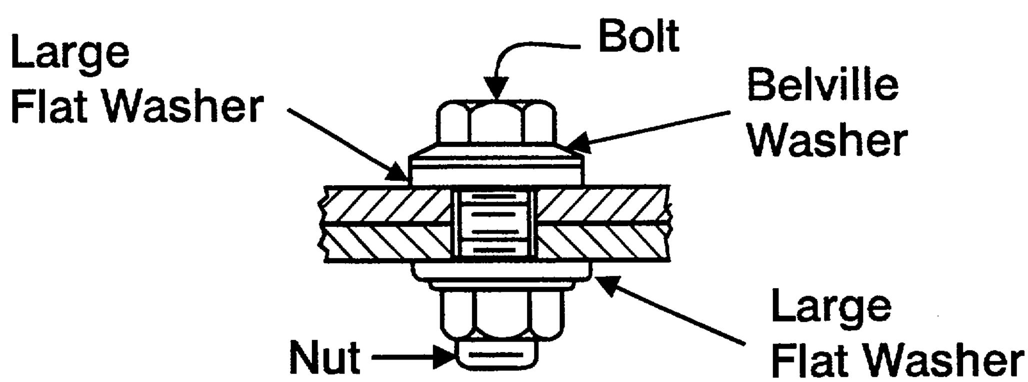


Figure 7-2 — Bolted Electrical Joint with Belleville Washer.

7.2.1 Bolt Torquing

A torque wrench or other torque limiting tool is used to assemble electrical joints. The bolt is tightened until the required torque is achieved. If the procedure is stopped prior to achieving the required torque, the breakaway torque will be significantly higher and will not represent the actual joint tightness.

7.2.2 Special Problems

Increased temperature difference in electrical bolted joints is due to high short circuit ratings or increased current duration. The temperature of an electrical bolted joint will rise and the stress will increase with increasing current duration. If this temperature increase is not taken into consideration, loose, failure prone joints will result. Belleville washers selected to be flat or almost flat at the installation torque may be used to accommodate the temperature increase. In extreme cases, using an oversized bus should be considered.

Hydrogen embrittlement is a recurring problem with Belleville washers and other springs. When springs are electroplated, the plating process forces hydrogen into the metal grain boundaries. If the hydrogen is not removed, the spring may spontaneously fail at any time while inservice. Although hydrogen embrittlement occurs infrequently, it is recommended that electroplated Belleville washers and other springs not be used. Hydrogen embrittlement does not affect Belleville washers having other finishes.

7.2.3 Terminal Blocks and Pressure Terminals

Equipment such as molded case circuit breakers, starters, and overhead relays, is marked with the correct installation torque. Tables 7-1 through 7-3 provide installation torque for unmarked equipment. For values of slot width or length not corresponding to those specified in Table 7-1, use the larger torque value for the given conductor size.

7.2.4 Torque for Electrical Bus Bars

Tables 7-4 through 7-8 give recommended torque, bolt stress, and bus stress for various combinations of bolt and bus materials. The operating conditions assumed in developing these tables are:

> Assembly temperature: 70°F
> Operating maximum allowable hot spot temperature: 221°F
> Short circuit or overload temperature: 297°F

The total temperature change (ΔT) from assembly to overload is 227°F.

Table 7-1*			
Tightening Torque for Screws			
Test Wire Size Installed in Terminal (AWG or MCM)	Tightening Torque for Slotted Head Screws No. 10 and Larger (in-lb)		Tightening Torque using Hexagonal Head External Drive Socket Wrench (in-lb)
	Slot Width ≤ 0.047 in Slot Length ≤ 1.4 in	Slot Width > 0.047 in Slot Length > 1.4 in	
22-10	20	35	75
8	25	40	75
6-4	35	45	110
3	35	50	150
2	40	50	150
1	-	50	150
1/0-2/0	-	50	180
3/0-4/0	-	50	250
250-400	-	50	325
500-750	-	50	375
800-1000	-	50	500
1250-2000	-	-	600

Slot width is the nominal design value. Slot length is measured at the bottom of the slot.

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-2*		
Tightening Torque for Slotted Head Screws Smaller than No. 10		
Slot Length of Screw (in)	Tightening Torque for Slot Width of Screw ≤ 0.047 (in-lb)	Tightening Torque for Slot Width of Screw > 0.047 (in-lb)
< 4.32	7	9
5.32 - 7.32	7	12
1.4	9	12
9.32	-	15
> 9.32	-	20

Data is for use with No. 8 AWG and smaller conductors. Slot lengths are measured at the bottom of the slot. For values of slot length which are between those specified in Table 7-2, the torque corresponding to the next shorter length should be used.

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-3*	
Tightening Torque for Recessed Socket Screws	
Socket Size Across Flats (in)	Tightening Torque (in-lb)
1/8	45
5/32	100
3/16	120
7/32	150
1/4	200
5/16	275
3/8	375
1/2	500

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-4*							
Bronze Bolts Used with Copper Bus Bolts: ASTM F-468 No. 651 Copper Bus: ASTM B-187 Temper HO4							
Bolt Size (in)	Torque		Installed Tensile Load (lbs)	Installed Stress (ksi)	Thermal Stress (ksi)	Total Bolt Stress (ksi)	Bus Stress (ksi)
	ft-lb	in-lb					
1/4-20	5	60	1200	37.73	0	37.74	5.21
5/16-18	10	120	1920	36.64	0	36.64	7.36
3/8-16	20	240*	300	41.29	0	41.29	7.12
1/2-13	40	480*	4800	33.83	0	33.83	4.90
5/8-11	55	660*	5280	23.36	0	23.36	3.77

All stresses are within the elastic range (less than the 45 ksi yield strength for bronze bolts and the 40 ksi yield strength for copper bus).

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-5*							
High Strength Steel Bolts Used with Copper Bus Bolts: SAE Grade 5, ASTM A-325, ASTM A-490 Copper Bus: ASTM B-187 Temper HO4							
Bolt Size (in)	Torque		Installed Tensile Load (lbs)	Installed Stress (ksi)	Thermal Stress (ksi)	Total Bolt Stress (ksi)	Bus Stress (ksi)
	ft-lb	in-lb					
1/4-20	5	60	1200	37.73	15.72	53.46	7.34
5/16-18	12	144	2304	43.97	14.52	58.49	11.79
3/8-16	20	240	3200	41.29	15.05	56.34	9.70
1/2-13	50	600	6000	42.23	15.59	57.82	8.37
5/8-11	95	1140	9120	40.35	15.26	55.61	8.97

All torque values are for zinc plated bolts.

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-6*							
Austenitic Stainless Steel Bolts Used With Copper Bus Bolts: ASTM A-193 Type 1, Class B8A Copper Bus: ASTM B-187 Temper HO4							
Bolt Size (in)	Torque		Installed Tensile Load (lbs)	Installed Stress (ksi)	Thermal Stress (ksi)	Total Bolt Stress (ksi)	Bus Stress (ksi)
	ft-lb	in-lb					
1/4-20	5	60	1200	37.74	0	37.73	5.21
5/16-18	10	120	1920	36.64	0	36.64	7.36
3/8-16	20	240	3200	41.29	0	41.29	7.12
1/2-13	40	480	4800	33.83	0	33.83	4.90
5/8-11	55	6600	5280	23.36	0	23.36	3.77

Bolts of most sizes are stressed into the plastic range (over the 30 ksi yield strength but less than the 75 ksi tensile strength). This is acceptable, because no additional thermal stresses are present.

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-7*							
Aluminum Bolts Used with Aluminum Bus Bolts: ASTM Aluminum, Grade 2024-T Aluminum Bus:							
Bolt Size (in)	Torque		Installed Tensile Load (lbs)	Installed Stress (ksi)	Thermal Stress (ksi)	Total Bolt Stress (ksi)	Bus Stress (ksi)
	ft-lb	in-lb					
3/8-16	15	180	3428	44:23	0	44.23	7.63
1/2-13	25	300	6857	48.32	0	48.32	7.02
5/8-11	40	480	5485	24.27	0	27.27	3.91

All torque values are for lubricated bolts.

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-8*							
High Strength Steel Bolts Used with Aluminum Bus Bolts: SAE Grade 5, ASTM A325, ASTM A-490 Aluminum Bus: ASTM B-317 Grade 6061-T61							
Bolt Size (in)	Torque		Installed Tensile Load (lbs)	Installed Stress (ksi)	Thermal Stress (ksi)	Total Bolt Stress (ksi)	Bus Stress (ksi)
	ft-lb	in-lb					
1/4-20	5	60	1200	37.74	029.48	67.22	9.41
5/16-18	12	144	2304	43.96	26.14	70.10	14.13
3/8-16	20	240	3200	41.29	27.59	68.88	11.86
1/2-13	50	600	6000	42.28	34.10	76.38	11.10
5/8-11	95	1140	9120	40.35	28.16	68.51	11.05

All torque values are for zinc plated bolts.

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

7.3 Assembly of Miniature Fasteners

Proper assembly tools and control of key assembly variables ensures acceptable fastener performance. The deflections of threads, bearing areas, and clamp affected zones are extremely small during installation and tightening of miniature fasteners. Therefore, the tolerance and control of pitch diameter, head diameter, and driven depth are critical. If problems are anticipated in these areas, apply appropriate tolerances and statistical process controls to maintain strict compliance. Tables 7-9 and 7-10 provide assembly torque information for small bolts.

Table 7-9*						
Small Bolt Assembly Torque						
Screw Size	Nominal Diameter (in)	Threads per Inch	Thread Root Diameter (in)	Root Area (in^2)	Torque at 30 ksi Stress (in-lb)	Torque at 45 ksi Stress (in-lb)
0	0.060	80	0.045	0.00150	0.46	0.69
1	0.0743	64	0.054	0.00218	0.81	1.22
2	0.086	56	0.064	0.00310	1.36	2.04
3	0.099	48	0.073	0.00406	2.05	3.08
4	0.112	40	0.081	0.00496	2.84	4.25
5	0.125	40	0.094	0.00672	4.28	6.43
6	0.138	32	0.099	0.00745	5.24	7.86
8	0.164	32	0.126	0.01195	10.00	15.00
10	0.190	24	0.138	0.01149	14.04	21.06
12	0.216	24	0.165	0.02057	22.66	33.99

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

Table 7-10*					
Assembly Torque					
Nominal Pipe Diameter (in)	Threads per Inch	Thread Root Diameter (in)	Thread Root Area (in^2)	Torque at 30 ksi Stress (in-lb)	Torque at 45 ksi Stress (in-lb)
1/4	20	0.185	0.027	4	6
5/16	18	0.240	0.045	8	12
3/8	16	0.294	0.068	12	18
7/16	14	0.345	0.093	20	30
1/2	13	0.400	0.125	30	45
9/16	12	0.454	0.162	45	68
5/8	11	0.507	0.202	60	90
3/4	10	0.620	0.302	100	150
7/8	9	0.731	0.419	160	240
1	8	0.838	0.551	245	368
1 1/8	8	0.963	0.728	355	533
1 1/4	8	1.088	0.929	500	750
1 3/8	8	1.213	1.155	680	1020
1 1/2	8	1.338	1.405	800	1200
1 5/8	8	1.463	1.680	1100	1650
1 3/4	8	1.588	1.980	1500	2250
1 7/8	8	1.713	2.304	2000	3000
2	8	1.838	2.652	2200	3300
2 1/4	8	2.088	3.423	3180	4770
2 1/2	8	2.388	4.292	4400	6600
2 3/4	8	2.588	5.259	5920	8880
3	8	2.838	6.324	7720	11580

**Reference EPRI NP5067 "Good Bolting Practices", Volume 2.*

7.4 Machine and Mechanical Joints

The most common assembly method for machine joints is torque control. Chapter 3, Sections 3.2 and 3.7 discuss factors that affect the preload achieved when torque control is used to assemble a bolted joint. If reasonable care is exercised, a preload accuracy of approximately ± 30% can be achieved from torque control. Most machine joints are designed such that preload control of ± 30% is sufficient for successful performance of the joints. The following factors must be controlled:

- Thread fit, surface finishes, and material hardness
- Selection and application of lubricants
- Selection of nut factor.

7.5 Stretch Control

Measurement of fastener stretch is often used to control and monitor preload for critical applications. Stretch measurements require measurement tools such as micrometers, depth micrometers, datum rods, or ultrasonic stretch measuring devices. Chapter 3, Sections 3.4 and 3.5 discuss stretch control of preload.

7.6 Turn of Nut

In the turn of nut method, the fastener is strained as the nut is turned against the joint. Turn of nut assembly is most effective when the fastener is taken beyond the yield stress and the procedure is determined empirically. This method may be used to load fasteners in the elastic range, but the procedure must be carefully tested.

The American Institute of Steel Construction (AISC) recommends turn of nut method for structural joints using A325 and A490 bolts. The AISC assembly procedures where developed by tests and are designed to produce a minimum of 70% of tensile stress in the bolts which is above the yield stress.

7.6.1 Model for Turn of Nut

Two models are suggested for calculating loads from turn of nut method. These models are the lead screw formula and the joint stiffness model.

7.6.1.1 Lead Screw Formula

If the joint members are infinitely stiff all of the turn goes into bolt stretch. As the nut is advanced one revolution, the fastener is stretched by an amount equal to the thread pitch. The equation in relating stretch to turn is given as:

$$\Delta L = \alpha p \;/\; 360^\circ \tag{7-1}$$

Rewriting Equation 7-1 for preload (F_p)

$$F_p = (\Delta L \ / \ L)AE \qquad (7\text{-}2)$$

where, L = Original unstressed length of fastener (inches)
α = Angle of turn (degree)
p = Thread pitch (inch per thread) (p = 1/n where n is threads per inch)
A = Cross sectional area of fastener (inches^2)
E = Modulus of elasticity (psi)

This equation ignores the relative stiffness of the joint elements. Notch (32) indicated that joint stiffness is an important parameter in preload development using turn control. When the joint system is soft with many plies or when thin washers are used on oversized holes, more turn is required to produce a given load. A soft make up can require twice the turn as a hard make up.

7.6.1.2 Stiffness Formula

This formula accounts for the stretch of the bolt and compression of the joint as the nut is turned.

$$F_p = (K_B \, P \, \alpha \ / \ 360°) \ / \ (L + K_B \ / \ K_J) \qquad (7\text{-}3)$$

where, K_B = Fastener Stiffness (lbs/inch)
K_J = Joint stiffness (lbs/inch)
p = Thread pitch (inch per thread)
α = Angle of turn (degree)

The stiffness for the fastener (K_B) may be calculated using Hooke's law (see Section 3.4 or Section 6.2.6.2). The joint stiffness (K_J) may be calculated using equations of Section 6.2.6.2.

8.0 Inspection

8.1 Inspection of Mechanical Joints

Table 8-1 provides guidance on quality control inspections for mechanical joints. These guidelines are recommendations to be used in conjunction with experience-based inspections.

Table 8-1

Quality Control Guidelines

Activity or Equipment	Key Inspections
General Disassembly	• Parts, components, nuts, and studs are properly stored, bagged, and tagged. • Machine or sealing surfaces are protected during storage. • Open systems, machinery, and components are protected from damage, dirt, and foreign material. • Internal parts are inspected and evaluated for normal or abnormal wear patterns, steam cuts, etc. • Leak indicators (e.g. boric acid, lube oil buildup, etc.) are examined in an "as found" inspection. • Rigging adheres to procedural requirements.
Pumps: Operating Inspections	• Foundation bolt integrity • Fluid leakages • Vibration levels • Operating bearing temperatures • Degradation of fasteners • Seal indications of pump/driver misalignment • Lube oil leakage.
Bolting, Studs, and Fasteners	• All surface areas, especially the thread root area, are examined for evidence of corrosion or erosion, cracking, galling, pitting, and mechanical damage. • Assemblies are inspected for proper thread engagement, correct size, proper lubricant, and torque values, where specified. • Code material requirements, bolt and nut markings, heat numbers, and material identification are examined. NOTE: For installed fasteners, only those areas which are accessible require inspection.

8.2 Inspection of Electrical Bolted Joints

Inspect bolted joints for evidence of overheating, signs of burning or discoloration, and indications of loose bolts. The bolts should not be retorqued unless the joint requires service or the bolts are clearly loose. Verifying the torque is not recommended. The torque required to turn the fastener in the tightening direction (restart torque) is not a good indicator of the preload once the fastener is in service. Due to relaxation of the parts of the joint, the final loads are likely to be lower than the installed loads. However, this load reduction has little effect on electrical conductivity or joint performance. Check the joint resistance of bolted joints using a low range ohm meter.

9.0 Troubleshooting and Repair

9.1 Analysis of Failure Modes

Table 9-1 identifies common failure modes for mechanical joints and the appropriate corrective actions. A discussion of fatigue failures and loosening failures is given in the following subsections.

Table 9-1

Mechanical Joint Failures and Corrective Actions

Failure Modes and Causes	Corrective Action
Bolt yielding • Overtightening; overloads while in service	Improve assembly specifications and control, improve joint design, and evaluate size and number of fasteners.
Thread stripping • Soft nuts; short thread length; shallow threads	Use Grade 10 nuts; thread length 1.00D in steel, 1.25D with stud, 1.50D with cap screw in cast iron, and 2.00D in aluminum; 55% - 65% thread depth; coarse threads in cast iron and non-ferrous materials.
Shear failures • Transverse loads which act on shear planes	Increase clamp loads to increase friction; use bushings to carry shear loads on pivot joints; design for shear through the body of the bolt, not the threads; use larger bolts.
Fatigue failures • Low clamp load with high cyclic stress	Wrench to a higher percentage of bolt strength; use larger bolts.
• Stress concentration at the radii under the head, under the first thread, or under first thread under the load	Use proper radii under the head and at the root of the thread; more threads (3-6) in the grip length; roll the threads after heat treatment.
• Bending stresses which increase the stress at the concentration points	Increase the clamp load to reduce the bending stresses; increase the flexibility of the bolt by using a smaller diameter or longer length.
Loosening of nut • Axial vibration • Self loosening by friction changes	Increase the clamp load; use a locking device on the bolt or the threads.

Table 9-1	
Mechanical Joint Failures and Corrective Actions	
Failure Modes and Causes	**Corrective Action**
Wearing of surface • Transverse vibration	Increase the clamp load; decrease the bearing surface stress.
Embedment of bearing surfaces • High compressive stresses on soft joints	Increase the bearing surface area with a head bolt or nut, closer clearance holes, or a washer; use harder joint materials.
Loss of clamp force on early loading • High localized stresses and crushing of surface roughness	Bearing surface of fastener and joint interfaces, spot the face; flatten; clean the dirt; mask the paint from the surface; check the gaskets.

9.2 Fatigue Failure

9.2.1 General Discussion

Failure may be due to fatigue when:

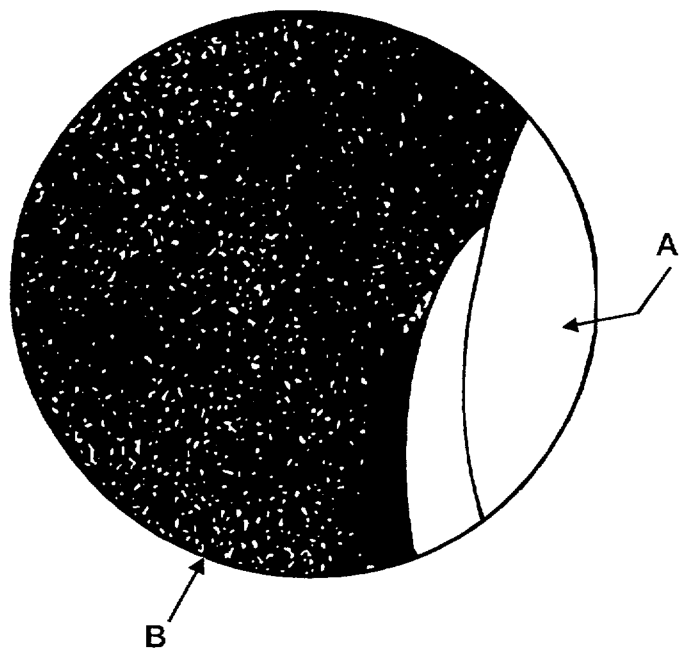
- The failure is sudden, with little or no "necking down" of the part.
- The component has been subjected to cyclic tensile loads.
- Cyclic loads are below the material tensile strength.

Identify fatigue failures by observing the appearance of the fracture surface. The surface may bear beach marks, polished areas, or corrosion. Figure 9-1 depicts the typical appearance of fatigue failures. Fatigue failure depends on many factors, including the fastener material properties, the manufacturing process, defects in the fastener, stress levels, and the fastener shape (i.e., thread radius or fillet radius).

9.2.2 Preventing and Analyzing Fatigue Failure

Fatigue failure is a complex subject involving many variables. Recurring problems are best solved by a team of professionals. This team may include a metallurgist, fastener manufacturer, design engineer, and assembly expert. Prevent fatigue failure by:

- Checking the preload and torque specifications. Rapid fatigue failure results if preloads are too low. If preload is so low that the external load causes the joint to unload, then the bolt will carry a larger portion of an external load. In addition, prying loads are increased. This step includes a review of assembly tools and procedures as well as relaxation effects.



A = Failure During Crack Initiation and Growth - Smooth and Shiny Surface
B = Rapid Failure - Rough Surface

Figure 9-1 — Cross Section of Fatigue Failure.

- Inspecting fasteners before assembly and replacing defective fasteners. Fatigue cracks can start at minor defects, pits, cracks, or folds in bolt surfaces. Examine fastener surfaces visually, or use dye penetrant or magnaflux. Use ultrasonic techniques to detect cracks in fasteners both before and after assembly. Decarburized fastener surfaces are easily cracked, encouraging fatigue. Ensure that bolts have been properly heat treated.

- Checking the bolt perpendicularity. Check the perpendicularity between the nut face and the bolt axis. Tests show that an angularity of nut face to bolt axis of 4° reduces fatigue life to practically zero. Possible solutions include spot facing or milling the flange surface and using tapered washers or self aligning nuts. Figure 9-2 demonstrates these principles.

- Replacing bolts periodically. For life-limited fastener design, replacing fasteners on a prescribed service-interval basis can be employed to prevent service failures.

- Ensuring that proper fastener materials are used. Material characteristics that may increase fatigue life include:
  - — Ultimate tensile strength of 160 ksi. Ensure that material is resistant to stress corrosion cracking.
  - — 7% minimum elongation.
  - — Through hardened material. Examples include alloy steel such as AISI 4340 or 4140.
  - — High notch strength.

One bolt material with these characteristics is AISI 4340 steel heat treated to 160 ksi tensile strength. Yield strength is 145 ksi.

- Ensuring proper nut and bolt shape. Minor shape differences can have significant effects on fatigue. The following characteristics affect the fatigue life:
  - — Thread root radius. Large, smooth radii have longer fatigue life than sharp roots.
  - — Thread type. Threads rolled after heat treatment have a longer fatigue life than do machined threads.
  - — Fillet shape between the head and the body
  - — Blending of the threaded section with an unthreaded body
  - — Shape of the bolt head
  - — Nut shape. This includes using flanged nuts with tapered threads at the first thread of engagement.
  - — Nut length.

- Eliminating unnecessary stress concentrations. If the thread run out is too close to the nut bearing surface, the thread stresses at the first thread of engagement of the nut will increase.

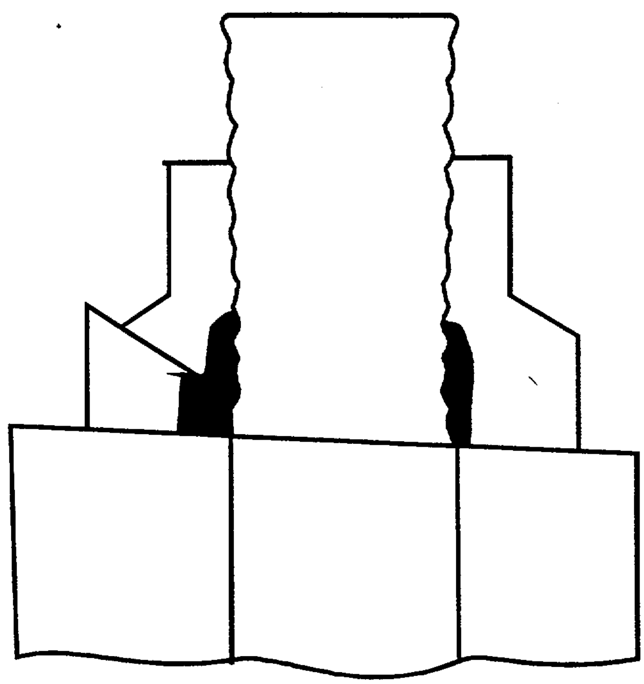


Figure 9-2 — Cross Section of Two-Piece Spherical Nut Showing How it Conforms to a Joint Surface Which is Not Perpendicular to the Bolt Axis.

If fatigue failure does occur, preserve and protect all broken parts. Examination of a broken bolt can provide the following information:

- When and where the failure started, as revealed by quench cracks, inclusions, and pitting.
- A time history of the crack progression, as revealed by beach marks, corrosion products and polishing of the surface.
- The external load on the fastener, as revealed by the final overload area.

9.2.3 Fatigue Tests

Fatigue tests are not accurate predictors of fatigue life since the tests are normally conducted on polished test coupons. The actual fatigue life of a fastener is usually one half to one fourth the life of the test coupon. The reduced life span is due to stress concentrations, head to body fillets, processing defects, and the environment. Qualified laboratories can conduct fastener fatigue tests under conditions that simulate the applications as closely as possible.

In field situations, it is usually impossible to reduce the loads on bolts. These loads are a function of the operating temperature and pressure, the equipment design, and the joint design. Equipment should be designed so that the amplitude of cyclic stress is lower than the endurance limit (refer to Section 6.3.3).

9.3 Vibration Loosening

9.3.1 Conditions for Loosening

Vibration can loosen bolts and often causes a complete loss of the nut and bolt. While various explanations are given for bolt loosening, none fully explain all of the problems encountered in the field. The following conditions typically exist when a nut is fully loosened under vibration:

- The vibratory force has components at right angles to the axis of the bolts. This force causes the joint members to slip past each other. Vibratory forces parallel to the bolt axis will partially loosen the bolt, causing a loss of 10% - 20% of the initial preload. Only right angled forces will fully loosen the nut.
- There is slip clearance between the male and females threads and between the bolts and joint members. This allows transverse slip to occur.

If the above conditions exist, severe slip cycles (e.g., thermal cycles, flexing of the structure, flexing of joint members) may fully loosen the nut. More commonly, hundreds or thousands of vibration induced micro-slip motions can fully loosen a nut.

9.3.2 Minimizing Loosening

A number of techniques minimize self loosening. The following methods are listed in order of increasing cost or complexity:

- Increase thread and joint friction forces. If these forces are high enough, no transverse slip will occur. Friction can be increased by the following:
  - Eliminating thread lubricants
  - Increasing preloads
  - Compensating for the relaxation of the fasteners; for example, use an extra pass with the wrench. This increases residual preload and results in higher friction forces.

- Use anaerobic adhesives to "glue" the male and the female threads together. These are available for operating temperatures up to 450 °F.

- Add collars under the bolt head and nut(s) and use longer bolts. Some practitioners say a bolt with a length to diameter ratio of 8:1 or more will "never" loosen.

- Use bolts with fine threads instead of coarse threads. The slope of the helix angle for fine threads is smaller than that for course threads. This smaller slope helps to reduce loosening.

- Use vibration resistant nuts such as those with a nylon locking collar or interference fit threads. Figure 9-3 is an illustration of a nylon collar nut.

- Pin or tack weld joint members to prevent relative slip. Dowel pins or interference body bolts can be used.

- Reduce the amount of vibration in the joint. Some ways to do this include:
  - Adding stiffeners to joint members.
  - Adding mass to the joint to change natural frequencies of the assembly.
  - Using shock absorbers and/or shock mounts to dampen the vibration.

- Have a vibration specialist evaluate the system to determine the causes and possible cures for the vibration.

- Reshape joint members to prevent relative slip. See Figure 9-4.

- Redesign the joint so that the axis of the fasteners is parallel to the direction of vibration rather than perpendicular to it.

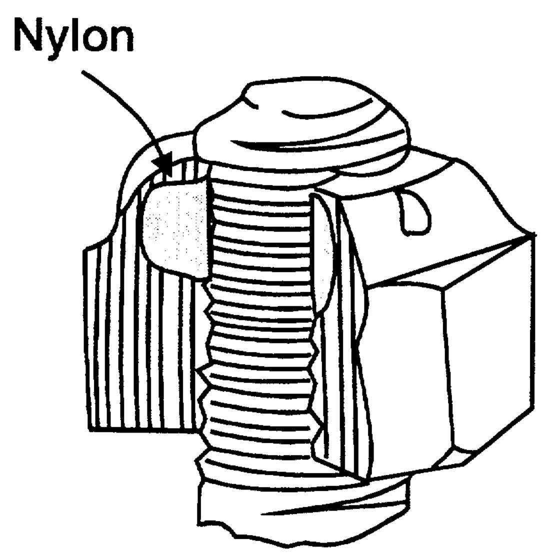


Figure 9-3 — Vibration Resistant Nut with a Nylon "Collar".

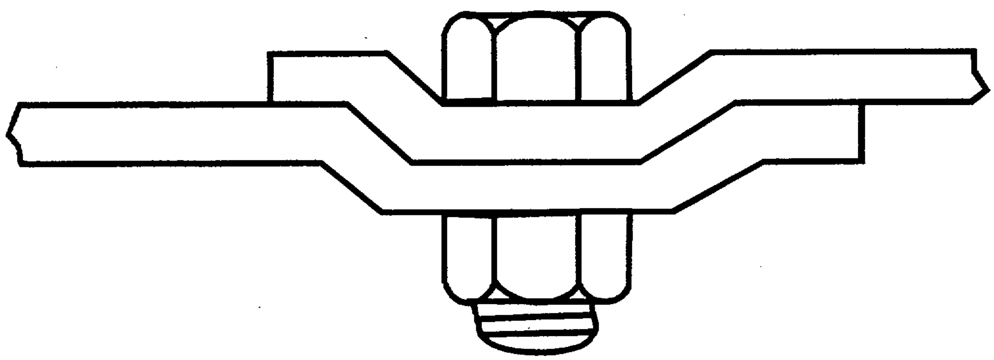


Figure 9-4 — The Joint Members have been Shaped to Resist Transverse Slip.

9.4 Thread Stripping

Fastener threads will strip when the axial forces on the fastener exceed the shear strength of the male or female threads. The main factors determining stripping strength include:

- The strength of the fastener material
- The length of engagement of the threads
- The size of the fastener.

A number of secondary factors also affect stripping strength. These are discussed in Section 9.4.5.

9.4.1 Material Strength

The shear strength of most fastener materials is approximately 50% to 60% of the ultimate strength. The ANSI B1.1 specification assumes that the shear strength is 50% of the tensile strength. The shear strength of cast iron and cast aluminum is quite low and depends on the quality of the casting. Caution should be exercised in choosing a strength.

9.4.2 Length of Engagement

Increasing the length of engagement between male and female threads increases the cross sectional area of the material which must be sheared to strip the threads. As a result, heavy hex nuts or deep tapped holes are less likely to strip than are regular nuts or shallow holes. Length of engagement does not affect the stripping strength as much as might be expected, however. The first threads of engagement carry most of the load transmitted from the male to the female threads. Therefore, adding more lightly loaded threads at the other end of the nut does not increase stripping strength significantly.

Table 9-2 provides thread stripping areas for a length of engagement equal to one diameter of the fastener. For example, a 1 1/4-8 Class 2A thread requires a length of engagement equal to about three fourths of the fastener diameter to develop the full strength of the body. As a result, a heavy hex nut with a height equal to the diameter should never strip. Table 9-3 provides minimum length of thread engagement for blind holes. These are recommended values so one should always test critical applications.

Table 9-2*		
Stripping Areas for External Threads		
Thread	Stripping Area Calculated in Accordance with ANSI B1.1 Requirements (Inches^2 per Length of Engagement Equal to One Nominal Diameter)	
	Class 2A Thread	Class 3A Thread
1/4-20	0.092	0.096
5/16-18	0.147	0.157
3/8-16	0.216	0.232
7/16-14	0.296	0.321
1/2-13	0.390	0.427
9/16-12	0.502	0.548
5/8-11	0.624	0.681
3/4-10	0.908	1.01
7/8-9	1.25	1.38
1-8	1.66	1.82
1 1/8-8	2.13	2.329
1 1/4-8	2.65	2.913
1 3/8-8	3.22	3.55
1 1/2-8	3.86	4.26
1 5/8-8	4.55	5.04
1 3/4-8	5.30	5.03
1 7/8-8	6.09	6.81
2-8	6.96	7.72
2 1/4-8	8.84	9.83
2 1/2-8	10.95	12.18
2 3/4-8	13.28	14.80
3-8	15.84	17.67

Table 9-2*		
Stripping Areas for External Threads		
Thread	Stripping Area Calculated in Accordance with ANSI B1.1 Requirements (Inches^2 per Length of Engagement Equal to One Nominal Diameter)	
	Class 2A Thread	Class 3A Thread
3 1/4-8	18.62	20.8
3 1/2-8	21.63	24.15
3 3/4-8	24.79	27.79
4-8	28.28	31.64

**Reference EPRI NP5067 "Good Bolting Practices".*

Table 9-3 (21)					
Minimum Length of Thread Engagements for Blind Holes					
Screw Property Class		8.8	8.8	10.9	10.9
Thread Fineness Factor (DIA/Pitch)		<9	≥9	<9	≥9
DIN	Type of Material				
AlCuMg 1 F 40	Light alloys	1.1 d	1.4 d		-
GG-222	Cast (grey) iron	1.0 d	1.2 d		1.4 d
ST 37	Low carbon steel	1.0 d	1.25 d		1.4 d
ST 50	Medium carbon steel	0.9 d	1.0 d		1.2 d
C 45 V	Heat treated steel (AISI 1045)	0.8 d	0.9 d		1.0 d

Use only as guidelines. Test for critical applications.

9.4.3 Size of the Fastener

The threads on a large fastener are "longer" per turn than the threads of a small fastener. In addition, the section through the shear plane of threads is larger. This means that the thread area that resists stripping is greater on the larger fastener. As a result, larger fasteners have greater stripping strength.

Doubling the diameter of a fastener approximately doubles the stripping strength per inch of thread engagement. Doubling the diameter also typically doubles the length of engagement. The industry standard is a nut height or tapped hole depth equal to one nominal diameter. Stripping strength approximately quadruples as fastener diameter is doubled. Doubling the diameter also increases the tensile strength of the body by a factor of 4. The stripping strength keeps pace with the tensile strength as the nominal size is changed.

9.4.4 Stripping Strength Equations

Both external and internal male and female threads can strip along several different cross sections. The pattern depends on the relative hardness of the nut and bolt materials, the tightness of fit, etc. The equations and thread dimensions for thread stripping calculations can be found in the ANSI B1.1 Standard or in Machinery's Handbook (Industrial Press, Inc., New York).

An estimate of the load required to strip the threads can be calculated by multiplying the shear strength of the material by the stripping area tabulated in Table 9-2. Note that these areas are for external threads and for a length of engagement equal to one nominal diameter of the fastener. The tabulated areas are for UNC or UN threads. The stripping strength of other threads can be estimated by using:

- The stripping areas of female threads are 1.3 to 1.5 times the areas of the mating male threads. The multiplier is smaller for larger fasteners.
- The stripping areas of fine pitch external threads (UNF) are approximately equal to that of coarser threads up through diameters of 1 7/8 inches. As diameters increase, the finer pitch (UN) become more vulnerable. At 4-inch diameter, the stripping area of the UNC thread is 5% greater than that of the UN thread.
- The stripping areas for fine pitch internal threads are 4% to 5% less than those for coarser threads for all sizes. The exception are those threads between 1 inch and 1 3/8 inches. In this range, the coarse and fine areas are essentially equal.
- Table 9-4 gives minimum mechanical properties for various tapped materials.

Table 9-4 (35)

Minimum Mechanical Properties for Various Tapped Materials

Material	Tensile		Shear	
	psi	MPa	psi	MPa
1/ SAE 1010 Hot Rolled	47,000	320	35,000	240
1/ SAE 1020 Hot Rolled	55,000	380	41,200	280
1/ SAE 1035 Hot Rolled	72,000	500	54,000	370
2/ ESE-M1A 104-A Grade B Grey Iron	18,000	125	23,000	160
2/ ESE-M1A 116-A and ESE-M1A 235-A Grade AC Grey Iron	30,000	210	39,000	270
3/ SAE 303, 306, 308, 309 Aluminum Die Casting	26,000	180	21,000	145

9.4.5 Secondary Factors

A number of other factors may also affect stripping length. These factors include the following:

- A tighter fit between male and female threads increases the stripping strength. When fasteners are coated, an allowance is made for the coating thickness. This can contribute to lower stripping strength.
- A thin-walled nut will strip more readily than a thick-walled nut due to the greater nut dilation.
- Lubricating the fastener before assembly increases the dilation and reduces the stripping strength slightly.
- Threads strip more readily when the bolt and nut materials are of equal strength. To ensure that the bolt will break before the nut threads strip, use a nut with a specified proof load 20% greater than the bolt ultimate tensile strength.
- Threads will strip more readily if the fastener is torqued (instead of tensioned) during assembly. Torquing also enhances dilation.

10.0 Design

10.1 Introduction

The majority of bolted structural joints are designed and assembled in accordance with the American Institute of Steel Construction (AISC) Specification for structural joints using A325 or A490 bolts. This specification includes the design and assembly of structural joints using ASTM A325 high strength carbon steel bolts and ASTM A490 high strength alloy steel bolts, or equivalent fasteners. While equivalent fasteners are loosely defined, the probable intent is fasteners with mechanical properties similar to the A490 and A325. The important mechanical properties of these fasteners are strength, ductility, and resistance to stress corrosion cracking (SCC).

Since AISC specification deals with only ASTM A325 and A490 materials, this specification does not provide guidance for high strength fasteners of other chemical compositions, mechanical properties, or sizes. This specification does not apply when materials other than steel are included in the joint. In addition, high strength anchor bolts are excluded from this specification.

10.1.1 Joint Types

Two types of joints are covered by the AISC specification. These are shear joints and tension joints. Figure 10-1 depicts two types of shear joints. Figure 10-2 is a typical tension joint.

Shear joints resist loads at right angles to the bolt axes. Shear joints are subdivided into two types, friction and bearing. Friction joints, also known as slip critical joints, transmit the shear loads by frictional resistance developed at the joint faying surfaces. Frictional forces depend on high bolt preloads and the coefficient of friction between the joint surfaces. Bearing joints, on the other hand, do not require high bolt preloads. The resistance to slip in bearing joints is provided by the bolts acting as pins. The bolts carry the load in shear and bear across the thickness of the joint plies.

A tension joint carries the load which is applied parallel to bolt axes. Tension joints are further subdivided into joints carrying static or cyclic loads.

10.1.2 Bolts

The A325 and A490 bolts are classified as high strength bolts. The tensile strength of A325 bolts is 105 ksi, while A490 bolts have a tensile strength of 120 ksi. The more important characteristic of these bolts is ductility. A325 and A490 bolts are preloaded to a minimum of 70% of the tensile strength. The bolts will likely be loaded above the yield point at assembly. Bolts must be ductile to prevent failure at assembly, see illustration in Figure 10-3 which shows preload versus ductility in a properly assembled structural joint and that the assembly results in very good preload control.

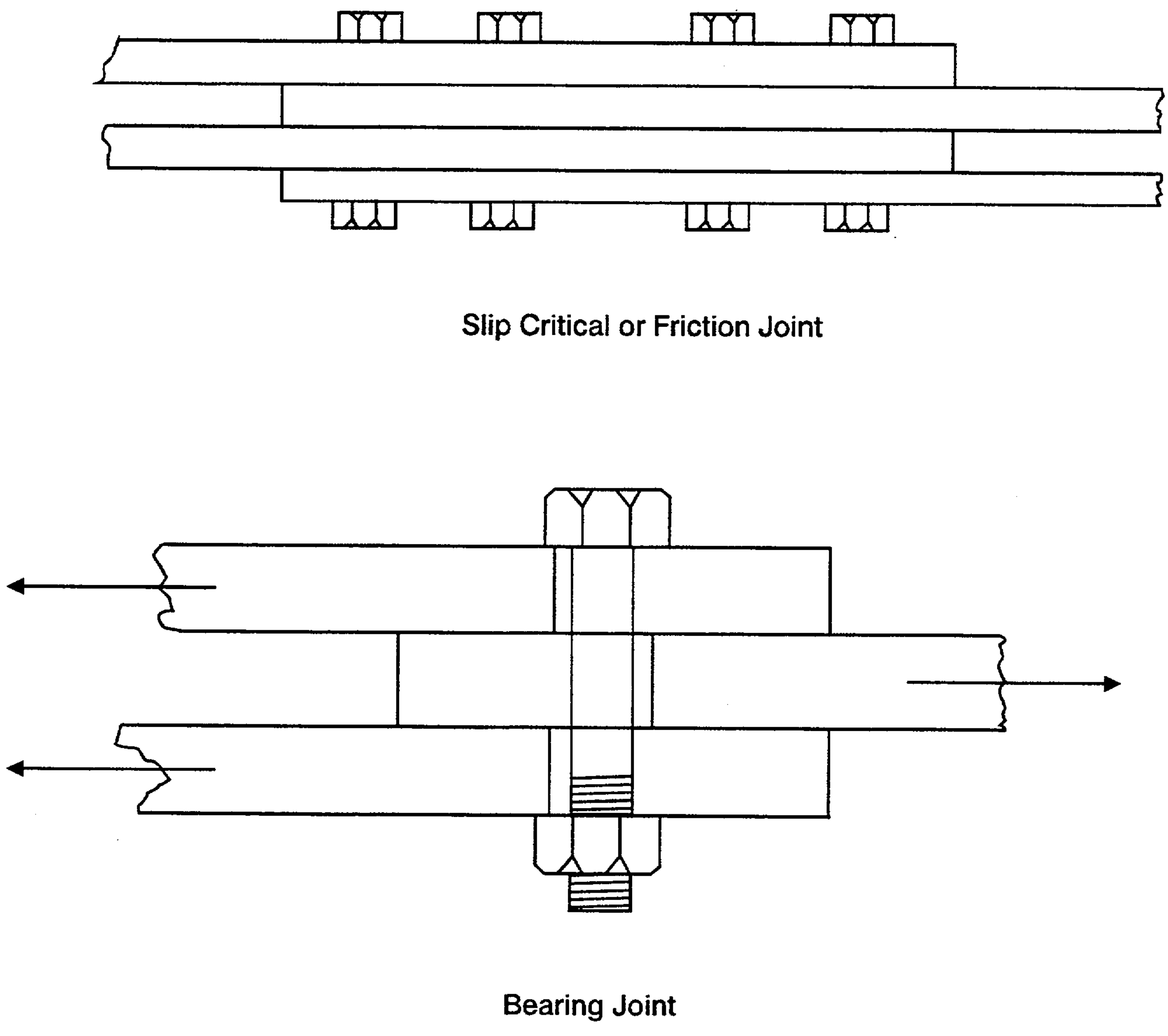


Figure 10-1 — Shear Joints.

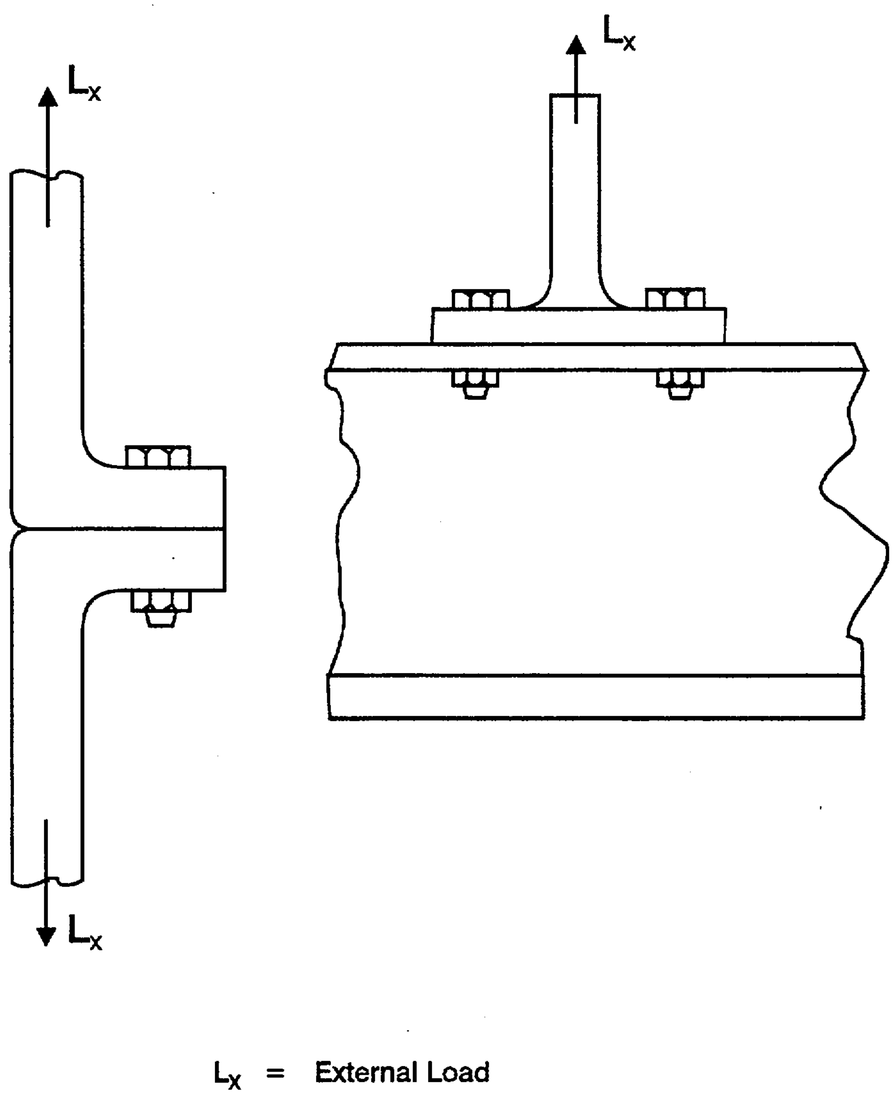
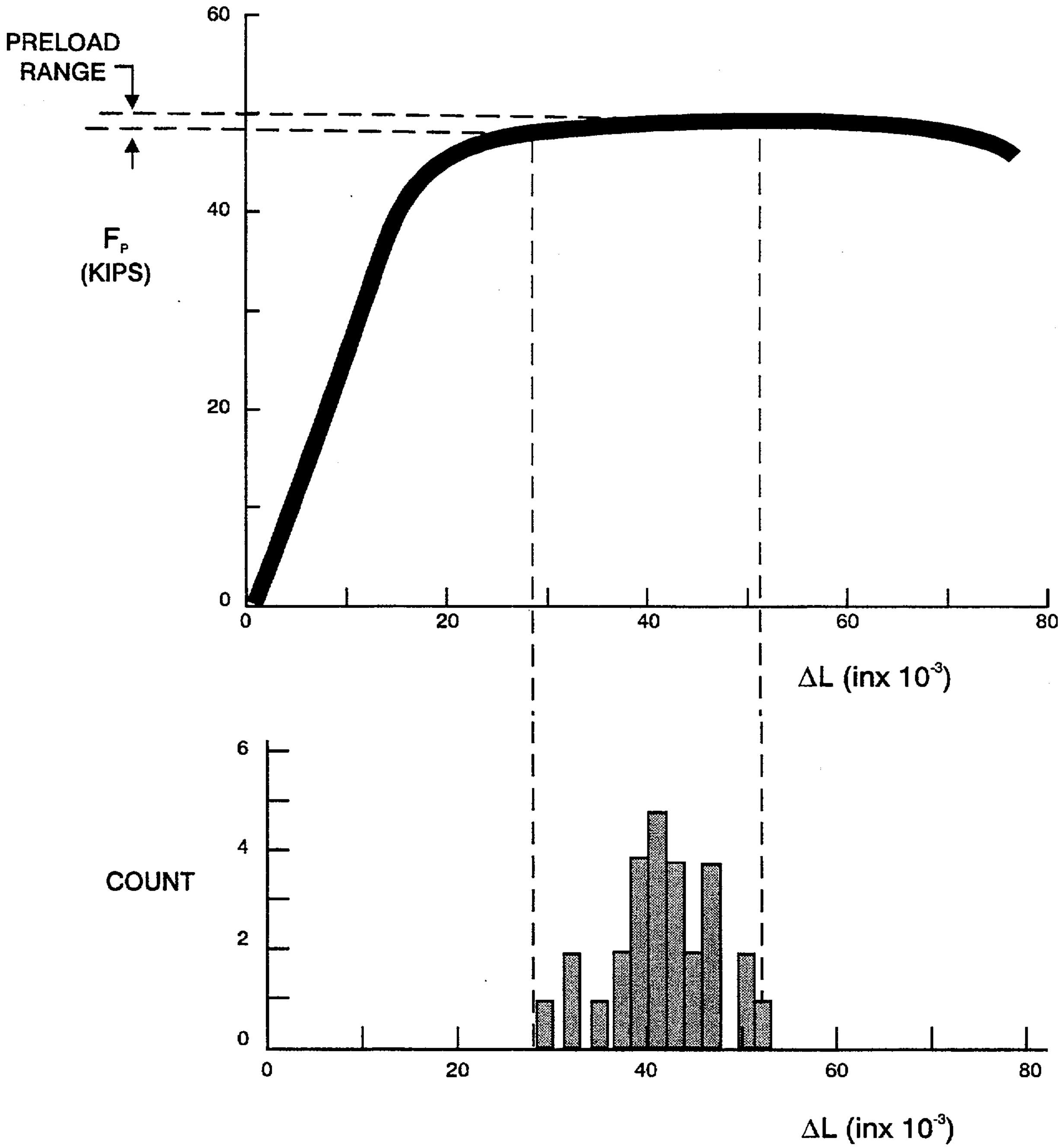


Figure 10-2 — Tension Joints.



F_P = Bolt Preload

Δ_L = Change in Length of a Bolt

Figure 10-3 — Illustration of Bolt Ductility.

10.1.3 Washers

The AISC Specification requires the use of washers under these conditions.

- Beveled washers are required if the flange surface under the bolt head or nut is sloped greater than 1:20.
- Hardened washers are required under the following conditions:
  - Under the turned element if a torque wrench assembly is used
  - Under the bolt lead and the nut when A490 bolts are used with joint material with a yield strength less than 40 ksi
  - With oversized holes or slots
  - When A490 bolts over 1 inch diameter are used with oversized holes, heavy (5/16 inch thick) hardened washers are required on both sides.

10.2 Strength Designs for Bolted Connections

The design philosophy for structural joints is summarized by the following quotation (33):

> *In bolted connections subject to shear type loading, the load is transferred between the connected parts by friction up to a certain level of force. This force is dependent on the total clamping force on the faying surfaces and the coefficient of friction of the faying surfaces. The connectors are not subject to shear nor is the connected material subject to bearing stress. As loading is increased to a level in excess of the frictional resistance between the faying surface, slip occurs. However, failure in the sense of rupture does not occur. As even higher levels of load are applied, the load is resisted by shear on the fastener and bearing on the connected material plus some uncertain amount of friction between the faying surfaces. The final failure will be by shear failure of the connectors, by tear out of the connected material, or by unacceptable ovalization of the holes. The final failure load is independent of the clamping force provided by the bolts.*

The design of AISC high strength bolted connections begins with the determination of the strength required to prevent premature failure by shear of the fasteners or by bearing failure of the connected material. The next important design determination is to check "slip critical" connections for slip load resistance. Slip critical connections include those connections in which slip could theoretically exceed 1/16 inch. A slip of this magnitude could affect the suitability of the structure for service by allowing excessive distortion and reduction in strength or stability. This condition may exist even though the resistance to fracture may be adequate. Other examples of slip critical connections are those cases where slips of any magnitude must be prevented (e.g., joints subject to fatigue or reverse loading).

The AISC specification publishes tables of allowable working stress for fasteners in different types of joint configurations. These tables are based on test results sponsored by the Research Council on Riveted and Bolted Structural Connections. Fisher's "Guide to Design Criteria for Bolted and Riveted Joints" (34) is a comprehensive summary of test data. Fisher's guide also includes an analysis and explanation of the methodology used in developing the Code rules, procedures, and allowable stresses.

An allowable working stress design technique is a simplified method of sizing bolts and determining the number of bolts required in a joint. The designer computes the working load by summing the external load and the tension resulting from prying, if applicable. The working load is assumed to act directly on the bolts. Calculate the working stress by dividing the working load by the nominal area of the fasteners. This calculation approach is conservative. The compressive load at the joint interface would have to approach zero to have the entire external load carried by the fasteners (Chapter 6, Sections 6.2.1 provides a discussion of joint diagrams which illustrates how a preloaded joint carries external load). Despite the conservatism, this approach lends itself to an uniform treatment of each joint type.

The designer calculates a very conservative working stress and assumes that the joint has been properly preloaded. The calculated stress is then compared to the stress published by the AISC specification.

10.2.1 Designs for Friction or Slip Critical Joints

After a slip critical joint is checked for shear, the joint load carrying capacity is assessed. The load carrying capacity of the joint is proportional to:

- Friction of faying surfaces
- Number of bolts
- Bolt preload
- Number of slip surfaces.

Surface treatments are also used to control friction. If the faying surfaces of the joint are coated, the joint slip resistance must be demonstrated by test. Figure 10-4 presents the model for analyzing a slip critical joint.

10.2.2 Allowable Slip Load for Slip Critical Joints

The force on a slip critical joint shall not exceed the allowable resistance (P_S) of the connection as defined by the following equation:

$$P_S = F_S A_B N_B N_S \tag{10-1}$$

where,
P_S = Allowable resistance at the connection (kip)
F_S = Allowable slip load per unit area of bolt from Table 10-1 (ksi)
A_B = Area corresponding to the nominal body area of the bolt (inches^2)
N_B = Number of bolts in the joint
N_S = Number of slip planes

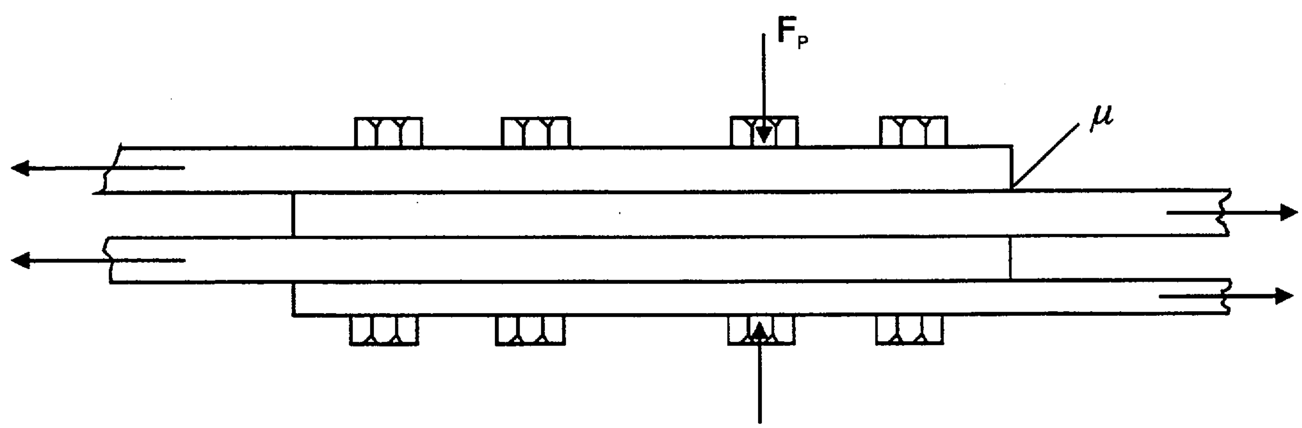


Figure 10-4 — Model for Analyzing a Slip Critical Joint.

Class A, B, or C surface conditions of the bolted parts as defined in Table 10-1 (35) should be used in joints designated as slip critical. Class A and B coatings include those that provide a mean slip coefficient of not less than 0.33 or 0.50, respectively.

Table 10-1

Allowable Load for Slip Critical Connections (Slip Load per Unit of Bolt Area, ksi)

Contact Surface of Bolted Parts	Hole Type and Direction of Load Application							
	Any Direction				Transverse		Parallel	
	Standard		Oversized & Short Slots		Long Slots		Long Slots	
	A325	A490	A325	A490	A325	A490	A325	A490
Class A (Slip Coefficient 0.33): Clean mill scale and blast cleaned surfaces with Class A coatings*	17	21	15	18	12	15	10	13
Class B (Slip Coefficient 0.50): Blast cleaned surfaces and blast cleaned surfaces with Class B coatings*	28	34	24	29	20	24	17	20
Class C (Slip Coefficient 0.40): Hot dip galvanized and roughened surfaces	22	27	19	23	16	19	14	16

** Coatings classified as Class A or Class B include those coatings which provide a mean slip coefficient not less than 0.33 or 0.50, respectively, as determined by the Testing Method to Determined the Slip Coefficient for Coatings Used in Bolted Joints (See ASME Section III, Appendix A.).*

10.2.3 Designs for Tensile Joints

The general model used to analyze tensile joints is shown in Figure 10-5. The figure shows the external load (2T) and the prying loads (Q). Prying loads are the external tension loads on a fastener magnified by a lever action when the line of action of the external load does not lie along the axis of the fastener. Prying almost always exists and is not a problem unless joint members are flexible or the line of action of the load is a considerable distance from the bolt axis (see Figure 10-6). The prying load is related to the stiffness of the structure. In a tension joint, the bolts are sized according to:

$$B_{all} = 0.375 A_B T_{u\ spec} \tag{10-2}$$

where,
- B_{all} = Allowable bolt load on each bolt (kip)
- A_B = Nominal cross sectional area of the bolt (inches^2)
- $T_{u\ spec}$ = Minimum specified tensile strength (ksi)

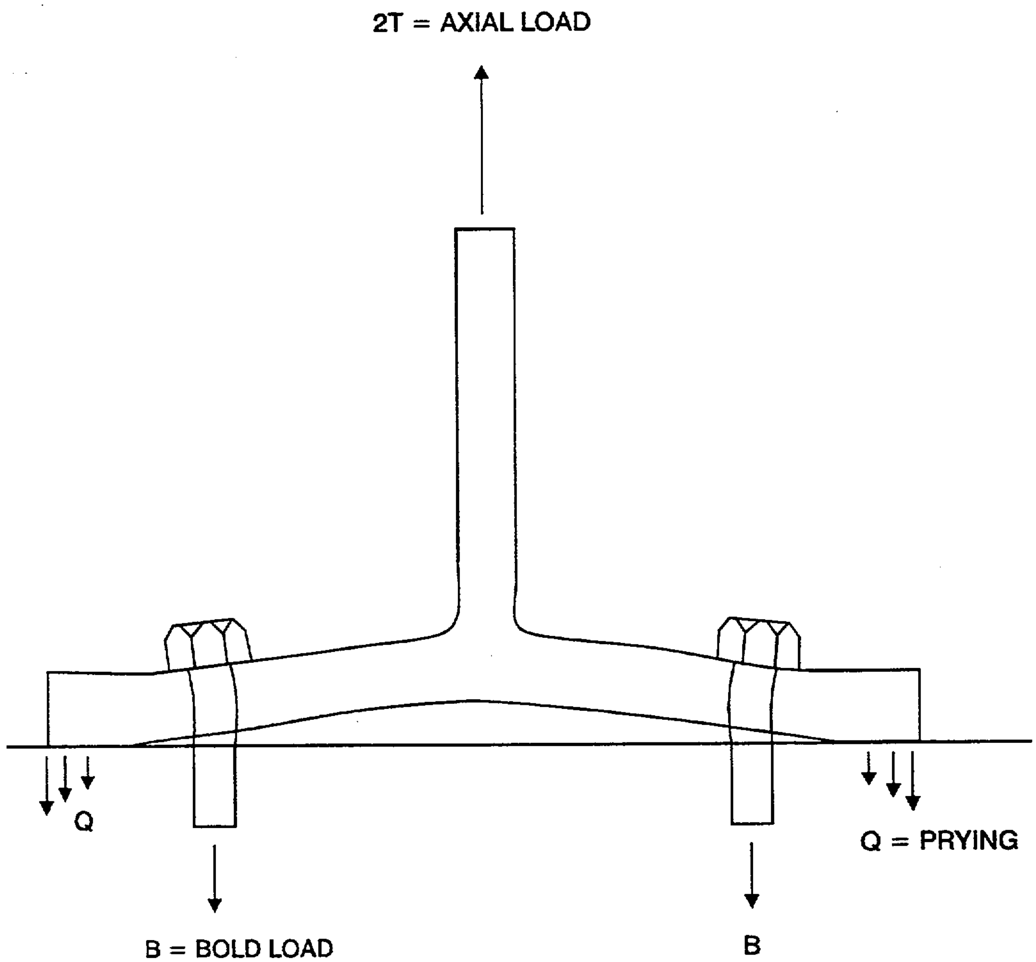


Figure 10-5 — Model for Analyzing a Tensile Joint.

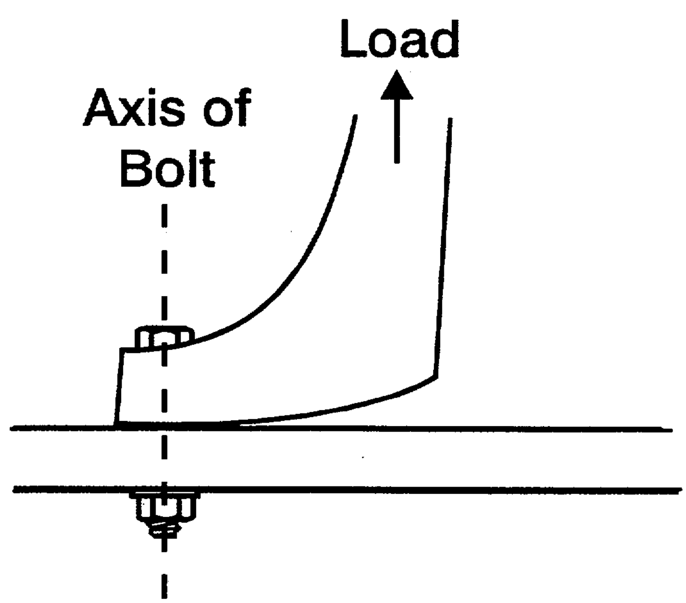


Figure 10-6 — Illustration of Prying Action.

The allowable stress is compared to the external load plus any prying force which is developed. The working load is approximately two thirds of the load induced during tightening.

This design approach subtly employs joint diagram analysis. The joint diagram of a preloaded structure predicts that only a small portion of the external load goes to increase the bolt load. The amount of load increase in the bolt is governed by the relative stiffness of the bolt and the joint. This condition applies until the external load is high enough to separate the joint members. When an external load is applied to a properly tightened joint, the bolt experiences little if any increase in stress.

10.3 ASME Section III, Division I, Subsection NF (6)

American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section III, Division I, Subsection NF-1110 covers component supports intended to conform to the requirements for Classes 1, 2, 3, and MC construction as set forth in Subsections NB, NC, ND, and NE. Component supports are metal elements that transmit loads between nuclear power plant components and the building structure. These supports carry the weight of components or provide structural stability. Component supports may be of the plate shell type such as a vessel skirts or of the linear type such as a clevis and pin arrangement. Hangers are considered structural supports in accordance with Subsection NCA-3240 of ASME Section III.

10.3.1 Subsection NF-1130 Jurisdiction Boundaries

Rules for determining an appropriate interface between NF components and other elements, such as pressure retaining elements, intervening elements, and building structures, are detailed in Subsection NF-1130. Thirteen sketches and explanations cover the possible combinations of elements. Jurisdictional boundaries may be summarized as follows:

- If the bolted joint has at least <u>one</u> NF component, the connection is designed and assembled in accordance with NF.
- If the joint has <u>no</u> NF components (i.e., a joint between an intervening element and a building structure), NF is not applicable.
- For a joint between a NF support and a building structure that contains embedded studs, the following jurisdiction applies:
  - Non-integral support to the building structure: NF-1131-1(f). The connection is governed by NF; therefore, the studs are NF.
  - Integral support to the building structure: NF-1131-1(m). The boundary is at the surface of the building structure (NF-1132.7). The design rules for the studs are <u>undefined</u>, since they cross the jurisdictional boundary.

Items that are not covered by NF include the following:

- Intervening elements in the load path as well as the building structure. The owner is responsible for assuring the adequacy of the building structure and the intervening elements (NF-1111).
- Dynamic loads. The function of a structural element is to carry dynamic loads caused by a postulated loss of pressure retaining integrity (e.g., pipe whip restraints).
- Deterioration that may result from erosion, corrosion, or radiation effects.

10.3.2 Subsection NF-2000 Materials

10.3.2.1 Permitted Material (NF-2121)

Materials must conform to the requirements of the specifications listed in Appendix I. Bolting requirements for all classes of construction are found in Tables I-1.3, I-13.1, I-13.3, I-3.1, and I-3.2.

10.3.2.2 Test Coupon (NF-2224 {b})

This paragraph specifies the location a test coupon is taken from a nut, a stud, or a bolt.

10.3.2.3 Impact Testing (NF-2311)

- Design specification (NCA-3250) states whether impact testing is required for all classes of supports.
- Bolting including studs, nuts, and bolts less than M24 (1.0 inch diameter) are exempted from impact tests for all classes of construction.
- Other exemptions are given based on the class of support, material, thickness, and lowest service temperature.

10.3.2.4 Impact Test Criteria (NF-2333)

For bolting, three specimens must meet the lateral expansion values of Table NF-2333-1 (0.65 mm for bolts over 25 mm diameter).

10.3.2.5 Sampling Frequency (NF-2345)

One sample is required per lot. The lot size is based on the diameter and weight.

10.3.2.6 Required Examinations (NF-2581.1)

Visually examine all Class 1 bolting as required by NF-2582. Diameters greater than M48 (approximately 2.0 inches diameter) required magnetic particle or liquid penetrant inspection per NF-2583. This inspection is performed on the finished bolting material or the material stock of approximately the finished diameter after heading. Linear indications greater than 25 mm long are unacceptable. Sizes greater than M100 (4 inches diameter) also require ultrasonic inspection over the entire volume prior to threading per NF-2584.

10.3.2.7 Visual Examination (NF-2582)

This applies to Class 1, 2, 3, and MC Construction. Visually examine threads, shanks and heads of final machined parts for discontinuities such as laps, seams, or cracks that are detrimental to the intended service.

10.3.2.8 Weld Repair (NF-2586)

Weld repair is not permitted for threaded fasteners.

10.3.2.9 Documentation (NF-2610)

A quality system in accordance with NCA-3800 is required. A Certificate of Compliance is required for small products (less than M48) that do not require impact testing. Furnished Certified Material Test Reports when impact tests are required.

10.3.3 Design Procedure (NF-3000)

The following steps are guidelines for designing structural joints:

1. General design requirements (NF-3100). The joints shall be designed for the following loads:
   - Weight of piping components and the contents under normal operating or test conditions. This includes loads due to static and dynamic head and fluid flow effects
   - Weight of piping or component supports
   - Superimposed loads and reactions induced by supported system components
   - Dynamic loads including earthquake
   - Thermal expansion effects
   - Anchor and support movement effects
   - Environmental loads such as wind and snow.
2. Design loadings (NF-3112). The design loadings, design temperature, and service conditions are established in the Design Specification in accordance with NCA-2142. Component supports are designed using different methods, depending on the type of support and the class of component being supported. The maximum shear stress theory is used in the design by analysis of plate and shell type component supports, such as cylindrical skirts. The maximum stress theory is used in the design of linear

supports. Component standard supports can be designed by load rating (prototype testing) or experimental stress analysis.

3. Design requirements (NF-3324.6). Subsection NF-3324.6 covers the design requirements for all classes of joints subject to tensile loads, shear loads, and a combination of tensile and shear loads. For Class 1 joints, additional consideration must be given to fatigue loading.

4. Tension joints (NF-3324.6{l}). For joints carrying tensile loads, the allowable bolt stress (F_{tb}) is specified as:

$$F_{tb} = S_u \;/\; 2 \text{ for ferritic steel} \tag{10-3}$$

$$F_{tb} = S_u \;/\; 3.33 \text{ for austenitic steel} \tag{10-4}$$

where, S_u = Ultimate strength of the material at design temperature (psi)

The computed tensile stress in the bolt is based on the following:

- The tensile area of the bolt cross section
- The applied load, defined as the sum of the external loads and any tension resulting from prying action.

Preload stress in the bolts is not considered in computing the tensile stress in the bolt. The computed stress must be less than the allowable stress for a satisfactory design.

5. Shear joints (NF-3324.6{2}). The allowable shear stress for bolts in a bearing type shear joint (F_{vb}) is given as:

$$F_{vb} = 0.62 \; S_u \;/ 3 \text{ for ferritic steel} \tag{10-5}$$

$$F_{vb} = 0.62 \; S_u \;/ 5 \text{ for austenitic steel} \tag{10-6}$$

The nominal bolt diameter is used to calculate the bolt shear area. When the fastener threads are in the shear place, the root diameter is used to calculate the shear area. In shear joints, the bolts are assumed to bear uniformly across the thickness of the plates, with the bolts carrying the load in shear.

6. Bearing joints (NF-3324.6(a)). The allowable bearing stress on the projected area of the bolts in a bearing type connection (F_p) given in NF-3324.6 is:

$$F_p = (LS_u \;/\; 2d) - 1.5S_u \tag{10-7}$$

where, L = Distance from center of the bolt hole to the edge of the plate
d = Bolt diameter (inches)

The bearing area is calculated as the projected bolt diameter times the length of the bolt in bearing.

Other rules of NF 3324.6 include:

- Minimum and maximum distances from the center of the bolt hole to the nearest edge are specified.
- Minimum hole spacing is specified.
- For SA-307 bolts with grips greater than five times the diameter, the number of bolts must be increased by 1% for each additional 1.6 mm in the grip.
- Anchor bolts shall be designed to provide resistance to all tension and shear at the bases of any columns, including the net tensile components of any bending moments resulting from fixation or partial fixation of columns.

7. Combined tensile and shear joint (NF-3324.5{3}). When a joint is subjected to both tensile and shear loads, the following equation must be satisfied:

$$\{(f_t)^2 \; / \; F_{tb}\} + \{(f_v)^2 \; / \; F_{vb}\} \leq 1.0 \qquad (10\text{-}8)$$

where, f_t = Computed tensile stress (psi)
F_{tb} = Allowable tensile stress at design temperature (psi)
f_v = Computed shear stress (psi)
F_{vb} = Allowable shear stress at design temperature (psi)

8. Friction joints (NF-3324.6{3}{b}). Friction joints rely on a high clamping bolt force to develop frictional resistance at the joint mating surfaces. These frictional forces transmit the external loads across the joint. The maximum slip resistance of the joint (P_s) is calculated in the equation 10-9.

$$P_s = mnT_iK_s \qquad (10\text{-}9)$$

where, m = Number of shear planes per bolt
n = Number of bolts in the joint
T_i = Initial clamping force per bolt
K_s = Slip coefficient for a particular surface condition. See Table NF-3324.6{a}{4}-1 of ASME Section III, Subsection NF

If the joint clamping force will be reduced by a direct tension load on the joint, T_i should be reduced by an equivalent amount. SA-307 or austenitic steel bolting shall not be used for friction type joints.

Subsection NF-3225.2 lists stress limit factors (K_{bo}) ranging from 1.0 for service level A to 1.25 for test loading. The stress in a component must not exceed the allowable of NF-3324.6 times the stress limit factor for the appropriate loading condition. The product must not exceed the yield strength at temperature. These multiplying factors are not applicable for friction type joints.

9. Design by load rating (NF-3281). When bolted assemblies are tested to establish a load rating, the bolted joints for the test sample should be made using the lowest strength bolt material and minimum edge distance allowed by specification. The NF joint design rules of ASME Section III are similar to the AISC rules. The NF rules are specific for tensile and shear joints where:

   - All joint members are metal
   - One or more of the members are NF.

   The joint design for an integral NF support attached to a building structure with embedded studs is not adequately covered by NF. The jurisdictional boundary for this joint is at the surface of the building structure (NF-1132.7). This boundary sets the studs in the interface. It is not clear whether the studs are structure or NF. It is recommended that the studs be designed using the more stringent requirements (i.e., NF). However, NF does not contain rules for embedment design.

10. Fatigue analysis. High cycle fatigue analysis is required per NF-3143 for linear supports for Class 1 construction. NF-3330 prescribes the rules for high cycle fatigue design:

   - If the members are subjected to a repeated variation of live load, the range of stress and the number of stress cycles must be considered in the design analysis (NF-3332.1).
   - Properly tightened ASTM A325 or ASTM A490 bolts do not require a fatigue analysis (NF-3332.5).
   - The maximum stress, including the effects of prying action, should not exceed the allowable of NF-3332.4 subject to the following conditions:
     - Joints subject to 20,000 to 500,000 cycles of direct tension may be designed for the stress by summing the applied axial load and the prying load, provided the prying load does not exceed 10% of the external load.
     - If the prying load exceeds 10% of the external load, then the allowable stress of NF-3324.6 is reduced to 40% of that applicable to the external load alone.
     - Joints subject to greater than 500,000 cycles are designed as above with the prying load limited to 5% of the external load. If the prying load exceeds this 5% value, then the allowable is reduced by 50%.
     - The use of threaded fasteners in tensile fatigue loadings is not recommended.
     - Design shear joints subject to fatigue loading as bearing type connections incorporating the fatigue strength of the fastener.

11.0 Assembly

11.1 Introduction

The American Institute of Steel Construction (AISC) maintains that high bolt preloads are desirable for three reasons:

1. High bolt preloads increase joint rigidity.
2. High bolt preloads result in better stress patterns in the connected plies.
3. High bolt preloads provide security against loosening.

The AISC specification for structural joints using A325 or A490 bolts mandates minimum preload values of 70% of the ultimate strength of the bolt. No maximum preload is mandated. If the bolt does not fail on installation, it will not fail in service. On the other hand, if a bolt breaks during tightening, it is simply replaced.

The same minimum bolt preload is specified for slip critical and tensile joints. Bearing shear joints do not require bolt preload.

11.2 AISC Joint Assembly and Preload Control

The AISC prescribes four methods for preloading bolts in a joint assembly:

- Turn of nut
- Calibrated torque wrench
- Load indicating devices
- Alternate design bolts.

11.2.1 Turn of Nut

The turn of nut method was developed by the American Association of Railroads for use in remote areas where power tools were unavailable. Bethlehem Steel modified the method as it exists today.

The procedure calls for turning the nut down the bolt to a snug position. The nut is then turned an additional one half or three fourths turn, depending on the length of the bolt. The snug torque is designed to achieve approximately 10% of the desired load. The additional turn achieves a load beyond the specified minimum bolt load, usually beyond the bolt yield. Figure 11-1 illustrates the turn of nut process.

When the turn of nut method is properly executed, the bolt preloads are commonly above the minimum specified values. The most significant problem with the procedure is controlling the snugging value. The whole joint must be snug before the final turn

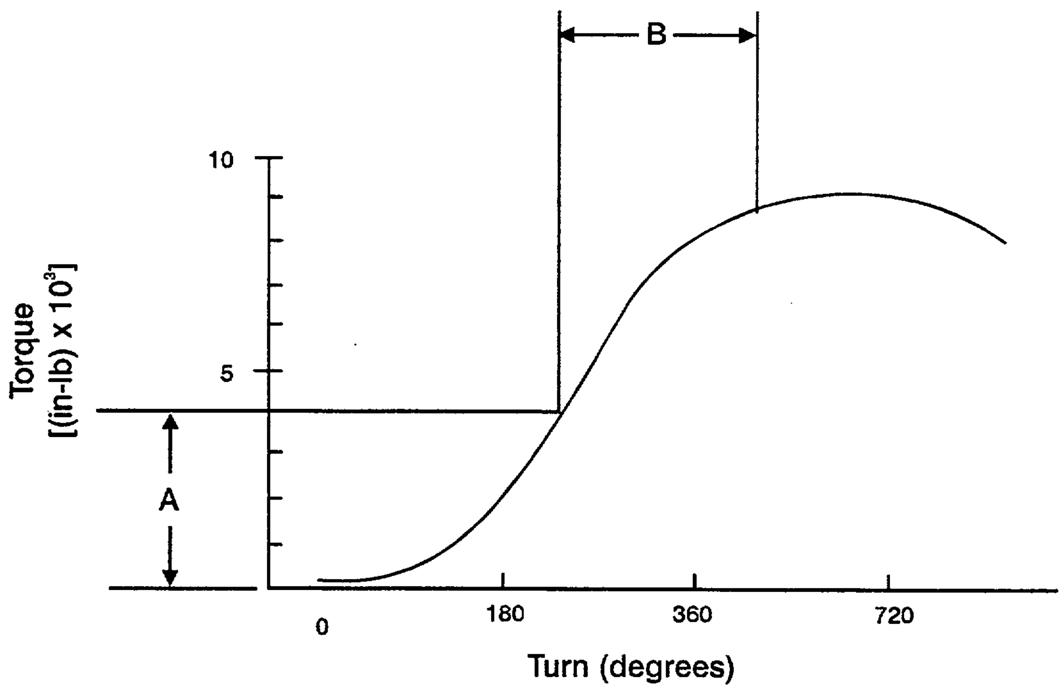


Figure 11-1 — In Turn of Nut Techniques, the Nut is First Tightened with an Approximate Torque (A) and Then Further Tightened with a Measured Turn (B).

increment is applied. If the joint members are not snug, the effect of the final turn is not achieved. Instead, the final turn pulls the plies together rather than stretching and preloading the bolt.

11.2.1.1 Snug Tight Considerations

The AISC specification provides guidelines for achieving a snug joint before the incremental turn is applied. Snug tight is defined as a few impacts of an impact wrench or the full effort with a spud wrench. The specification suggests that a sufficient number of bolts in the joint be snug to pull the joint plies together. Another recommendation is to work from the most rigid part of the joint to the free edges. This ensures that the incremental turn of the nut is used to pull the joint together rather than to preload the bolt.

11.2.1.2 Assembly Verification

The AISC specification requires validation of the turn of nut procedure prior to the start of work. Validation is done with a tension measuring device (e.g., a Skidmore Wilhelm tester), refer to Figure 11-2. The turn of nut procedure is tested on three bolts of each size to be installed. The tension developed by the method must be 5% greater than the prescribed minimum tension for each bolt size.

11.2.2 Calibrated Torque Wrench

The AISC has changed philosophy on the use of the torque wrench method. Torque wrench assembly was allowed prior to 1980. In 1980, the method was withdrawn but was reinstated in 1985. AISC defined a number of conditions for the use the torque wrench method in the 1985 reinstatement.

Two prevalent perspectives on torque wrench assembly exist. Field personnel use torque wrenches because of quick assembly. From a technical and theoretical perspective, the method is less favored. The Research Council on Structural Connections expressed concerns in 1980 about the torque method. The following is from the 1980 edition of the AISC specification on torque withdrawal (6):

> *Early editions of this specification listed torque values described as approximately equivalent to the minimum bolt tension specified for various size bolts. It was explained that these values are no more than observed experimental averages. The value to be used, both in installing bolts and in inspection procedures, should be determined by the actual condition of the application. With this edition of the specification, recognition of torque control methods is withdrawn because of the large variability of torque to tension relationships for seemingly similar bolts and conditions.*

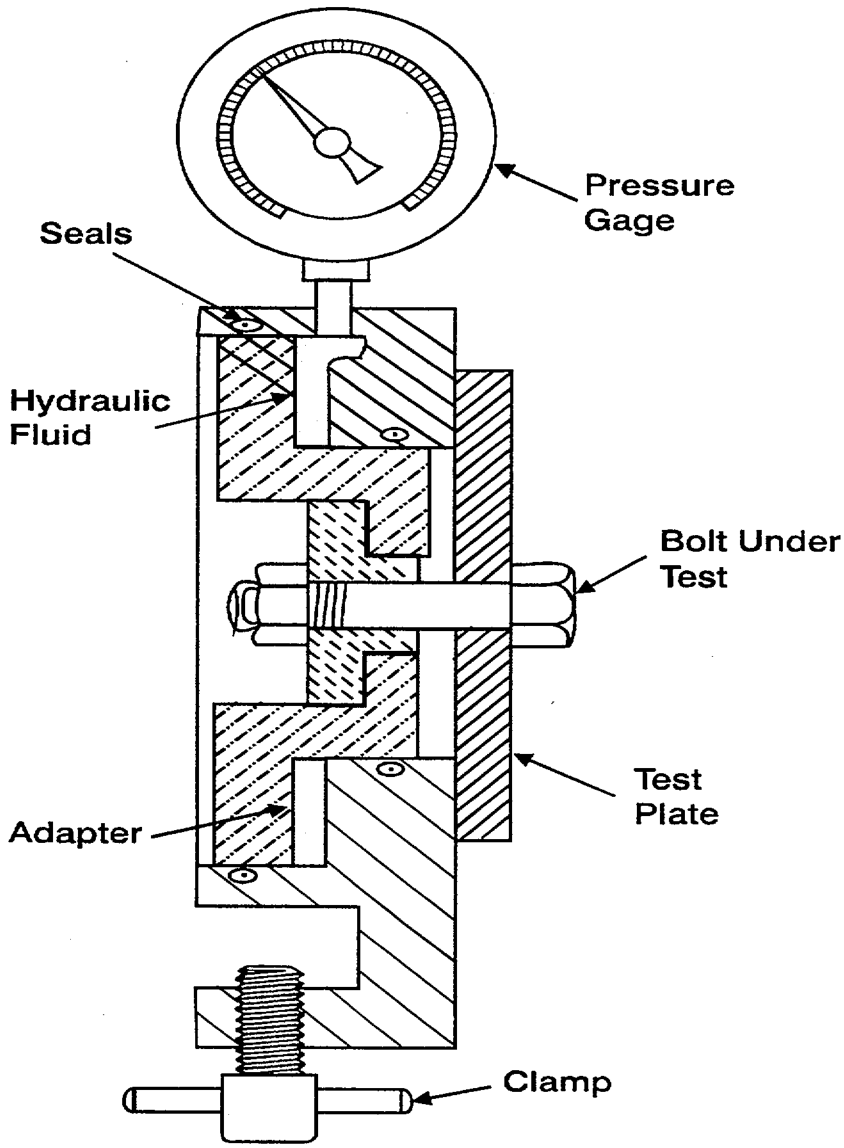


Figure 11-2 — Skidmore Wilhelm Tensile Tester.

One key phrase in the excerpt, that the installation torque values should be "determined by the actual condition of the application", is an excellent guideline. However, this guideline is virtually impossible to implement in a field situation. To simulate field conditions, tests are required on the actual field structure. The problem is to measure the fastener load resulting from applied torque. This is a complex testing condition and several options are available:

- Inserting a load indicating cell under the bolt head
- Using strain gaged bolts
- Using an extensometer device to measure load related extension
- Simulating the joint in a Skidmore Wilhelm tension indicating device. Figure 11-2 illustrates this device.

The 1985 edition of the AISC Code allows torque wrench assembly, provide these requirements are met:

- Install a hardened washer under the turned element.
- Calibrate the installation procedure daily. Demonstrate the torque setting produces the minimum specified preload.
- Recalibrated if a difference in the surface condition of bolt threads, washers, or nuts is observed.
- Verify the amount of turn from snug during tightening so that the prescribed turn is not exceeded. Even though torque is being used, there is a check of turn.

11.2.3 Load Indicating Devices

The most commonly used tension indicators in structural steel assembly are called Direct Tension Indicators (DTI). A typical DTI washer arrangement is shown in Figure 11-3. At assembly, the bolts are loaded until a specified clearance exists between the washers. This clearance is checked with feeler gages. The washers cannot indicate and respond to load relaxation, since there is deformation. Hardened washers are used in conjunction with indication washers to prevent the indication washer from deforming in the joint member hole. The AISC specification requires the crush of the washer be measured and be related to the minimum specified preload.

11.2.4 Alternate Design Bolts

Alternate design bolts are allowed by the AISC specification. Two types of alternate design bolts are illustrated in Figure 11-4. To use these designs, the following conditions must be met:

- The fasteners meet the chemistry manufacturing and strength requirements of ASTM A325 or A490 specifications.

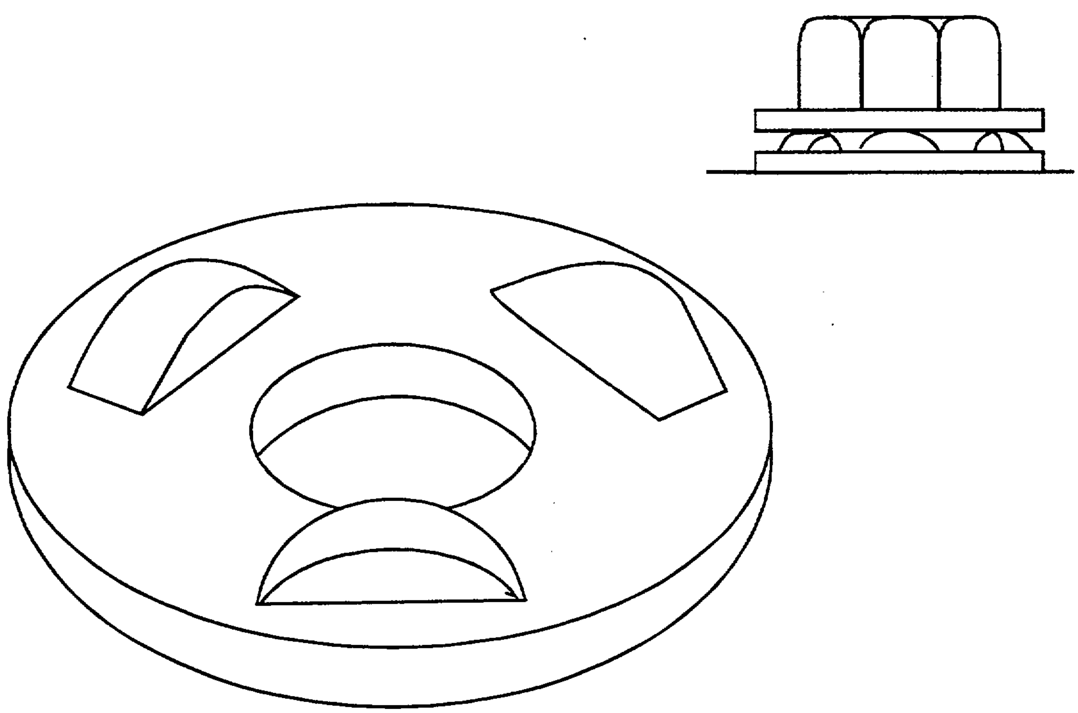
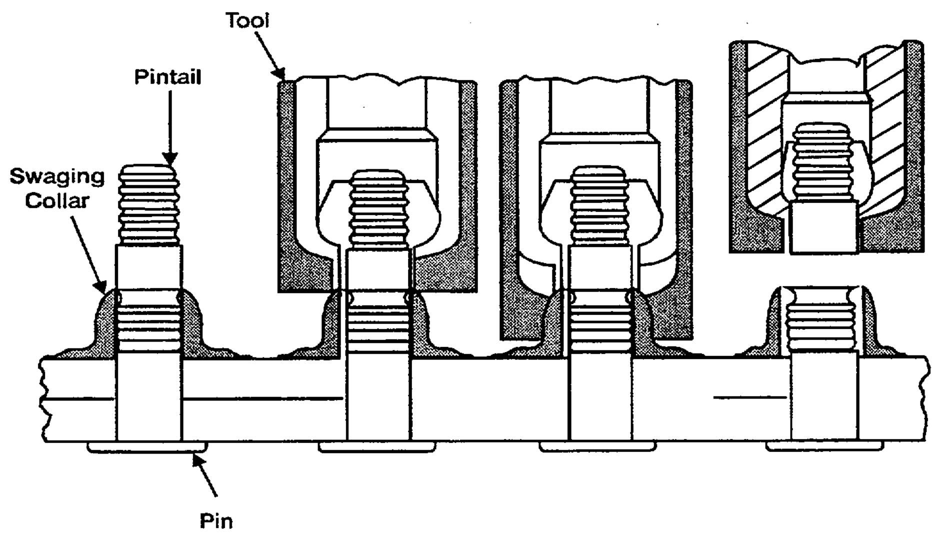
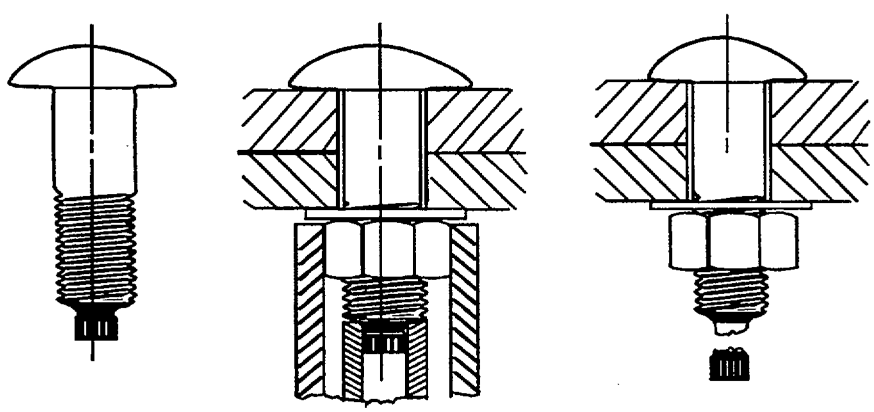


Figure 11-3 — Crush Washer Used to Measure Preload.



a) Lock Bolt (Huck Mfg. Co.)



b) "Twist Off" on Tension Bolts

Figure 11-4 — Alternate Design Bolts.

- The responsible engineer approves and documents the installation and inspection methods.

- The preload developed by the assembly system is demonstrated using a tension measuring device at the initiation of the job.

11.3 Structural Joint Assembly Procedures

The following is an assembly procedure for structural joints:

1. Line up joint members using drift pins in a few holes. Install bolts in the remaining holes hand tight. Snug all the bolts, including the first bolts installed, to 10% to 30% of the final torque. Snugging torque is applied first to the bolts in the most rigid part of the pattern, then to those bolts farther out, and finally to the bolts on the free edges of the joint. Ensure that joint members are pulled together fully. Apply additional snugging torque, if necessary.

2. Remove the drift pins, if any, while the joint is not under load. Install and apply snugging torque to the remaining bolts.

3. Apply the final torque to each bolt. Begin with the most rigid part of the joint and work toward the free edges.

4. Remove the remaining drift pins after all bolts have been fully tightened. Install and fully tighten the remaining bolts.

5. For critical joints, reapply the final torque to all of the bolts in the joint, following a reverse sequence. Other preload measuring devices such as stretch of fasteners can be used.

11.4 Turn of Nut Assembly Procedures

The following is an assembly procedure using the turn of nut method:

1. If bolts are to be tightened past the yield strength, ensure that the bolt material is ductile. Examples of acceptable materials include ASTM A325 and A490.

2. Ensure that the joint components are correctly assembled. Consider:
   — Bolts are the proper type and length.
   — The correct washer is installed, where required. This may include standard, beveled, or heavy washers.
   — Joint alignment is ensured using drift pins. Mark the socket or turned element of the fastener (nut or bolt head) so that the turn may be measured.

3. For best results, apply a high snugging torque. Measure the turn. This torque should produce 10% to 30% of the minimum specified preload.

When the turn of nut method is used, enough bolts must be snug tight to ensure that all parts of the joint are in contact with each other. Snug tight is defined by AISC as the tightness attained by a few impacts of an input wrench or the full effort of an ordinary spud wrench. If bolts are larger than 7/8 inch in diameter, more force will be required to achieve a snug tight fit.

After the initial snugging, place the remaining bolts in connection holes and snug the bolts. Additional tightening is required of all joint bolts. Use nut rotation values specified in Table 11-1. Work from the most rigid part of the joint to the free edges. During this operation, only the wrench is allowed to rotate. SA 307, Grade A bolts should be tightened snug tight without additional nut rotation.

Table 11-1

Effective Bolt Length (Distance from Inside Face of Bolt Head to Outside Face of Nut Plus One Tread)	Disposition of Outer Faces of Bolted Parts: Both Faces Normal to Bolt Axis	One Face Normal to Bolt Axis & Other Face Sloped Not More Than 1:20 (Bevel Washers Not Used)	Both Faces Sloped Not More Than 1:20 From Normal to Bolt Axis (Bevel Washers Not Used)
Up to and including 4 diameters	1/3 turn	1/2 turn	2/3 turn
Over 4 diameters but not exceeding 8 diameters	1/2 turn	2/3 turn	5/6 turn
Over 8 diameters but not exceeding 12 diameters	2/3 turn	5/6 turn	1 turn
Note: For bolting ≥ 1/2 in. A325 or A490			

Table 11-1 applies to ASTM A325 and A490 bolt materials. These bolts are designed to be installed with a minimum preload of 70% of the tensile strength. The bolts will likely be loaded above the proof strength. As a result, some bolt breakage may occur at assembly. This breakage is acceptable. Personnel using the turn of nut method must be trained on how to perform the tightening.

Acceptable locking devices for threaded fasteners include elastic stop nuts (when compatible with the service temperature), lock nuts, jam nuts, drilled nuts, and wired nuts. Upset threads developed by peening or by other approved methods may also serve as locking devices.

Nut rotation is relative to the bolt, regardless of whether a nut or a bolt is being turned. For bolts installed by one half a turn and less, the tolerance should be ±30°. For bolts installed by two thirds turn or more, the tolerance should be ±45°.

To establish a turn of nut method for bolt lengths greater than 12 diameters and/or for bolt diameters greater than 1 1/2 inches, the required rotation from snug is determined by a tension measuring device, simulating the actual joint conditions used. This also applies when bolt materials other than ASTM A325 or A490 are used.

11.5 ASME Section III, Subsection NF Assembly Procedures

Bolt preload is considered in NF support (NF-3000) design only in friction joints (NF-3324.6{3}{b}). To maintain the contact forces between the plies of joined material, preloads are essential.

The phrase "properly tightened ASTM A325 or A490 bolts" is used in NF-3332.5. Bolts tightened to these specifications are tightened with a minimum preload of 70% of the tensile strength. This is the only reference to preload in NF-3000, and it pertains to fatigue considerations for Class 1 joints.

Subsection NF-4725 states that preload is an acceptable locking mechanism. The preload must be 20% above the maximum load on the fastener for the specified loading condition. An upper limit of 70% of the tensile strength of the fastener contradicts the 70% lower limit for ASTM A325 and A490 materials.

11.5.1 Joint Assembly (NF-4700)

Subsection NF-4700 of ASME Section III defines the requirements for bolted construction. Many details of NF-4700 are similar to those addressed by the AISC specification. The following subjects are addressed:

- NF-4711: Threads engagement over the full length of the load carrying nut
- NF-4712: Suitable thread lubricant for the service conditions
- NF-4721: Guidance for the use of oversized and slotted holes and hardened washers over these holes
- NF-4722: The use of beveled washers for slopes of more than 1:20
- NF-4723: The condition of the contact surfaces of the joint materials, including chips, scale, and deleterious material
- NF-4724: The use of a hardened washer under the turned element when a calibrated torque wrench is used
- NF-4724: Achieving the bolt tension by tightening to a torque not less than that given in the design specification

- NF-4724: The use of the turn of nut method
- NF-4724: Control of bolt tension by load indicating washers or by direct extension indicators
- NF-4724: The use of hardened washers when using indicating washers, or when the joint material has less than 275 MPa yield strength
- NF-4725: The use of locking devices to prevent loosening
- NF-4725: The use of sufficient fastener preload as a locking device
- NF-4725: Testing of threaded assemblies for dynamic loading conditions specified in the design specification
- NF-4725: Verification of the established preload in the assembly by properly calibrated torque wrenches, hydraulic tensioners, direct extension indicators, turn of nut method, or ultrasonic method. The results of the test, the required preload, and the specified thread lubricant are provided in the design report.

11.6 Tensioning Assembly Procedures

It is best to follow the instructions outlined by the manufacturer of the tensioner during assembly. General recommendations are:

- Use multiple tensioners together to reduce the elastic interactions, if possible.
- Use a uniform run down torque. Ensure that the nut turns.
- Verify that the specified hydraulic pressure is applied to the tensioner.
- Check all hydraulic connections.
- Check the pressure before running the nut down on each stud. When using multiple tensioners ganged to one pump, running one nut down may reduce the system pressure.
- The nominal tensioner load should be 20% to 30% higher than the desired stud preload to compensate for relaxation effects.
- Perform a final check after the specified tightening passes by applying the final tensioner pressure to each stud and attempting to run the nut down. If the nut moves, the residual preload is low, and additional tightening passes are required.

11.7 Torquing Assembly Procedures

The following is an assembly procedure for torquing:

1. Ensure that the load bearing surfaces are in good condition. Check the thread flanks, the bearing surface of the nut or bolt head, the washers, and the flange surface.

2. Clean and lubricate the threads and the nut and/or bolt head bearing surface. Use the specified lubricant, and apply it uniformly as directed.

3. Use hardened washers to improve the torque/preload relationship. This practice is required by some Codes but is recommended in all situations.

4. Run the nuts or bolts down by hand. Defects in the thread may prevent the nut or bolt from moving freely.

5. Use a multiple pass, cross bolting procedure.

6. Torque as many studs as possible simultaneously. This will help pull the joint down evenly and reduce the loss of bolt load.

7. Use calibrated torque wrenches with adequate capacity.

8. Apply torque at a uniform rate. The final torque should be reached "in motion". If the final torque is exceeded, loosen the fastener and try again to reach the final point in motion.

9. Hold torque wrenches perpendicular to the axis of the bolt while torque is being applied.

10. Ensure that adequate action points are provided if hydraulically powered torque wrenches are used.

11. Torque the fasteners in the reverse sequence in the final pass. This step often improves preload uniformity. This is especially useful for joints with problem histories.

12. In critical situations, verify the preload by making stretch measurements of the fastener.

12.0 Inspection

12.1 Quality Control Methods

The American Institute of Steel Construction (AISC) devotes only one page to inspection activities. The specification states that in process inspection is required. The inspector monitors installation to ensure that approved procedures are being followed.

The concept that monitoring and controlling the assembly procedure is the best quality approach is echoed by Fisher (36). He says:

> *Inspection is usually unnecessary if the installation method has been carefully followed.*

An underlying reason is that the techniques for measuring bolt tension in an assembled joint are both difficult and unreliable. The most common method for determining preload in an assembled joint is to measure the restart torque with a calibrated torque wrench. This is the arbitration procedure defined in the AISC specification.

1. Set up three bolts in a calibrated load measuring device, such as a Skidmore Wilhelm tensioner tester.
2. Perform the prescribed turn of nut tightening including snugging to 15% of the load. Continue to tighten to the specified minimum load. The incremental turn required to achieve the specified load must be as specified by the Code.
3. Measure the torque required to restart the nut in the tightening direction and turn it 5° from the position of step 2. The average torque of the three bolts is the value used in the inspection, called the "job inspection torque".

Torque tension results obtained on a calibrated tension device are not the same as the torque tension relationship in an actual joint. AISC, Fisher (36), and Notch (32) acknowledge this finding.

Notch states that the job inspection torque value is non-conservative. This means that for a given fastener tension, more torque is required to restart the installed bolts than for the bolt setup in a Skidmore tension machine. Using the job inspection torque value as an estimate of installed bolt tension leads to an overestimate of the existing bolt preload.

12.2 Inservice Inspection

The ASME Boiler and Pressure Vessel Code Section XI provides requirements for those applications covered by ASME Section III of the Code, including Subsection NF connections. All structural connections in the plant merit a varying degree of attention to assure integrity.

Degradation during service that is of potential concern, is that - physical, mechanical or corrosion related damage that could affect appearance, functionality, strength, material dimensions, or tightness. Degradation may be caused by environmental damage, structural vibration, or loss of support by load shedding from adjacent elements or fasteners.

Experience has shown that most degradation of structural connections results from galvanic or anodic corrosion. Coated structural connections typically begin to degrade at local areas that are anodic to the rest of the joint, e.g., lesion or holiday in the coating, different material, etc. This phenomena is seen in automobiles, boat trailers, bridges, embedments, etc.

Joints that may be particularly vulnerable, include exposed joints in or near ocean environments or otherwise exposed to the elements. Structural joints inside containment should not be ignored due to moisture and for pressurized water reactors, the presence of boric acid.

Indications of potential problems include loss of coating integrity and obvious signs of corrosion, rust, etc. In addition, Gulf coast refinery experience documented by the American Petroleum Institute highlights degradation of embedded fasteners, noting that under certain severe environmental conditions embedded bolts may degrade - below surface of the embedment with little or no evidence of degradation on the heads of the fasteners.

Prioritization of inspection and repair should focus on critical applications. A structural connection may be considered suspect of degradation when the coating is degraded (e.g. evidence of flaking, peeling, rust, etc.) or the area of contact between bolts and concrete or steel show evidence of degradation.

The worst condition will most likely be found on the fastener that visually exhibits the worst appearance. Sampling inspection of that fastener can provide a biased inspection for determining condition of adjacent fasteners. This rationale will not be applicable to high strength fasteners or those subject to other modes of degradation such as fatigue or stress corrosion cracking.

If connections are found corroded during the visual inspection, a closer inspection to assess extent of corrosion may include, evaluation of:

- Nuts, are any of the flats unworkable, any evidence of cracking?
- Threads, are any wasted to the root diameter, any evidence of cracking?
- Any other signs of physical or mechanical distress?

13.0 Troubleshooting

13.1 Field Conditions

The design of structural joints does not consider corrosion, see ASME III Subsection NF and AISC. The assumption is that the fasteners will be protected and will not corrode. A comprehensive plant coatings program that includes attention to structural connections, is essential therefore to meeting code requirements.

Craft training and plant coating programs should emphasize skill of experienced inspection and maintenance personnel to identify conditions that are unsafe and in need of attention. Walkdowns and visual inspections should include examination of joints and the need to identify connections that are of a potential concern.

Degradation can be detected and measured either by removing the fastener, proof test by tension or torquing, by in situ ultrasonic tests, or hammer test.

Joint redundancy is a key to quantification of the effect of corrosion on the margin of safety in a structural connection.

13.2 Guidelines for Troubleshooting

Table 13-1 is used for structural joint troubleshooting.

Table 13-1

Structural Joint Troubleshooting

Problem	Cause	Resolution
Loose bolts	Joint interaction	More bolt passes required; start from center
Loose bolts	Turn of nut, joint soft, joint not properly snugged	Use hardened washers or snugging passes
Thread stripping	Overload, insufficient thread engagement, under design	Check nut and bolt threads, check nut engagement
Corrosion	Unpainted surfaces, wrong material choice, harsh environment	Paint, galvanize, change material

14.0 Fastener Selection

14.1 Introduction

The effective use of threaded fasteners is a four step process. The fastener must be selected, specified, inspected, and installed. The variety of fastener applications and the number of available fastener specifications complicate this process.

Appendix A contains a matrix of bolting materials commonly used in nuclear and similar applications. The matrix presented as a wall chart for quick reference is organized by application and is intended to facilitate the comparison of possible substitute materials. Information concerning material properties and suitable alternate bolts and nuts is presented.

An important relationship exists between the performance of a bolted joint, the fastener quality, and the fastener specification.

Carl Osgood (37), a noted author on mechanical design, described the relationship as follows:

> *Most machines and structures are so constructed and loaded that the bolts are in series with the load. Thus, there is a great advantage in that design approach which considers the assembly as a collection of joints connected by load carrying members. And from this situation arises the corollary requirement of high quality in the fasteners, or at least that the quality level be known. To these ends, many procedures and specifications have been devised.*

In a nuclear power plant, wide ranges of operation and design requirements for bolted joints exist. This leads to use of fasteners with varying degrees of specification and quality. The quality level of the fastener must be high enough to ensure the performance of the joint, but should not impose unnecessary cost. The wide spectrum of operational requirements and fastener specifications is illustrated by comparing a Class 1 pressure boundary joint to a nonsafety related structural joint.

Critical joints are those classified by American Society of Mechanical Engineers (ASME), Section III of the Boiler and Pressure Vessel Code as Class 1. These joints are critical to maintaining pressure containment integrity and usually have a demanding service loading. A failure of these joints would result in severe consequences. These joints are designed and fabricated to high quality standards and clearly require the highest quality fasteners.

Nonsafety related structural joints are on the opposite end of the design and quality spectrum from Class 1 joints. Structural joints are generally constructed of low strength materials. The sizing of the components provides wide margins of safety and redundancy. The quality level and fastener performance for structural joints are very different from Class 1 joints. Structural fasteners are generally manufactured to lower quality standards than Class 1 fasteners. The design process for structural joints allows for acceptable lower quality levels.

14.2 Performance and Fastener Specification

A fastener material specification addresses both the quality and performance criteria of the fastener. The performance criteria includes the strength, ductility, fatigue, and corrosion resistance. An example of performance elements of specifications may be aircraft fastener specifications that provide for high strength and fatigue resistance. In contrast, structural fastener specifications require lower strength, high ductility, and no criteria for fatigue performance.

Various fastener material specifications have been developed which provide specific performance in bolted joints. Design codes commonly list acceptable fastener material specifications and often specify supplementary quality requirements. In order to choose a fastener material, it is necessary to be familiar with the design code as well as fastener material specifications.

Numerous factors influence the performance of a fastener in service, including joint design, materials, quality control, installation, lubrication, and environment. The use of codes and specifications does not always address all of these factors. Other considerations must be applied to ensure a successful application.

Furthermore, neither a design code nor a fastener specification is a static document. Both are consensus documents that represent the best current knowledge. As experience increases, the codes and specifications are modified to reflect current experiences and knowledge.

14.3 Standardization

Many fastener types exist and are covered by an equal number of specifications. The Bolting Matrix contained in Appendix A illustrates that often only a slight difference in performance characteristics exists between similar specifications. This difference is usually associated with the quality level and the documentation required. With proper certification and testing, the fasteners in Table 14-1 can meet most bolting requirements. This illustrates that most applications can be satisfied by a limited selection of fastener grades.

Table 14-1

Comparison of Grades

Bolt or Stud	Washer	Nut
General Structural and Pressure Boundary		
A193 B7	F436 Type 1	A194 Gr. 7 or 2H
A320 L7	F436 Type 1	A194 Gr. 7
A540 B23 Class 5	F436 Type 1	A194 Gr. 7
A320 L43	F436 Type 1	A194 Gr. 7

Table 14-1		
Comparison of Grades		
Bolt or Stud	Washer	Nut
Corrosive Applications		
A193 B8, B8M	18/8 Stainless Steel	A194 Gr. 8, Gr. 8M
A564 630	18/8 Stainless Steel	F594 630

A194 Gr 2H nuts are not suitable if charpy impact test is required.

The replacement bolting must meet the requirements of the code of record of the original construction. Replacements may meet a portion or all of the revised code editions, if the changes are reconciled with the Owner's Design Specification.

14.4 Selection

The selection of material specifications for fastener components is based on the design requirements of the assembly. The desired combination of fastener properties is dictated by the service, the design, and the governing code. Some of the more important requirements are:

- Strength. The fastener components must be strong enough to provide a clamping force to the joint members throughout the service life. In some cases, the maximum fastener strength is limited to protect relatively weak joint members from assembly damage.
- Chemical and physical properties. Fasteners must have suitable corrosion resistance, strength at elevated temperature, thermal expansion coefficient, fracture toughness, and ductility.

W. Stewart (38) notes that

> *...In ordinary usage of steel bolts, the chemical composition of the material and the manufacturing process are of little interest to the user, provided that the service requirements are met.*

Fastener material specification systems have been developed according to this reasoning. Most specifications are basically performance specifications. They emphasize performance criteria such as strength, hardness, ductility, and impact resistance. The material composition is usually flexible. This allows the manufacturer to choose the best material for the fastener.

15.0 Specification and Procurement

15.1 Introduction

The material requirements for a fastener in nuclear service are a combination of requirements from several sources. The fastener is usually ordered to the specifications given by the American Society for Testing and Materials (ASTM) or the American Society of Mechanical Engineers (ASME). Supplementary tests and certification requirements are specified by the section of the Code governing the assembly. The Code requirements involve impact and hardness tests as well as visual and nondestructive examinations. The supplementary tests and certification requirements include such things as test reports, increased sampling, and qualifications of the supplier.

Section 15 summarizes the technical and quality requirements that relate to common fastener applications. This chapter also includes summaries of various specifications. These summaries should be useful for determining the significant requirements for the fastener material when used in conjunction with the ASTM A, ASME SA, and pertinent ASME and Code requirements.

Significant changes to the ASTM specifications are noted to assist with determination if a material certified in a particular version of the Code meets the requirements of another version. These summary specifications are not intended to take the place of other Codes and specifications.

15.2 Definitions

The following definitions are taken from ASME Section III and are duplicated in this chapter for convenience.

- Certificate of Compliance (COC) (as required by ASME Section III) is a written statement attesting that the material is in compliance with the specific requirements of the material specification for the grade, class, and heat treated condition, as applicable. It is equivalent to an ANSI N45.2.10 Certificate of Conformance but is not equivalent to an ANSI N45.2.10 Certificate of Compliance, which requires both a Certified Material Test Report and a COC.

- Certified Material Test Report (CMTR) is a document attesting that material is in compliance with specified requirements, including the actual results of all required chemical analyses, tests, and examinations in accordance with guidelines of ASME Section III, NCA-3867.4 (NA-3767.4) and Appendix P. No references to NA paragraphs are applicable to plants and equipment with a Code Record earlier than the 1977 Edition of ASME Section III, Division I, Nuclear Power Plant Components, Summer 1977 Addenda.

- Mill Tèst Report (MTR) is a document, prepared by the manufacturer who poured the material melt, attesting that the material has been manufactured in accordance with the requirements of a material specification. The MTR provides the results of the chemical and/or physical tests required by the specification and associates the report with a unique heat number.

- Non-ASME Section III Bolting Material includes ASTM bolt materials covered by the following requirements: ANSI/ASME B31.1 Power Piping; ANSI B31.5 Refrigeration Piping; and AISC Steel Structures.

- Manufacturer Certification (as required by ASTM) consists of objective evidence of conformance of the material to the material specification for grade, class, and heat treated condition, as applicable. A COC or CMTR is a type of Manufacturer Certification. These guidelines differ for utilities committed to ANSI N45.2.10 definitions for Certificate of Compliance and Certificate of Conformance.

  For direct tension indicators, such as load indicator washers, the Manufacturer Certification should consist of a CMTR certifying that the material is in conformance with the requirements for direct tension indicators given in Reference (6). For alternate fasteners such as twist off spline bolts,the Manufacturer Certification should consist of a CMTR certifying that the fasteners are in conformance with the requirements of Section 2(d) and 5(a) of Reference (1).

15.3 Technical and Quality Requirements

This section recommends technical and quality requirements for the procurement and receipt inspections of safety related fasteners. It contains applicable requirements from the following:

- ASME Section III of the Boiler and Pressure Vessel Code, NCA 3800 and Subsections NB, NC, ND, and NF
- 10 CFR 50 Appendix B
- ANSI N45.2
- 10 CFR Part 21.

Table 5-1 summarizes the nondestructive examination, traceability, and certification requirements for bolting materials.

Table 15-1

NDE, Traceability, and Certification Requirements on Bolting Materials

ASME III			Size inch	Required NDE: Material Spec (1)	Required NDE: Code or PO Requirements	Traceability	Certification (2)
ASME III	Class 1		≤1	X	VT	X	Cert. of Comp.
ASME III	Class 1		>1	X	VT, MT or PT	X	CMTR (3)
ASME III	Class 1		>2	X	VT, MT or PT, UT (straight beam radial)	X	CMTR
ASME III	Class 1		>4	X	VT, MT or PT, UT (straight beam, radial/axial)	X	CMTR
ASME III	Classes 2, 3, MC		≤1	X		X	Cert. of Comp.
ASME III	Classes 2, 3, MC		≤1	X	VT	X	CMTR
ASME III	Classes 2, 3, MC		>2	X	UT (for ASME XI IWC 3513)	X	CMTR
ASME III	NF	Class 1	≤1	X	VT		Cert. of Comp.
ASME III	NF	Class 1	>1	X	VT	X	CMTR
ASME III	NF	Class 1	>2	X	VT, MT or PT	X	CMTR
ASME III	NF	Class 1	>4	X	VT, MT or PT, UT (st. beam radial)	X	CMTR
ASME III	NF	Classes 2, 3, MC	≤1	X	VT		Cert. of Comp.
ASME III	NF	Classes 2, 3, MC	>1	X	VT	X	CMTR
ASME III	NG		≤3/8	X	MT or PT	X	Cert. of Comp.
ASME III	NG		>½	X	MT or PT, UT (st. beam radial)	X	CMTR
ASME III	NG		>4	X	MT or PT, UT (st. beam, radial/axial)	X	CMTR
Non-ASME			≤1¼	X			Cert. of Conformance (4)
Non-ASME			>1¼	X	Per purchase order	Per purchase order	

(1) Only the following bolting material specs require NDE:
A325 VT (2.5 AQL)
A490 VT (2.5 AQL)
A490 MT (0.25 AQL)

(2) See NCA-3867

(3) Certified material test report

(4) A certification document, certifying conformance to the requirements of the material specification and purchase order.

In some cases, the recommendations included herein go beyond the minimum requirements of the Code and Regulations listed above. This chapter should be used in conjunction with the summary specifications which provide detailed requirements for individual material specifications. All materials should be produced and certified in accordance with the Code and Regulations listed above. For additional information about the production and certification of materials, consult the applicable documents.

If materials for non-pressure boundary safety related applications are purchased from unapproved vendors, the facility may adopt the requirements contained in this chapter and the applicable summary specification as the dedication criteria. An approved vendor is one which meets the criteria of NCA 3800 and 10 CFR 50 Appendix B, which require the plant owner to control the purchase of equipment. Dedication as defined by 10 CFR Part 21 involves verification of the form, fit, and substance of stock components. Dedication is not recommended nor is it easily accomplished for Section III fasteners over 1 inch in diameter.

This section is not intended to be used for vendor qualification purposes. The information contained assumes that the purchaser has properly qualified the vendor through normal qualification procedures required by the above referenced Code and Regulations.

15.3.1 Chemistry

A heat analysis or product analysis should be performed on each heat lot or shipping lot for non-pressure boundary applications by an approved vendor. The analysis should report all required elements and residuals.

For pressure boundary safety related material over 1 inch in diameter, the analysis should be a heat analysis performed by an approved mill on each heat, or a product analysis by an approved vendor on each piece of starting stock. If a chemical analysis has not been provided by an approved vendor, the facility may perform or subcontract the analysis in accordance with the dedication procedure.

15.3.2 Traceability

Traceability is the ability to verify the history, location, or application of an item by means of recorded identification. For the purposes of this chapter, traceability should be maintained by heat lot for pressure boundary safety related fasteners. The finished product must be traceable to the chemical analysis. Non-pressure boundary safety related fasteners may be traceable to the shipping lot.

A shipping lot is that quantity of materials presented for inspection at one time. Shipping lot traceability may be established by the vendor or the facility.

A heat lot for quenched and tempered material is one diameter of a single heat number of the same prior condition. If batch heat treating is used, the material in a heat lot is processed through one temper batch. If continuous heat treating is used, the material in a heat lot is tempered without interruption for a period not to exceed 24 hours at the same time and temperature.

For all other heat treated material, including austenitic stainless steel, precipitation hardening material, and non-ferrous material, a heat lot is one diameter of a single heat number. If batch heat treating is used, the material in a heat lot is processed as one batch. If continuous heat treating is used, the material in a heat lot is processed without interruption for a period not to exceed 24 hours at the same time and temperature.

For material which does not require heat treating, a heat lot is one diameter of a single heat number.

Traceability to the heat lot should be accomplished by use of a shop traveler or equivalent. This shop traveler should be documented as a quality record in accordance with an approved Quality System Program. The vendor quality system must meet the plant owner requirements in accordance with NCA 3800 and 10 CFR 50 Appendix B.

15.3.3 Heat Treatment

The vendor should record at all times the temperature and cooling media data for heat treated ASTM and ASME Section III materials with a diameter of 1/2 inch and greater. For austenitic stainless steel, the vendor should record the data for materials greater than 1 inch in diameter. The recorded data should include stress relieving, if applicable. A statement of the heat treated condition is acceptable for all other materials.

15.3.4 Identification and Marking

Bolting material should be marked for identification by the manufacturer or supplier in accordance with the applicable material specification and the grade of material. If no specific identification mark is provided in the specification, a marking symbol or code may be used which identifies the material with the material certification. Any such symbol or code should be explained in the CMTR or COC, as applicable. Table 15-2 summarizes the requirements for identification markings.

Table 15-2

Summary of Identification Marking Requirements

Description of Bolting	Code Class	Nominal Diameter	Marking Requirements
ASME Components	1, 2, 3, MC	≤ 1 inch ≥ 1 inch	Note 1 Note 2
ASME NF Component Supports	1, 2, 3, MC	≤ 1 inch ≥ 1 inch	Note 1 Notes 1 and 3
Non-ASME Bolting		All	Note 1

1. *Markings should be in accordance with the material specification. Heat code traceability is not required unless it is a material specification requirement.*
2. *Markings should be in accordance with the material specification. Heat code traceability is required.*
3. *If a CMTR is required, heat code traceability is required.*

Bolting furnished with a CMTR should be further identified by the manufacturer or supplier with the heat number or heat code, when required by the material specification or by ASME Code Section III. The bolting should also include any additional marking required to facilitate traceability of the material to the reports of the results of all tests and examinations performed on the material.

Bolting material furnished with a COC does not require heat code traceability.

SA or A307 bolt heads should be marked by the manufacturer selected method to identify the material manufacturer. This is the only bolting material specification which does not require additional identification marking. Thus, absence of additional markings identifies the nut and bolt as SA or A307, Grade A. SA or A307, grade B bolting material should be marked by the utility prior to installation with the same form of unique identification.

Bolting material should be marked by a method that will permanently identify the material and will not result in any harmful contamination or discontinuities. Stamping, when used, should be done with low stress stamps.

The facility should establish a control procedure to ensure material control and identification to the point of installation.

Material furnished bundled or boxed should be tagged on the container with the specification, grade, heat number, heat code or trace letter, quantity, and product description.

When upgrading of stock material is performed per NCA3867.4(e), each piece of starting stock should be marked with a unique trace letter. This trace letter should be transferred to each piece of finished product manufactured from the applicable piece of starting stock. If trace letters are used, heat codes are not required.

15.3.5 Mechanical Properties

The vendor or approved subcontractors should perform all mechanical tests required by the material specification or the purchase order in accordance with an approved Quality Assurance System.

If the material specification requires only machine specimen tensile testing, the vendor should also perform full size testing of tensile strength and proof load, per ASTM F606, for all externally threaded fastener sizes through 1 1/4 inches in diameter. Proof load should be determined as 95% of the required yield strength. Austenitic stainless steels are exempt from a full size tensile test. However, they do require proof load testing at 70% of the minimum yield strength.

If the material is procured from an unapproved source, the facility may perform or subcontract the mechanical tests on the finished product as part of the dedication procedure.

15.3.6 Supplementary Hardness Testing

If the facility requires hardness testing in addition to the specification requirements, the supplemental testing requirements should be indicated on the purchase order. The purchase order should state whether the supplemental hardness testing will be performed by the facility on receipt of the material and/or by the vendor or manufacturer before the material is delivered.

All material should be capable of meeting the hardness values required by the material specification. If no hardness is required by the specification, the material should be capable of meeting a hardness value established in ASTM A370 by converting the specification required tensile strength to the corresponding hardness. Testing should be performed per ASTM F606. The number of pieces to be tested with related acceptance criteria is given in Table 15-3.

Table 15-3

Maximum Acceptable Failures for Hardness Testing

Lot Size	Number Inspected	Maximum Acceptable Failures
1	1	0
2-8	2	0
9-15	3	0
16-25	5	0
26-50	8	0
51-90	13	0
91-150	20	0
151-280	32	1
281-500	50	1
501-1200	80	2
1201-3200	125	3
3201-10,000	200	5
10,001-35,000	315	7
35,001-150,000	500	10
150,001-500,000	800	14
over 500,000	1250	21

If the material fails to meet the supplemental hardness requirement, including the arbitration hardness described in ASTM F606, acceptance should be based on a tensile test or a proof load performed on the product having the most extreme variation in high and/or low hardness. The organization who performed the hardness tests is responsible for arranging any additional mechanical testing required to determine product acceptance.

If material in which hardness is the only required mechanical property fails to meet this supplemental hardness requirement, acceptance should be based on performing the arbitration hardness described in ASTM F606 on all failed pieces. If the arbitration hardness values are below the specification requirement, the material should be rejected.

Austenitic stainless steel materials which have a required minimum tensile strength and a maximum hardness should be accepted or rejected by the arbitration hardness being performed on each discrepant piece. Material which does not have a required tensile strength, proof load, or hardness is not subjected to supplemental hardness testing.

15.3.7 Charpy Impact Testing

Alloy materials which are larger than 1 inch in diameter should meet the impact requirements of ASME Section III, Subsection NB-2330 at a maximum temperature of 0°F. If a different temperature is required for the test, it should be specified in the purchase order.

Other materials which require impact testing under ASME Section III should meet the requirements of the appropriate section (e.g., NC-2330, ND-2330, etc.) at a maximum temperature specified in the purchase order.

This requirement is in addition to any impact testing required by the material specification.

15.3.8 Workmanship

The material should be free of loose scale, excessive burrs, laps, seams, cracks, bursts, voids, or other defects which would render the material unsuitable for the intended application.

15.3.9 Calibration

All equipment used by the vendor or approved subcontractor to test, examine, measure, or process materials should be in current calibration at the time of use and traceable to a national Standard, where such standards exist.

15.3.10 Dimensional Inspection

The vendor should perform a dimensional inspection of all finished materials in accordance with a sampling plan based on a MIL-STD-105, Level II inspection. Table 15-4 summarizes this dimensional inspection.

Table 15-4	
Summary of a Dimensional Inspection	
Lot Size	Number Inspected
1	1
2-8	2
9-15	3
16-25	5
26-50	8
51-90	13
91-150	20
151-280	32
281-500	50
501-1200	80
1201-3200	125
3201-10,000	200
10,001-35,000	315
35,001 - 150,000	500
150,001 - 500,000	800
Over 500,000	1250

This inspection should document all required dimensions from the applicable ANSI Standard on an inspection report. This inspection report becomes a quality record and is maintained under the vendor Quality Assurance System. No rejects are allowed in the sample inspected.

15.3.11 Visual Inspection

Material should be 100% visually inspected in the finished condition for the following defects: cracks, seams, bursts, voids, folds, excessive tool marks, burrs, scale, and nicks. Material with visual defects should be rejected. ASTM F788 gives guidance on surface defects.

15.3.12 Nondestructive Examination

15.3.12.1 Documentation

Documentation of supplier and plant or construction site nondestructive examination (NDE) results should be on file and available in the plant documentation system. The NDE includes a visual examination.

15.3.12.2 Procurement

Bolting for components, component supports, and structural assemblies should be purchased with the required NDE for the particular class of material. The required examination should be conducted and certified by the bolting material supplier. Alternatively, the NDE may be performed and documented on site for ASME purchased bolting material or bolting material manufactured on site which is required to be graded or classified in accordance with ASME Section III.

15.3.12.3 Examination

All bolting material must meet the NDE requirements of the materials specification and any specified purchase order requirements. In addition, ASME bolting must conform to the applicable ASME Section III requirements for the Code of record for the facility. When the requirements include an examination also required by the material specification, only one examination is necessary. However, the more stringent acceptance standards should apply.

15.3.12.4 ASME Bolting Material

The NDE of ASME bolting material should conform to the requirements of the ASME Boiler and Pressure Vessel Code, using ultrasonic, magnetic particle examination, and/or liquid penetrant examination as required. Acceptance requirements should be in accordance with the applicable subsection of the Code.

If the manufacturer did not certify the performance of a visual examination, then any bolting material to be used in ASME Section III applications should be visually examined by the facility prior to use. This examination should be performed in accordance with the requirements of ASME Section III, NX-2580 by certified visual examination personnel, as applicable.

15.3.12.5 Non-ASME Bolting Material

NDE requirements for non-ASME bolting, unless required by the material specification, should be specified on the procurement specification or in the applicable specifications. It is recommended that NDE personnel other than visual inspectors should be properly qualified in accordance with the applicable codes and standards or if none are applicable ASTM TC-1A as modified by ASME Section III, Subsection NB-5521 of the latest edition of the Code is suggested.

15.3.13 Quality Assurance Requirements

All pressure boundary safety related materials should be provided in accordance with the vendor Qualified System Program. When processes and/or operations are subcontracted, they should be accomplished by approved subcontract vendors, using approved procedures, whose quality systems have been qualified on the basis of an acceptable and documented survey, audit, or evaluation. The surveys, audits, or evaluations should be conducted in accordance with the applicable Quality Assurance requirements.

Traceability of the calibration to National Standards, where such standards exist, should be documented and retained as a quality record.

All material over 1 inch nominal diameter should be heat treated, if applicable, in ovens which are properly surveyed or which use a recording thermocouple in contact with the material during the heat treating process. The temperature measurement and recording devices should be certified and under current calibration at the time of use.

For pressure boundary safety related material over 1 inch nominal diameter, the shop traveler or equivalent records should provide a means for determining which linear measurement devices and heat treating ovens were used. The Quality Assurance Program of all parties performing work from the mill through the final shipment should be surveyed and approved by the vendor. If this latter requirement is not met, the vendor should upgrade by performing a product analysis and all other requirements of the material specification on each piece of stock material. If traceability has been established, all other requirements of the material specification may be performed on each heat lot of material versus each piece of stock. These requirements should be performed in accordance with the manufacturer approved Quality Assurance System. Welding can be detrimental to certain grades of bolting material (e.g., without proper care welding on low alloy steels can crack the material). Consequently, when material is upgraded, the vendor should also obtain objective evidence from the raw material producer that no welding has been performed on the heat of stock material.

When materials are procured from unapproved sources, the facility may dedicate the material by performing the analyses, tests, and examinations found in this Standard as part of the receipt inspection.

The requirements of 10 CFR Part 21 are imposed after dedication. Quality records should be retained for a minimum of five (5) years.

15.3.14 Certification Requirements

Each heat lot or shipping lot should be certified by reporting the heat number or lot number and the chemical results from the heat analysis or product analysis. The certification should indicate which method was used.

If upgrading was used, the trace letter from each piece of starting stock should be reported on the corresponding check analysis test report and should be marked on each piece of finished product. The heat analysis should also be reported.

Actual time, temperature, and cooling media data should be reported on the certification for all pressure boundary safety related heat treated materials of diameter 1/2 inch and greater (larger than 1 inch in diameter for austenitic stainless steel or non-pressure boundary safety related materials). If more than one heat treating cycle is used, the number of cycles should also be reported. For other material, the heat treated condition should be reported if applicable.

Chemical and mechanical test results should be reported for each test performed. The certification should also report the applicable grade mark, manufacturer mark, heat code, and trace letter. Additional markings should be explained on the certification. Plating, when required, must be certified to the specification invoked in the purchase order.

A visual inspection report for all materials and NDE reports for materials so examined should be submitted with the certification. The certification should also contain a statement of compliance to the material specification, the approved Quality Assurance System, and applicable invoked Code and Regulatory requirements.

No welding should be performed on this material.

Material furnished from unapproved sources should be certified in accordance with the applicable material specification and dedicated by the facility.

All non-ASME Section III bolting material, regardless of size, should be procured with a certification by the manufacturer that the material is in accordance with the material specification; type, grade, or class; and heat treated condition, as applicable.

15.4 Summary Specifications for Carbon and Low Allow Steel Bolts and Studs

15.4.1 Scope

This summary specification is a condensed version of ASTM specification requirements. The intent of this summary specification is to assist the facility in determining the significant specification requirements pertinent to procurement and receipt inspection.

This summary specification includes the following information:

- Unplated bolts and all thread studs and rods in sizes from 1/4 inch through 4 inches nominal diameter
- Requirements contained in all ASTM specification revisions from 1971 through the years referenced after the specifications listed below
- Significant changes to the specifications listed by revision year.

The specification covers a variety of carbon and low alloy steel bolting materials. These materials are commonly used in structural, piping, and mechanical applications. The materials covered by this specification include:

- ASTM A307 Grade B (1986a)
- ASTM A325 Type 1 (1986)
- ASTM A449 Type 1 (1987)
- ASTM A490 Type 1 (1985).

15.4.2 General Information

The following is general information about the materials covered in this specification:

- ASTM A307 Grade B is a low carbon steel bolting material specified for low pressure piping applications (e.g., those in which flanges are made of cast iron) and other general applications. Bolts should be a heavy hex with regular thread length.
- ASTM A325 Type 1 is a medium carbon, quenched and tempered high strength steel bolting material. This material is often used for structural applications and may be used at elevated temperatures. Bolts should be a heavy hex with structural thread length.
- ASTM A449 Type 1 is a medium carbon steel high strength bolting material specified for general applications. Type 1 bolts are recommended for elevated temperature applications. Bolting materials are usually hex cap screws with regular thread length.
- ASTM A490 Type 1 is an alloy steel high strength bolting material specified for structural applications.

15.4.3 Ordering Information

The purchase order should include the following information:

- Appropriate specification (e.g., ASTM A307 Grade B)
- Quantity
- Required product type (e.g., hex bolts: 1/4 inch through 7/16 inch in diameter; heavy hex bolts: 1/2 inch through 1 1/2 inches in diameter). (See the Specification for information on other product types and special requirements for plated and coated bolts and studs.)

- Dimensions (e.g., nominal diameter, threads per inch (UNC for all sizes), thread fit (Class 2A per ANSI B1.1), and length. Thread pitch and fit may be otherwise specified.

- Specification Year. The latest specification year will be furnished unless otherwise specified.

- Recommended Requirements. Invoke the recommended technical requirements of this section and the technical and quality requirements in Section 15.3.

- Certification. Invoke the certification requirements of this section.

- Hardness. The specification requires hardness testing. The user may also invoke the supplementary hardness requirements in Section 15.3.6.

- Supplementary requirements. The ASTM A307 specification contains a supplementary requirement for bolts suitable for welding. If welding may be performed on the bolting material, the facility must invoke this supplementary requirement.

  The ASTM A325 specification contains a supplementary requirement for bolts threaded full length. If these bolts are required, the user must invoke this supplementary requirement.

15.4.4 Pertinent Specification Requirements

The following are summaries of the requirements contained in the latest specification revision. These summaries do not include most mill processing requirements and other information considered inconsequential to the facility.

- Chemical Composition. Each material should conform to the chemical composition of the appropriate ASTM specification. The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.

  Resulfurized A307 material is not subject to rejection based on product analysis for sulfur.

  For A490 Type 1, the steel should contain sufficient alloying elements to qualify it as an alloy steel. The manufacturer or the vendor should provide the name of the alloy used to produce the bolts and the allowed chemical limits for all required elements.

- Mechanical Properties. All completed bolts or studs should be capable of meeting the hardness requirements of the appropriate ASTM specification. Bolts having a length of less than 3 diameters or with drilled or undersized heads are subject only to hardness testing.

All completed bolts or studs should be capable of meeting the tensile requirements of the appropriate specifications. Full size testing is preferable.

— Hardness for decarburization. The surface hardness of A490 Type 1 bolts, measured at a maximum of 0.003 inch from the surface, should not be more than three (3) Rockwell C points, or the equivalent, higher than the hardness taken at a distance of 1/8 inch from the surface. Both hardness readings should be taken simultaneously on the same axial longitudinal section through the threaded portion of the bolt. In addition, the same hardness scale should be used for both readings.

— Wedge tensile test. A full size tensile testing is preferable. However, when equipment of sufficient capacity for full size testing is not available or when the length of the bolts makes full size testing impractical, machined specimen tensile test may be conducted to meet the requirements of the particular specification.

— Conflicting test data. If both full size testing and machined specimen testing are performed, the full size results take precedence. Tensile results take precedence over low hardness readings.

— Test methods and speed of testing. Mechanical testing should be performed per ASTM F 606. The maximum speed of a free running cross head should be 1 inch per minute for tensile testing.

— Number of mechanical tests. Mechanical tests should be performed on samples randomly chosen by the manufacturer during continuous mass production. When specified in the purchase order, the manufacturer should furnish a test report certified to be the last complete set of mechanical tests for each stock size in each shipment. If the production (i.e., heat lot) method was used for certification, it should be indicated on the purchase order.

Table 15-5 gives the number of additional tests which are specified in the purchase order to be performed for each requirement.

Table 15-5

Number of Additional Mechanical Tests Required

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

For non-pressure boundary safety related fasteners, a lot should consist of all material offered for inspection at one time with the following common characteristics: product type, size, and length. A325 and A490 bolts may be certified under the production (i.e., heat lot) method.

- Retests. If any machined test specimen shows defective machining, it may be discarded and another specimen substituted.
- Marking. All bolts should be marked on the top of the head with the manufacturer identifying symbol and grade marking. A307 bolts do not require a grade marking.
- Certification. A copy of the inspection and test reports for each lot should be provided to the facility when specified in the purchase order. Individual heats of steel are not customarily identified since production is normally not controlled by heat number.

- Other
  - Dimensions. See the appropriate ASTM specification for the ANSI standard governing the fastener dimensions.
  - Nondestructive examination. See ASTM A490 for requirements on magnetic particle inspection for surface discontinuities, visual inspection for head bursts, and increased sampling for dimensional inspection.
  - Inspection. All required tests and inspections should be conducted prior to shipment. The purchaser inspector should have full access to the manufacturing and/or vendor facility at all times while work is being performed on the order.
  - Rejection and rehearing. Material found to be defective by the facility may be rejected. Any rejection should be promptly reported in writing to the manufacturer or vendor. In the case of dissatisfaction with the facility findings, the manufacturer or vendor may make a claim for a rehearing.
  - Nuts. See the appropriate ASTM specification for details on nut selection. The bolting matrix, given in Section 14.3, shows suitable nuts.

15.4.5 Recommended Technical Requirements

The adoption of these requirements will enhance the quality, traceability, and reliability of materials intended for nuclear applications. Since the specification was not written exclusively for nuclear applications, the recommendations in this section are not specification requirements. For items marked with "*", refer to Section 15.3.

15.4.5.1 Chemical Analysis

The vendor should provide a heat analysis or product analysis, performed by an approved vendor, for each heat lot. The analysis should report all required elements and residuals. Table 15-6 gives the recommended number of tests if product analyses are performed.

Table 15-6	
Number of Tests for Product Analyses of Carbon and Low Alloy Steel Bolts and Studs	
Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

15.4.5.2 Traceability

Traceability should be maintained by heat lot or by shipping lot.

15.4.5.3 Mechanical Properties

- Hardness. All completed studs should be subjected to the same hardness testing as bolts and should meet the same requirements.
- Tensile. All bolts of at least 3 diameters in length and studs of at least 4 diameters in length, through 1 1/2 inches in diameter, should be tested full size.

15.4.5.4 Charpy Impact Testing

Carbon steel will not predictably meet charpy impact testing requirements. If impact tests are required, consideration should be given to the use of ASTM A193 Grade B7.

15.4.5.5 Marking

Traceability should be established by marking all bolts of 1/4 inch in diameter and larger and all studs of 3/8 inch in diameter and larger with a unique heat code.

15.4.5.6 Heat Treated Data

The vendor should record all time, temperature, and cooling media data for heat treated ASTM and ASME Section III pressure boundary safety related material whose diameter is 1/2 inch and larger (1 inch in diameter for austenitic stainless steel and non-pressure boundary safety related). The recorded data should include stress relieving, if applicable. A statement of the heat treated condition is acceptable for all other materials.

No heat treating requirement exists for A307 bolting.

15.4.5.7 Workmanship

Bolts should exhibit a workmanlike finish. For additional requirements, refer to Section 15.3.

15.4.5.8 Inspection

The facility should have full access to the manufacturer and/or vendor facility at all times while work is being performed on an order.

Dimensional inspection should be per MIL-STD-105, Level II. See Section 15.3.10 for a sampling plan.

Visual inspection should be 100% for head bursts and the other criteria outlined in Section 15.3.

15.4.5.9 Supplementary Hardness

The user may choose to invoke supplementary hardness requirements.

15.4.5.10 Rejection and Rehearing

The results of tests, inspections, or reviews performed by the facility may constitute a basis for rejection.

15.4.5.11 Certification

The vendor certification should include the following:

- Material description: diameter, thread pitch, and length
- Specification and revision
- Quantity certified
- Heat number or shipping lot number
- Marking on the part
  - Manufacturer mark
  - Grade mark
  - Unique heat code
- Chemical analysis: results of all tests and the method of analysis
- Heat treatment data
- Mechanical test results for each test performed
  - Final hardness
  - Full size wedge tensile and proof load or alternate proof load
  - Machined specimen tensile test if performed

- — Supplementary hardness results if invoked
- — Impact test results if applicable

- Visual inspection report

- Statement of compliance to the approved Quality Program and the technical and quality requirements given in Section 15.3.

15.4.6 Summary of Changes to Materials Specifications

The following is a summary of significant changes to material specifications through the listed edition. If later editions of these specifications are used, the reader is cautioned that a review should be conducted to determine subsequent changes.

15.4.6.1 Summary of Changes for ASTM A307 Grade B

The following are significant ASTM A307 specification changes made between 1971 and 1986:

- Chemical composition
  - — 1971-1976: Restrictions on the intentional inclusion of bismuth, selenium, tellurium, or lead were not included.

- Mechanical properties
  - — Hardness
    - — 1971-1976: Brinell hardness of 121 to 207
  - — Full size tensile test
    - — 1971-1976: Full size testing was not required for bolts 1 1/4 inches in diameter and larger.
  - — Machined specimen tensile test
    - — 1971-1976: 1 1/4 and 1 3/8 inch diameter bolts could be tested in this manner.
  - — Test methods
    - — 1971-1984: Testing was to be conducted per ASTM A370.

- Other
  - — 1971-1982: All tests except product analysis should be made at the place of manufacture. A provision for rehearing was not included.
  - — 1971-1982: A 30 day limit for reporting defects to the manufacturer was in place.
  - — 1971-1978: The specification included nuts.

15.4.6.2 Summary of Changes for ASTM A325 Type 1

The following are significant ASTM A325 Type 1 specification changes made between 1971 and 1986:

- Ordering information
  - — 1971-1976: The manufacturer was required to supply Type 1 bolts if the purchase order did not specify the type.

- Chemical composition
  - Heat analysis
    - 1971-1986: The following were percentages for heat analysis:
      - Carbon: 0.30% minimum
      - Manganese: 0.50% minimum
      - Phosphorus: 0.040% maximum
      - Sulfur: 0.050% maximum
    - 1971-1976: Restrictions on the intentional inclusion of bismuth, selenium, tellurium, or lead were not included.
  - Product Analysis
    - 1971-1986: The following were percentages for product analysis:
      - Carbon: 0.27% minimum
      - Manganese: 0.47% minimum
      - Phosphorus: 0.048% maximum
      - Sulfur: 0.058% maximum

- Mechanical properties
  - Hardness
    - 1971-1976: For ASTM A325 Type 1 bolting materials of diameter 1/2 inch to 1 inch, the Rockwell "C" hardness was specified as 23 to 35, and the Brinell hardness was specified as 241 to 331.
  - Full size tensile testing
    - 1971-1976: Testing was not required for bolts 1 1/4 inches in diameter and larger.
    - 1971-1980: The full size proof load or alternate proof load was required for bolts which required full size testing.
  - Machined specimen tensile testing
    - 1971-1976: 1 1/4 inches in diameter bolts could be tested in this manner.
  - Conflicting test data
    - 1971-1976: Full size testing was not mentioned as taking precedence over machined specimen testing. For larger bolts, tension testing was not mentioned as taking precedence over low hardness readings.
  - Test methods
    - 1971-1980: Testing was to be conducted per ASTM A370.
  - Retest
    - 1971-1983: The stipulation that a lot should be rejected if any of the test specimens fail the applicable test requirements was not mentioned.
  - Visual inspection for head bursts
    - 1971-1976: This inspection was not required.
  - Certification
    - 1971-1974: Certification was not referenced.

- Other
  - 1971-1980: All tests except product analysis should be made at the place of manufacture.
  - 1971-1983: A 30 day limit for reporting defects to the manufacturer existed. A provision for rehearing was not included.
  - 1971-1978: Suitable nuts and washers were included.

15.4.6.3 Summary of Changes for ASTM A449 Type 1

The following are significant ASTM A449 specification changes made between 1971 and 1987:

- Chemical composition
  - Heat analysis
    - 1971-1976: Restrictions on the intentional inclusion of bismuth, selenium, tellurium, or lead were not included.
- Mechanical properties
  - Hardness
    - 1971-1976: For ASTM A449 bolting materials of diameter 1/2 inch to 1 inch, the Rockwell "C" hardness was specified as 23 to 32, and the Brinell hardness was specified as 241 to 302.
  - Full size tensile testing
    - 1971-1976: Testing was not required for bolts with drilled or undersize heads or those having a diameter less than 1 1/4 inches
  - Machined specimen tensile testing
    - 1971-1976: 1 1/4 inch diameter bolts could be tested in this manner.
  - Conflicting test data
    - 1971-1976: Full size testing was not mentioned as taking precedence over machined specimen testing. Tension testing was not mentioned as taking precedence over low hardness readings.
  - Test data
    - 1971-1982: Testing was to be conducted per ASTM A370.
  - Retests
    - 1971-1980: All tests except product analysis should be made at the place of manufacture.
    - 1971-1983: A 30 day limit for reporting defects to the manufacturer existed. A provision for rehearing was not mentioned.
    - 1971-1977: Suitable nuts were not referenced.
    - 1971-1982: Suitable washers were not referenced.

15.4.6.4 Summary of Changes for ASTM A490 Type 1

The following are significant ASTM A490 specification changes made between 1971 and 1986:

- Heat treatment
  - 1971-1983a: The minimum tempering temperature was 900°F.
- Mechanical Requirements
  - Hardness
    - 1971-1974: For ASTM A490 bolting materials, the Rockwell "C" hardness was specified as 32 to 36, and the Brinell hardness was specified as 302 to 341.
  - Full size tensile testing

- — 1971-1976: Bolts of 1 1/4 inches in diameter and less required full size wedge tensile and proof load or alternative proof load testing.
- — Conflicting test data
  - — 1971-1976: Tension tensile test was not mentioned as taking precedence over machined specimen testing for larger bolts.
  - — 1971-1978: Tension testing was not mentioned as taking precedence over hardness readings.
- — Test methods
  - — 1971-1980: Testing was to be conducted per ASTM A370.
- — Number of mechanical tests
  - — 1971-1984: The number of required mechanical tests during these years is given in Table 15-7.

Table 15-7

Number of Required Mechanical Tests from 1971 to 1984

Production Lot Method		Shipping Lot Method	
Lot Size	Sample Size	Lot Size	Sample Size
1-800	1	1-150	1
801-8000	2	151-280	2
8001-35,000	3	281-500	3
35001-150,000	8	501-1200	5
Over 150,000	13	1201-3200	8
		3201-10,000	13
		Over 10,000	20

- — Magnetic particle inspection
  - — The sample size and allowed rejects to be used for inspection of longitudinal discontinuities and transverse cracks from 1971 to 1983 is given in Table 15-8.

Table 15-8				
Magnetic Particle Inspection Criteria from 1971 to 1983				
	Sample Size		Allowed Rejects	
Lot Size	1971-1980	1981-1983	1971-1980	1981-1983
1-50	All	All	0	0
51-500	50	50	0	0
501-1200	50	80	0	0
1201-3200	50	125	0	1
3201-10,000	50	200	0	1
10,001-35,000	N/A	315	N/A	2

- Visual inspection for head bursts
  - The sample size and allowed rejects to be used for inspection of head bursts from 1971 to 1983 is given in Table 15-9.

Table 15-9				
Visual Inspection for Head Bursts from 1971 to 1983				
	Sample Size		Allowed Rejects	
Lot Size	1971-1980	1981-1983	1971-1980	1981-1983
1-2	All	All	0	0
2-5	All	2	0	0
6-8	5	2	0	0
9-15	5	3	0	0
16-25	5	5	0	0
26-150	5	20	0	1
151-280	20	32	1	2
281-500	20	50	1	3
501-1200	32	80	2	5
1201-3200	50	125	3	7
3201-10,000	80	200	5	10
10,001-35,000	N/A	315	N\A	14

- Rejection and rehearing
  - 1971-1983: A 30 day limit for reporting defects to the manufacturer existed. A provision for rehearing was not included.

15.5 Summary Specification for Alloy and Stainless Steel Bolts and Studs

15.5.1 Scope

This summary specification is a condensed version of ASTM specification requirements. The intent of this summary specification is to assist the facility in determining the significant specification requirements pertinent to procurement and receipt inspection.

This summary specification includes the following information:

- Screws and bolts, socket head cap screws, all thread studs and rods, and full diameter bars in sizes from 1/4 inch to 4 inches in nominal diameter
- Requirements contained in all ASTM specification revisions referenced after the specifications listed below
- Significant changes to the specification listed by revision year.

This specification covers a variety of alloy and stainless steel bolting materials commonly used in pressure boundary flanged connections. The materials covered by this specification include:

- ASTM A193 Grade B7 (1986)
- ASTM A574 (1983)
- ASTM A193 Grade B8 (1986)
- ASTM A193 Grade B8M (1986)
- ASTM A564 Type 630 (1987a).

15.5.2 General Information

- ASTM A193 Grade B7 bolting material is an alloy steel high strength screw, bolt, stud, or bar specified for high pressure applications where material toughness is required.
- ASTM A574 bolting material is an alloy steel high strength socket head cap screw specified for general and miscellaneous high pressure applications.
- ASTM A193 Grade B8 Class 1 or Class 1A bolting material is an austenitic stainless steel screw, bolt, stud, or bar specified for corrosive environments.

- ASTM A193 Grade B8M Class 1 or Class 1A bolting material is an austenitic stainless steel screw, bolt, stud, or bar specified for corrosive environments. The material performs well in a boric acid environment.
- ASTM A564 Type 630 Condition H1100 bolting material is an age hardened stainless steel high strength material specified for high temperature applications up to 600°F where corrosion resistance is required. This material also performs well in a boric acid environment.

15.5.3 Ordering Information

The facility purchase order should include the following information:

- The appropriate specification (e.g., ASTM A193 Grade B7)
- Quantity. The number of pieces or weight required.
- The required product type (e.g., hex cap screws: 1/4 inch through 7/16 inch in diameter; heavy hex bolts (modified): 1/2 inch through 1 1/2 inches in diameter). See the specification for information on other product types.
- Dimensions. A574 hex socket cap screws require a Class 3A thread fit. All other products require a Class 2A thread fit.

  Full diameter bars are ordered to nominal diameter and length.
- Specification year. The latest specification year will be furnished unless otherwise specified.
- Recommended requirement. Invoke the recommended technical requirements in this section and the technical and quality requirements in Section 15.3.
- Certification. Invoke the certification requirements of this section.
- Hardness. This specification requires hardness testing. The facility may invoke the supplementary hardness requirements from Section 15.3.
- Supplementary requirements. The ASTM specification contains requirements which are considered supplemental, such as high temperature tests, charpy impact tests, (per ASTM A320), marking, stress relief, and magnetic particle examination. If any of these requirements are desired, the facility must invoke them in the purchase order. Information on most of these requirements is provided in Section 15.5.5.
- Internal quality. The producing mill quality control procedures should provide sufficient testing to assure the internal quality of the product. The recommended method is to perform one macroetch test per ASTM 381 for the lesser of each heat or 10,000 pound lot. Visual examination should show no unacceptable

imperfections or distinct zones of solidification. Other methods may be employed if acceptable to the facility.

15.5.4 Pertinent Specification Requirements

The following are summaries of the requirements contained in the latest ASTM specification revision. This summary does not include most mill processing requirements and other information considered inconsequential to the facility.

- Chemical composition. Each material should conform to the chemical composition of the appropriate ASTM specification.

- Mechanical properties
  - Hardness: All completed bolts or studs should be able to meet the hardness requirements of the appropriate ASTM specification.
  - Tensile: All completed bolts or studs should be able to meet the tensile requirements of the appropriate ASTM specification.
  - Marking: All bolts larger than 1/4 inch in diameter and studs larger than 3/8 inch in diameter should be marked with the appropriate grade mark, and the manufacturer identification symbol. In addition, a unique heat code identification should be provided. A574 and A564 Type 630 specifications have no marking requirements.
  - Certification: The manufacturer should report results of chemical analyses and mechanical tests required by the specification. Alternatively, a Certificate of Conformance to the material specifications may be provided if agreed on, in writing, by the purchaser.

- Other
  - Dimensions: See the appropriate ASTM specification for the dimensional standard for the product.
  - Discontinuities: Surface discontinuities should conform to the requirements of ASTM F788. This is a requirement of ASTM 574 and may be imposed as a supplementary requirement for other bolting specifications. Discontinuities should not affect the usability and performance of the screw.
  - Wedge tensile test: The wedge tensile test demonstrates the head quality and ductility of the bolt. It is recommended that a representative sample of bolts be wedge tested per paragraph 3.5 of ASTM Method F606. This test may be a supplementary requirement, since the wedge tensile test is not required by all specifications for all sizes of bolts.
  - Tensile test: Full size tensile testing is preferable. However, when equipment of sufficient capacity for full size testing is not available or when the length of the bolts makes full size testing impractical, machined specimen tensile test may be conducted to meet the requirements of the particular specification.
  - Conflicting test data: If both full size testing and machined specimen testing are performed, the full size results take precedence.
  - Tensile results take precedence over low hardness readings.

- Test methods and speed of testing. Mechanical testing should be performed per ASTM A370. The maximum speed of a free running cross head should be 1 inch per minute for tensile testing.
- Number of mechanical tests: Mechanical tests of A574 material should be performed on samples randomly chosen by the manufacturer during continuous mass production. When specified in the purchase order, the manufacturer should furnish a test report certified to be the last complete set of mechanical tests for each stock size in each shipment. All other materials are tested on a "heat lot" basis.
- Inspection: All required tests and inspections should be conducted prior to shipment. The purchaser inspector should have full access to the manufacturing facility at all times while work is being performed on the order.
- Rejection and rehearing: Material found to be defective by the facility may be rejected. Any rejection should be promptly reported in writing to the manufacturer or vendor. In the case of dissatisfaction with the facility findings, the manufacturer or vendor may make claim for a rehearing.
- Nuts: See the appropriate ASTM specification for details on nut selection. The bolting matrix, Section 14.3, shows suitable nuts.

15.5.5 Recommended Technical Requirements

The adoption of these requirements will enhance the quality, traceability, and reliability of materials intended for nuclear applications. Since the Specification (15.5.4) was not written exclusively for nuclear applications, the recommendations in this section are not specification requirements. For items marked with "*", refer to Section 15.3.

15.5.5.1 Chemical Analysis

The vendor should provide a heat analysis or product analysis, performed by an approved vendor, for each heat lot. The analysis should report all required elements and residuals. Table 15-10 gives the recommended number of tests if product analyses are performed.

Table 15-10

Number of Tests for Product Analyses of Alloy and Stainless Steel Bolts and Studs

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

For pressure boundary safety materials over 1 inch in diameter, the analysis should be a heat analysis performed by an approved mill on each heat or a product analysis performed by an approved vendor on each piece of starting stock.

15.5.5.2 Traceability

Traceability should be maintained by heat lot.

15.5.5.3 Mechanical Properties

All screws or bolts of at least 3 diameters in length and studs or rods of at least 4 diameters in length from 1/4 inch through 1 1/4 inches in diameter should be able to pass full size testing (per ASTM F606).

15.5.5.4 Charpy Impact Testing

Charpy tests should be performed at a maximum test temperature of 0°F for sizes over 1 inch nominal diameter.

If Charpy tests are required, the manufacturer should be notified in the inquiry. Producing the screws from AISI 4340 material will enhance impact results. Austenitic stainless steel (B8 and B8M) bolting material is exempt from Charpy Impact Tests.

15.5.5.5 Marking

Traceability should be established by marking all bolts and studs of 1/2 inch or larger diameter with a unique heat code.

15.5.5.6 Heat Treated Data

The vendor should record all time, temperature, and cooling media for heat treated ASTM and ASME Section III pressure boundary safety related materials whose diameter is 1/2 inch and larger (larger than 1 inch in diameter for austenitic stainless steel or non-pressure boundary safety related). The recorded data should include stress relieving, if applicable. A statement of the heat treated condition is acceptable for all other materials.

For A574 cap screws, the manufacturer or vendor should perform a test for carburization and decarburization for each heat lot in accordance with the ASTM A574 Specification. The number of tests should be the same as required for full size mechanical testing.

15.5.5.7 Workmanship

Bolts should exhibit a workmanlike finish. For additional requirements, refer to Section 15.3.

15.5.5.8 Inspection

The facility should have full access to the manufacturer and/or vendor facility at all times while work is being performed on an order.

Dimensional inspection should be per MIL-STD-105, Level II. See Section 15.3.10 for a sampling plan.

Bolts should be 100% visually inspected for head bursts and other criteria outlined in Section 15.3.

All material over 1 inch in diameter should be inspected by a magnetic particle or liquid penetrant examination.

15.5.5.9 Supplementary Hardness

The facility may choose to invoke supplementary hardness requirements.

15.5.5.10 Rejection and Rehearing

The results of tests, inspections, or reviews performed by the facility may constitute a basis for rejection.

15.5.5.11 Certification

The vendor certification should include the following:

- Material description — diameter, thread pitch, and length
- Specification and revision
- Quantity certified
- Heat number or shipping lot number
- Marking on the part
  - Manufacturer mark
  - Grade mark
  - Unique heat code
- Chemical analysis — results of all tests and the method of analysis
- Charpy Impact Test results if applicable
- Objective evidence of adequate internal material quality
- Heat treatment data
- Mechanical test results for each test performed
  - Final hardness
  - Full size wedge tensile and proof load or alternate proof load
  - Machined specimen tensile test if performed
  - Supplementary hardness results if invoked
- Visual inspection report
- Nondestructive examination report for sizes over 1 inch nominal diameter
- Statement of compliance indicating that the appropriate number of decarburization tests have been successfully performed
- Statement of compliance to the approved Quality Program and the Technical and Quality Requirements*.

15.5.6 Summary of Changes to Materials Specifications

The following is a summary of significant changes to material specifications through the listed edition. If later editions of these specifications are used, the reader is cautioned that a review should be conducted to determine subsequent changes.

15.5.6.1 Summary of Changes for ASTM A193 Grade B7

The following are significant specification changes made between 1971 and 1986:

- Internal quality. No requirements were imposed until 1985.
- Heat treatment
  - 1971-1985: Heat treatment may be by quenching and tempering or normalizing.
    The responsible ASTM subcommittee informally advised against heat treating by normalizing.
  - 1986: Heat treatment must be by quenching and tempering.
- Stress relief
  - 1971-1973: No stress relief was required.
  - 1973a-1979a: Material which was cold drawn after heat treatment should be stress relieved.
  - 1980c-1986: Material which was cold drawn after heat treatment should be stress relieved. the minimum stress relief temperature should be 100°F (55°C) below the tempering temperature. Tests for mechanical properties should be performed after stress relieving.
- Marking
  - 1971-1977a: All material must be marked with the "B7" grade mark and the manufacturer mark.
  - 1978a-1979a: studs under 3/8 inch in diameter are exempt from marking.
  - 1980a-1986: bolts under 1/4 inch in diameter and studs under 3/8 inch in diameter are exempt from marking.
- Chemical composition
  - 1971-1976: Steel to which lead is added should not be used.
  - 1977-1986: The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.
  - Table 15-11 gives the changes to the chemical composition specification for various elements.

Table 15-11				
Changes in Chemical Composition Specification for ASTM A193 B7				
	Percent (%)			
Element	1980c	1978a	1976	1971
Carbon	0.37-0.49	0.37-0.49	0.38-0.48	0.38-0.48
Manganese	0.65-1.10	0.65-1.10	0.75-1.00	0.75-1.00
Phosphorous (maximum)	0.035	0.04	0.04	0.04
Sulfur (maximum)	0.04	0.04	0.04	0.04
Silicon	0.15-0.35	0.15-0.35	0.15-0.35	0.20-0.35
Chromium	0.75-1.20	0.75-1.20	0.80-1.10	0.80-1.10
Molybdenum	0.15-0.25	0.15-0.25	0.15-0.25	0.15-0.25

15.5.6.2 Summary of changes for ASTM A574

The following are significant specification changes made between 1971 and 1983:

- Chemical composition
  - 1971-1977: There was no prohibition referenced in the Specification against intentionally adding bismuth, selenium, tellurium, or lead.

- Mechanical properties
  - Final hardness
    - 1971-1975: The size range for required hardness was as follows:
      - For a diameter of #0 to #10, Rockwell "C" should be 39 to 45.
      - For a diameter of 1/4 inch and larger, Rockwell "C" should be 37 to 45.
  - Full Size Wedge Tensile and Proof Load
    - 1971-1975: The minimum wedge tensile and proof load requirements were lower for the sizes listed in Table 15-12.

Table 15-12		
Changes in Minimum Wedge Tensile and Proof Load Requirements from 1971 to 1975		
Size	Minimum Wedge Tensile (pounds)	Minimum Proof Load (pounds)
1/4-20	5,410	4,290
1/4-28	6,190	4,910
5/16-18	8,910	7,070
5/16-24	9,860	7,830
3/8-16	13,200	10,500
3/8-24	14,900	11,900
7/16-14	18,100	14,400
7/16-20	20,200	16,000
1/2-13	24,100	19,200
1/2-20	27,200	21,600
9/16-12	30,900	24,600
9/16-18	34,500	27,400

- Test Method
  - 1971-1980: Mechanical testing was to be conducted per ASTM A370.
- Carburization and Decarburization
  - 1971-1975: The test methods and decarburization limits were different (see the Specification for details).
- Discontinuities
  - 1971-1983: Discontinuities were not required to comply with ASTM F788.
  - 1971-1977: Requirements for thread discontinuities were not mentioned in the Specification.

- Other
  - 1971-1983: Threads were to be UNC or UNF.

15.5.6.3 Summary of Changes for ASTM A193 Grade B8

The following are significant specification changes made between 1971 and 1986:

- Internal Quality. No requirements were imposed until 1985.

- Marking
  - — 1971-1977a: All material must be marked with the "B8" or "B8A" grade mark and the manufacturer mark.
  - — 1978a-1979a: Studs under 3/8 inch in diameter are exempt from marking.
  - — 1980a-1986: bolts under 1/4 inch in diameter and studs under 3/8 inch in diameter are exempt from marking.
- Hardness
  - — Table 15-13 gives the changes in Rockwell "B" and Brinell hardness specifications from 1971 to 1979a.

Table 15-13

Changes in Hardness Specification for ASTM A193 B8

	1971-1973		1973a-1979a	
	Maximum Rockwell B	Maximum Brinell	Maximum Rockwell B	Maximum Brinell
Class 1	91	192	97	223
Class 1A	91	192	91	192

Rockwell hardness values were determined by converting Brinell hardness values using ASTM A370.

- Chemical composition
  - — 1971-1976: The chrome requirement was 8.00 to 12.00 versus 8.00 to 15.00 currently required. Steel to which lead is added should not be used.
  - — 1977-1986: The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.

15.5.6.4 Summary of Changes for ASTM A193 Grade 8M

The following are significant specification changes made between 1971 and 1986:

- Internal Quality. No requirements were imposed until 1985.
- Marking
  - — 1971-1977a: All material must be marked with the "B8M" or "B8MA" grade mark and the manufacturer mark.
  - — 1978a-1979a: Studs under 3/8 inch in diameter are exempt from marking.
  - — 1980a-1986: Bolts under 1/4 inch in diameter and studs under 3/8 inch in diameter are exempt from marking.
- Hardness
  - — Table 15-14 gives changes in Rockwell "B" and Brinell hardness specifications from 1971 to 1979a.

Table 15-14				
Changes in Hardness Specification from 1971 to 1979a				
	1971-1973		1973a-1979a	
	Maximum Rockwell B	Maximum Brinell	Maximum Rockwell B	Maximum Brinell
Class 1	91	192	97	223
Class 1A	91	192	91	192

Rockwell hardness values for 1971-1979a were determined by converting Brinell hardness values using ASTM A370.

- Chemical composition
  - 1971-1976: Steel to which lead is added should not be used.
  - 1977-1986: The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.

15.5.6.5 Summary of Changes for ASTM A564 Type 630

The following are significant specification changes made between 1971 and 1987a:

The Specification invokes all of the requirements of ASTM A484 and specifically references it for finish and marking requirements. Previous editions of the ASTM A484 were not reviewed for changes, so furnish and marking requirements may have varied.

- General requirements
  - 1971-1981: The Specification did not specifically invoke all of the requirements of ASTM A484.
  - 1971-1983: The Specification did not state that,, if the condition was not specifically mentioned in the purchase order, it could be chosen by the manufacturer.
- Heat Treatment
  - 1971-1980: The age hardening temperature was defined as 110°F ±15°F. The holding time at temperature was specified at four (4) hours.
- Mechanical properties
  - Final hardness
    - 1971-1984: The final hardness requirement was as follows:
    - Minimum Rockwell "C": 32
    - Minimum Brinell: 311.
  - Charpy Impact Tests
    - 1971-1983: Charpy impact tests were required to be performed at 70-80°F.

  - 1971-1974: Two Charpy specimens were required for testing.
  - 1974-1983: Three Charpy specimens were required for testing.

  Standard charpy specimens can only be obtained from materials having a 5/8 inch minimum diameter. The Specification required charpy impacts for all sizes. There was no difference to explain how to resolve this conflict.
- Number of mechanical tests
  - 1971-1983: The Specification did not clearly define the number of hardness and machine specimen tensile tests required for age hardened material.
- Marking
  - 1971-1980: Each bundle or box was required to be tagged with the Specification (ASTM A554), type (630), condition (H1100), heat number, and size.

15.6 Summary Specification for Carbon, Alloy, and Stainless Steel Nuts

15.6.1 Scope

This summary specification is a condensed version of ASTM specification requirements. The intent of this summary specification is to assist the facility in determining the significant specification requirements pertinent to procurement and receipt inspection.

This summary specification includes the following information:

- Finished hex nuts, heavy nuts, and jam nuts from 1/4 inch to 4 inches nominal diameter
- Requirements contained in all ASTM specification revisions from 1971 through the years referenced after the specifications listed below
- Significant changes to the specification listed by revision year.

The specification covers a variety of nut materials, commonly used in nuclear power plants. The materials covered by this summary specification include:

- ASTM A194 Grade 7 (1985a)
- ASTM A194 Grade 2H (1985a)
- ASTM A563 Grade A (1984)
- ASTM A563 Grade C (1984)
- ASTM A194 Grade 8 or 8A (1985a)
- ASTM A194 Grade 8M or 8MA (1985a).

15.6.2 General Information

- ASTM A194 Grade 7 nut is an alloy steel high strength nut specified for high pressure applications. These nuts are most frequently used with carbon or alloy steel bolting in pressure piping applications over 1 inch in diameter where impact properties are required.

- ASTM A194 Grade 2H nut is a carbon steel high strength nut specified for high pressure applications. These nuts are most frequently used with carbon or alloy steel bolting in pressure piping applications. If impact properties are required, ASTM A194 Grade 7 nuts are recommended.

- ASTM A563 Grade A nut is a carbon steel nut specified for low pressure piping and general applications. These nuts are most frequently used with low carbon steel bolting such as A307 Grades A and B or A394.

- ASTM A563 Grade C nut is a carbon steel high strength nut specified for structural and general applications. These nuts are most frequently used with carbon or alloy steel bolting such as A325 Type 1 and 2, A449, A354 Grade BC, A307 Grade A and B, A394.

- ASTM A194 Grade 8 or 8a nut is an AISI A304 austenitic stainless steel nut specified for corrosive environments.

- ASTM A194 Grade 8 and Grade 8A nuts have identified chemical and tensile properties. Grade 8 requires no specific heat treatment and is marked "8" to distinguish it from Grade 8A which is solution annealed in the final product form and marked "8A". Either Grade 8 or 8A may be provided unless the facility specifies Grade 8A.

- ASTM A194 Grade 8M (AISI 316) and Grade 8MA have identical chemical and tensile properties. Grade 8M requires no specific heat treatment and is marked "8M" to distinguish it from Grade 8MA which is solution annealed in the final product form and marked "8MA". Either Grade 8M or 8MA may be provided unless the facility specifies Grade 8MA.

- If the material is being purchased for inventory and the facility has flexibility in choosing grades of austenitic stainless steel, consideration should be given to ASTM A194 Grade 8M because of its superior resistance to boric acid. See summary specification ASTM A194 Grade 8M or 8MA.

15.6.3 Ordering Information

The facility purchase order should include the following information:

- Specification. The appropriate specification (e.g., ASTM A193 Grade 7).

- Quantity. The number of pieces required.

- Required Product Type. Heavy hex nuts will be provided unless otherwise specified. Finished hex nuts, heavy hex jam nuts, or finished hex jam nuts must be specified in the purchase order.

  The A563 specification states that the requirements for any grade of nut may, at the vendor option, and with notice to the purchaser, be fulfilled by furnishing nuts of one of the stronger grades specified in the ASTM Specification unless such substitution is barred in the inquiry and purchase order.

- Dimensions. Nominal diameter, threads per inch, and thread fit (i.e., Class 2B per ANSI B1.1). Thread pitch and fit may be otherwise specified in the purchase order.

- Specification Year. The latest specification year will be furnished unless otherwise specified.

- Recommended Requirement. Invoke the Recommended Technical Requirements in this Section and the Technical and Quality Requirements, Section 15.3.

- Certification. Invoke the certification requirements of this section.

- Hardness. The specification requires hardness testing. The facility may invoke the supplementary hardness requirements from Section 15.3.

- Supplementary Requirements. The ASTM specification contains requirements which are considered supplemental, such as chemical certification, proof load testing of large nuts, retest by the purchaser and control of product by heat number. If any of these requirements are desired, the facility must invoke them in the purchase order. Information on most of these requirements is provided in the Recommended Technical Requirements section of this summary specification.

15.6.4 Pertinent Specification Requirements

The following are summaries of the requirements contained in the latest ASTM specification. This summary does not include most mill processing requirements and other information considered inconsequential to the facility.

- Internal Quality. The producing mill quality control procedures should provide sufficient testing to assure the internal quality of the product. The recommended method is to perform one macroetch test per ASTM E381 for the lesser of each heat or 10,000 pound lot. Visual examination should show no unacceptable imperfections or distinct zones of solidification. Other methods may be employed if acceptable to the facility. This provision does not apply to A563.

- Method of Manufacture and Heat Treatment. ASTM A194 Grade 7 nuts should be hot or cold formed or should be machined from hot rolled, hot forged, or cold drawn bars. The material should be reheated above the critical range, quenched in a suitable medium, and tempered at a minimum of 1,100°F. Nuts machined from heat treated bars need not be heat treated again. These nuts should receive a "7B" grade mark to distinguish the method of manufacture.

ASTM A194 Grade 2H nuts should be hot or cold formed or should be machined from hot rolled, hot forged, or cold drawn bars. The material should be reheated above the critical range, quenched in a suitable medium and then tempered at a minimum of 805°F. Nuts machined from heat treated bars need not be re-heat treated. These nuts should receive a "2HB" grade mark to distinguish the method of manufacture.

ASTM A563 Grade A nuts may be made by hot or cold forming, pressing, punching or machining from bar. Heat treatment is not required.

ASTM A563 Grade C nuts may be made by hot or cold forming, pressing, punching or machining from bar. Nuts do not require a specific heat treatment. If nuts are heat treated they should meet the following requirements.

Low carbon steel A563 nuts made of steel having carbon content not exceeding 0.2% phosphorus not exceeding 0.04%, an sulfur not exceeding 0.05% by heat analysis may be heat treated by quenching in a liquid medium from a temperature above the transformation temperature and need not be tempered. When this heat treatment is used, caution should be exercised not to exceed the maximum hardness requirement.

Medium carbon steel nuts made of any permitted steel may be heat treated by quenching in a liquid medium from a temperature above the transformation temperature and tempering at a minimum temperature of 800°F.

Grade 8 and Grade 8M nuts should be hot or cold formed or should be machined from hot rolled, hot forged, or cold drawn bars. There are no heat treatment requirements.

Grade 8A and Grade 8MA after forming or machining, should subsequently be carbide solution annealed by heating them for a sufficient time at a temperature to dissolve the chromium carbides. The nuts should then be cooled in a test medium at a rate sufficient to prevent reprecipitation of the carbides.

Grade 8MA nuts after forming or machining, should subsequently be carbide solution annealed by heating them for a sufficient time at a temperature to dissolve the chromium carbides and cooled in a liquid medium at a rate sufficient to prevent reprecipitation of the carbides.

- Chemical Composition. Each material should conform to the chemical composition of the appropriate ASTM specification.
- Mechanical Properties. All completed nuts should be capable of meeting the hardness requirements of the appropriate ASTM specification. Selected samples of A194 Grade 2H and Grade 77 nuts require a second hardness test after a 24

hour exposure to elevated temperature. All nuts should be capable of meeting the tensile requirements of the appropriate ASTM specification.

A194 Grade 2H and Grade 7 may be subjected to a cone proof load test. This test is mandatory when visible surface discontinuities become a matter of dispute between the manufacturer and the purchaser.

- Number of Mechanical Tests Required. Table 15-15 gives the number of pieces which should be tested.

Table 15-15

Number of Samples Required for Mechanical Tests

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

- Marking. All nuts should be marked with the appropriate grade marking and the manufacturer identification symbol.

  For A563 Grade A, no marking is required. When specified in the purchase order, the nuts should be marked with an "A" grade mark.

- Certification. The manufacturer should furnish either a Certificate of Conformance to the material specification or a Material Test Report providing the results of chemical analyses and mechanical tests as well as the minimum tempering temperature, as specified in the purchase order. Material Test Reports may not contain the heat number since control by heat is not required by the Specification.

- Other. Nuts should be free of defects and exhibit a good commercial finish. If visible imperfections become an issue, the cone proof load test should be employed. If a scale free bright finish is required, it must be specified in the purchase order.

15.6.5 Recommended Technical Requirements

The adoption of these requirements will enhance the quality, traceability, and reliability of materials intended for nuclear applications. Since the Specification was not written exclusively for nuclear applications, the recommendations in this section are not specification requirements. For items marked with * refer to Section 15.3.

15.6.5.1 Chemical Analysis

The vendor should provide a heat analysis or product analysis, performed by an approved vendor, for each heat lot. The analysis should report all required elements and residuals. Table 15-16 gives the sample size required if product analyses are performed.

Table 15-16

Number of Samples Required for Product Analyses

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

For materials over 1 inch in diameter, the analysis may be either of the following:

- A heat product analysis performed by an approved mill on each heat
- A product analysis performed by an approved vendor on each piece of starting stock.

15.6.5.2 Traceability

Traceability should be maintained by heat lot.

15.6.5.3 Threaded Mandrel Proof Load Testing of Larger Nuts

A sample of nuts through 1 1/2 inches in diameter should be tested by the manufacturer.

15.6.5.4 Cone Strip Proof Load Test

This test should be performed by the manufacturer regardless of the surface condition of the material. If the specification does not give the cone proof load values, the formula from Table 4 of ASTM A194-85a may be used to calculate the cone proof load.

15.6.5.5 Hardness

Hardness is a specification requirement. The facility may also choose to invoke supplementary hardness requirements.

15.6.5.6 Marking

Traceability should be established by marking all bolts and studs of 1/2 inch or larger diameter with a unique heat code.

15.6.5.7 Heat Treatment Data

Certifications for materials 1/2 inch nominal diameter and larger should report actual heat treatment times, temperatures, and cooling media. A statement of heat treated condition (i.e., quenched and tempered) is adequate for smaller sizes.

15.6.5.8 Charpy Impact Testing

Charpy tests should be performed at a maximum test temperature of 0°F for sizes over 1 inch nominal diameter.

Heat treated A194 2H or A563 Grade A and Grade C carbon steel nuts will not predictably meet impact requirements. It is recommended that ASTM A194 Grade 7 be specified if impact properties are required.

Austenitic stainless steels, such as A194 Grade 8, 8A, 8M, and 8MA, are exempt from impact tests.

15.6.5.9 Inspection

Dimensional Inspection should be per MIL-STD-105, Level II (See Section 15.3.10).

Visual inspection should be 100%.

All material over 1 inch nominal diameter should receive a magnetic particle or liquid penetrant examination.

15.6.5.10 Rejection

The results of tests, inspections or reviews performed by the facility should also constitute a basis for rejection.

15.6.5.11 Certification

The vendor certification should include the following:

- Material description: Product type, diameter, and thread pitch
- Specification and revision
- Quantity certified
- Heat number or shipping lot number
- Marking on the part
  - — Manufacturer mark
  - — Grade mark
  - — Unique heat code

- Chemical analysis: Results of all tests and the method of analysis
- Charpy results if applicable
- Provide objective evidence of adequate internal material quality
- Heat treatment data
- Mechanical test results for each test performed
  - Final hardness
  - Full size wedge tensile and proof load or alternate proof load
  - Machined specimen tensile test if performed
  - Supplementary hardness results if invoked
- Visual inspection report
- Nondestructive examination report for sizes over 1 inch nominal diameter
- Statement of compliance indicating that the appropriate number of decarburization tests have been successfully performed
- Statement of compliance to the approved Quality Program and the Technical and Quality Requirements.

15.6.6 Summary of Changes to Materials Specifications

The following is a summary of significant changes to material specifications through the listed edition. If later editions of these specifications are used, the reader is cautioned that a review should be conducted to determine subsequent changes.

15.6.6.1 Summary of Changes for ASTM A194 Grade 7

The following are significant specification changes made between 1971 and 1985a:

- Internal Quality. No requirements were imposed until 1985.
- Method of manufacture and heat treatment
  - 1971-1973: All nuts made by cold forging or machining from cold drawn bars should be stress relieved at 1,000°F minimum. Nuts made from hot-forged or hot rolled bars are exempt from stress relief requirements.
- Chemical composition
  - 1971-1976: Steel to which lead is added should not be used.
  - 1977-1986: The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.
  - 1985: A prohibition was imposed against choosing a starting material which requires the addition of any element not mentioned in the Chemical composition of the Specification.

— Table 15-17 gives changes in chemical composition specifications for ASTM A194 Grade 7.

Table 15-17

Changes in Chemical Composition Specification from 1971 to 1977a

Element	1971/1977a
Carbon	0.38%/0.48%
Manganese	0.75%/1.00%
Phosphorus	0.045% maximum
Sulfur	0.04% maximum
Silicon	0.20%/0.35%
Chromium	0.80%/1.10%
Molybdenum	0.15%/0.25%

- Mechanical properties
  - 1971-1973: The cone strip proof load test was mandatory. The threaded mandrel proof load test was not required. The Specification did not contain a recommended sample size for the number of required mechanical tests.
  - 1971-1979: Jam nut proof load testing was not required.

15.6.6.2 Summary of Changes for ASTM A194 Grade 2H

The following are significant specification changes made between 1971 and 1985a:

- Internal Quality. No requirements were imposed until 1985.

- Method of manufacture and heat treatment
  - 1971-1973: All nuts made by cold forging or machining from cold drawn bars should be stress relieved at 1,000°F minimum. Nuts made from hot forged or hot rolled bars are exempt from stress relief requirements.

- Chemical composition
  - 1971-1976: Steel to which lead is added should not be used.
  - 1977-1986: The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.
  - 1985: A prohibition was imposed against choosing a starting material which requires the addition of any element not mentioned in the Chemical composition section of the Specification.

- Mechanical properties
  - 1971-1973: The cone strip proof load test was mandatory. The threaded mandrel proof load test was not required. The Specification did not contain a recommended sample size the number of required mechanical tests.
  - 1971-1979: Jam nut proof load testing was not required.

15.6.6.3 Summary of Changes for ASTM A563 Grade A

The following are significant specification changes made between 1971 and 1984.

- Mechanical properties
  - 1971-1976a: Nut finish was not included in the "lot" definition to determine the number of additional tests required.
  - 1971-1976a: The speed of testing was not discussed in the specification.
  - 1977-1978a: Testing was to be conducted per ASTM A370.

- Marking
  - 1971-1976a: No manufacturer mark was required.
  - 1971-1976a: Markings, if any, could be raised or depressed on the bearing surface.

15.6.6.4 Summary of Changes for ASTM A563 Grade C

The following are significant specification changes made between 1971 and 1984:

- Mechanical properties
  - 1971-1976a: Nut finish was not included in the "lot" definition to determine the number of additional tests required.
  - 1971-1976a: The speed of testing was not discussed in the specification.
  - 1977-1978a: Testing was to be conducted per ASTM A370.

- Marking
  - 1971-1976a: No manufacturer mark was required.
  - 1971-1976a: Grade markings could be raised or depressed on the bearing surface.

- Recommended Bolting Materials
  - 1971-1976a: ASTM A354 Grade BC was not listed as a suitable bolting material for A553 Grade C nuts.

15.6.6.5 Summary of Changes for ASTM A194 Grade 8 or 8A

The following are significant specification changes made between 1971 and 1985a:

- Internal Quality. No requirements were imposed until 1985.
- Method of manufacture and heat treatment
  - 1971-1973: Grade 8A did not exist. All Grade 8 nuts required heat treatment, except air cooling was allowed.

- Chemical composition
  - 1971-1973: The nickel content was 8.00% to 12.00%.
  - 1971-1976: Steel to which lead is added should not be used.
  - 1977-1986: The intentional addition of bismuth, selenium, and tellurium or lead is prohibited
  - 1985: A prohibition was imposed against choosing a starting material which requires the addition of any element not mentioned in the Chemical composition section of the Specification.

- Mechanical properties
  - 1971-1975: No proof load testing was required.
  - 1971-1973: The final hardness for Grade 8 nuts changed as follows:
    - Rockwell "B" hardness was 60 to 90.
    - Brinell hardness was 126 to 192.
  - 1974-1975: The final hardness for Grade 8 nuts changed as follows:
    - Rockwell "B" hardness was 60 to 96
    - Brinell hardness was 126 to 223.

    The hardness requirements for grade 8A have not changed.
  - 1971-1973: The number of mechanical tests was not specified.
  - 1971-1979: Jam nut proof load testing was not required.

15.6.6.6 Summary of Changes for ASTM A193 Grade 8M or 8MA

The following are significant specification changes made between 1971 and 1985a:

- Internal Quality. No requirements were imposed until 1985.
- Method of manufacture and heat treatment
  - 1971-1973: Grade 8MA did not exist. All Grade 8M nuts required heat treatment; air cooling was allowed.

- Chemical composition
  - 1971-1976: Steel to which lead is added should not be used.
  - 1971-1986: The intentional addition of bismuth, selenium, tellurium, or lead is prohibited.
  - 1985: A prohibition was imposed against choosing a starting material which requires the addition of any element not mentioned in ASME Section III, Paragraph C.

- Mechanical properties
  - 1971-1975: No proof load testing was required.
  - 1971-1973: The final hardness for Grade 8M nuts changed as follows:
    - Rockwell "B" hardness was 60 to 90
    - Brinell hardness was 126 to 192.
  - 1974-1975: The final hardness for Grade 8M nuts changed as follows:
    - Rockwell "B" hardness was 60 to 96
    - Brinell hardness was 126 to 223.

The hardness requirements for Grade 8MA have not changed.

— 1971-1973: The number of required mechanical tests was not specified.

— 1971-1979: Jam nut proof load testing was not required.

15.7 Summary Specification for Carbon, Alloy, and Stainless Steel Washers

15.7.1 Scope

This summary specification is a condensed version of ASTM and ANSI specification requirements. The intent of this summary specification is to assist the facility in determining the significant specification requirements pertinent to procurement, and receipt inspection.

This summary specification includes the following information:

- Unplated carbon steel and stainless steel circular flat and lock washers from 1/4 inch to 4 inches nominal diameter
- Requirements contained in all ASTM specification revisions from 1976 through the years referenced after the specification listed below
- Significant changes to the specifications listed by revision year.

The specification covers a variety of washer materials commonly used in nuclear power plants. The materials covered by this specification include:

- ASTM F436 (1986)
- ASTM F844 (1983) and ANSI B18.22.1 (1981)
- ANSI B18.21.1 (1983).

15.7.2 General Information

- ASTM F436 is a hardened steel washer for structural and general purpose mechanical applications.
- ASTM F436 superseded ASTM A325 and is suitable for use with most carbon and alloy steel bolting material. Because of the versatility, F436 circular flat washers are recommended as the primary type of carbon steel flat washer for stock.
- ASTM F844 is a specification for flat, unhardened, steel washers for general use. ANSI B18.22.1 is a specification which covers flat washers of various materials.
- ANSI B18.21.1 specification covers helical spring lock washers made from various materials and carbon steel took lock washers.

15.7.3 Ordering Information

The facility purchase order should include the following information:

- Specification. The appropriate specification (e.g., ASTM F436).
- Quantity. The number of pieces required.
- Required Product Type. The required product (e.g., circular flat washers).

  See the Specification for information on other product types, materials, and special requirements for plated/coated washers.
- Dimensions. Nominal diameter.
- Specification Year. The latest specification year (1986) will be furnished unless otherwise specified.
- Recommended Requirement. Invoke the Recommended Technical Requirements in this section and Section 15.3.
- Certification. Invoke the certification requirements of this section.
- Hardness. For hardened washers, the facility may invoke the supplementary hardness requirements from the Technical and Quality Requirements, Section 15.3.
- Supplementary Requirements. The ASTM specification contains requirements for the surface finish of the washers. If these requirements are desired, they must be invoked in the purchase order.
- Chemical Composition. Each material should conform to the chemical composition of the appropriate ASTM or ANSI standard.
- Mechanical Properties. All completed washers should be capable of meeting the hardness requirements of the appropriate ASTM or ANSI specification. No specific processing requirements are imposed except that ANSI B18.21.1 requires control on decarburization.
- Marking. Flat washers should be marked with the manufacturer mark. all markings should be depressed. There is no required marking for lock washers or for ANSI B18.22.1 flat washers.
- Certification. The manufacturer should provide the results of the last completed set of hardness tests for each size and attest to compliance with the Specification. The certification should also provide the material grade. Material Test Reports may not contain the chemical composition or heat number, since control by heat is

not required by the specification. ASTM 844 washers do not require hardness testing.

- Other. All tests and inspections should be conducted at the place of manufacture unless otherwise stated in the purchase order. The purchaser inspector should have full access to the manufacturer facility at all times while work is being performed on his order.

  Washers should be free of excess scale, excess coating and foreign material on bearing surfaces. Arc and gas cut washers should be free of metal spatter.

  Material which fails to conform to the requirements of this specification may be rejected. Rejection should be promptly reported in writing to the manufacturer or vendor. In case of dissatisfaction with the results of the tests, the manufacturer or vendor may make claim for a rehearing.

15.7.4 Recommended Technical Requirements

The adoption of these requirements will enhance the quality, traceability, and reliability of materials intended for nuclear applications. As the Specification was not written exclusively for nuclear applications, the recommendations in this section are not specification requirements. For items marked with "*", refer to Section 15.3.

15.7.4.1 Chemical Analysis

The vendor should provide a heat analysis or product analysis, performed by an approved vendor, for each heat lot or shipping lot. The analysis should report all required elements and residuals.

Table 15-18 gives the number of samples to be tested if product analyses are performed.

Table 15-18

Number of Samples Required for Product Analyses

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

Carbon steel washers to ANSI B18.22.1 or ASTM F844 have no chemistry requirements. They must be a generally recognized carbon steel.

Carbon steel lock washers to ANSI B18.21.1 must have sufficient carbon to attain the desired hardness after heat treating.

15.7.4.2 Traceability

Traceability should be maintained by heat lot.

15.7.4.3 Mechanical Properties

Table 15-19 gives the number of hardness tests for each lot.

Table 15-19

Number of Samples Required for Hardness Tests

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

If any test specimen shows flaws, it may be discarded and another specimen substituted. Should any specimen fail to meet the requirements, double the number of specimens from the same lot should be successfully tested.

15.7.4.4 Hardness

Hardness is a specification requirement. The facility may also choose to invoke supplementary hardness requirements.

15.7.4.5 Marking

All materials should be bagged and tagged for shipment with a label showing the size, description, heat lot or shipping lot, and the customer purchase order number.

15.7.4.6 Heat Treatment Data

Since washers are considered compressional components, no heat treatment data is required.

15.7.4.7 Workmanship

Washers should be free of defects and exhibit a good commercial finish.

15.7.4.8 Inspection

The purchaser should have full access to the manufacturer facility to inspect the work.

Dimensional inspecting should be per MIL-STD-105, Level II (See Section 15.3.10.).

Visual inspection should be 100%.

15.7.4.9 Rejection

The results of tests, inspections, or reviews performed by the facility may constitute a basis for rejection.

15.7.4.10 Certification

The vendor certification should include the following:

- Material description — product type and nominal diameter
- Specification and revision
- Quantity certified
- Heat number or shipping lot number — indicate which
- Marking on the part
  - Manufacturer mark
- Chemical analysis — Results of all tests and the method of analysis
- Mechanical test results for each test performed
  - Final hardness if applicable
  - Supplementary hardness results if invoked
- Visual inspection report
- Statement of compliance to the approved Quality Program and the Technical and Quality Requirements.

15.7.5 Summary of Changes to Materials Specifications

The following is a summary of significant changes to material specifications through the listed edition. If later editions of these specifications are used, the reader is cautioned that a review should be conducted to determine subsequent changes.

15.7.5.1 Summary of Specification Changes for ASTM F436

The Specification was not written until 1976. Hardened flat washers were previously covered by ASTM A325. The following are significant specification changes made between 1976 and 1986:

- Heat treatment
  - 1976-1980: Washers should be thoroughly quenched and tempered.

- Mechanical properties
  - Surface hardness
    - 1976: The Specification required a Rockwell "C" surface hardness of 35 to 45.

    The hardness should be taken at a maximum of 0.003 inch from the surface and should not differ by more than the equivalent of 3 points Rockwell "C" from the core hardness.
  - Core hardness
    - 1976: The Specification required a Rockwell "C" core hardness of 36 to 45.

    A minimum of two readings should be taken 180° apart on at least one face, at a minimum depth of 0.015 inch
  - Test methods
    - 1976: Hardness testing was to be conducted per ASTM A370.
  - Dimensions
    - 1976-1983: The size range covered by the Specification excluded 1/4 inch through 7/16 inch nominal diameter.
  - Workmanship
    - 1976-1982: There was no specific workmanship requirement in the Specification.
  - Rejection
    - 1976-1983: It was necessary for the facility to notify the vendor or manufacturer within 30 days of receipt of material, that there was a reject. This notification need not be in writing and there was no specific recourse on the part of the vendor or manufacturer for rehearing.

15.7.5.2 Summary of Changes for ASTM F844 and ANSI B18.22.1

ANSI B18.22.1 has not changed since 1965. Prior to 1976, it was designated ASA B27-1965. ASTM F844 was not written until 1983 and has never been revised.

15.7.5.3 Summary of Changes for ANSI B18.21.1

ANSI B18.21.1 has not changed since 1972. Prior to that year it was designated ASA B27.1 1965. The following are significant specification changes made in 1972:

- Material. Stainless steel helical spring lock washers could formerly be produced from AISI 420 stainless steel.

- Modifications. Responsibilities for modifying the product were not clarified until 1972.

- Final Hardness. The maximum hardness for helical spring lock washers was formerly Rockwell C53.

- Workmanship and Finish. Specific criteria did not exist for lock washers until 1972.

- Embrittlement. Prior to 1972, no statement was made to prevent possible hydrogen embrittlement if the washers were plated.

- Dimensions. The inside diameters for helical spring lock washers were slightly larger prior to 1972.

15.8 Summary Specification for ANSI B18.6.3 Carbon and Stainless Steel Machine Screws and Nuts

15.8.1 Scope

This summary specification is a condensed version of Specification requirements and dimensional information from ANSI B18.6.3 which covers various types of machine screws and machine screw nuts for general purpose applications.

In addition to machine screws covered by ANSI B18.6.3, there are equivalent types of cap screws which are covered by ANSI B18.6.2. The major difference between machine screws and equivalent cap screws is in dimensional tolerances. Since neither machine screws nor equivalent cap screws are intended for pressure boundary applications, the machine screws are recommended for general applications and stock.

Machine screws are normally more available than equivalent cap screws. The facility should check with the vendor or manufacturer on product availability, since many of the sizes listed in the ANSI Standard may not be readily available.

The intent of this summary specification is to assist the facility in determining the significant specification requirements pertinent to procurement, receipt inspection and installation. Refer to the Specification for details on plating and coating requirements.

This summary specification includes the following information:

- Unplated carbon steel and austenitic stainless steel machine screws and machine screw nuts
- Round head, pan head, hex head, flat head, oval head, and fillister head machine screws as well as hex and square machine screw nuts
- Sizes from #0000 through 3/4 inch nominal diameter where dimensions are published by ANSI
- Requirements contained in the 1972 ANSI specification which are still in effect.

See the Specification for other product types and materials.

15.8.2 Ordering Information

The facility purchase order should include the following information:

- Specification. The appropriate specification (e.g., ANSI B18.6.3).

- Quantity. The number of pieces required.

- Product Type. The facility should specify the screw head style and recess type, given in Table 15-20.

Table 15-20	
Screw Head Styles and Recess Types	
Screw Head Styles	Recess Types
Round	Slotted
Pan	Cross Recess
Hex	
Flat (82°)	
Oval	
Fillister	

Hex machine screws are only available with unslotted and slotted heads.
The facility should specify the nut type as square or hex.
Refer to the Specifications for information on product types, recesses, and sizes.

- Material. The material should be carbon steel or ANSI 18/8 stainless steel.

  Carbon steel should be provided unless the material is specified in the purchase order. Other materials may be available on a special order basis only.

- Dimensions. Specify nominal diameter, threads per inch (i.e., UNC or UNF), thread fit (e.g., Class 2A for screws or 2B for nuts), and length (for screws). The manufacturer may choose to provide UNRC in lieu of UNC or UNRF in lieu of UNF.

  The length of machine screws with a flat bearing surface is measured from the plane of the bearing surface to the extreme end point. The length of countersunk machine screws is measured from the top of the head to the extreme end point. Care should be taken to properly specify the required length.

  Thread Fit: Sizes #0000, #000 and #00 need not have Class 2A thread fit. See the Specification for details.

- Specification Year. The latest specification year will be furnished unless otherwise specified.

- Recommended Requirement. Invoke the Recommended Technical Requirements in this section and the Technical and Quality Requirements in Section 15.3.
- Certification. Invoke the certification requirements of this section.
- Supplementary Requirements. Special plating, coating, material, and/or dimensions should be noted.

15.8.3 Pertinent Specification Requirements

The following are summaries of the requirements contained in the latest ANSI B18.5.3 (1972) specification revision. This summary does not include most mill processing requirements and information considered inconsequential to the facility.

- Heat Treatment. Machine screws and nuts under this specification do not require heat treatment unless special materials are ordered.
- Chemical Composition. There are no specific chemical compositions required.
- Mechanical Properties. Carbon steel machine screws should be fabricated from a carbon steel having a minimum tensile strength of 60,000 psi. There are no other required mechanical properties including hardness.

  There are no specific mechanical properties required for austenitic stainless steel machine screws and nuts or for carbon steel nuts.
- Dimensions.
  - Availability. Not all of the sizes listed in the ANSI Specification are production items. The facility should consult the manufacturer or vendor to determine availability of specific sizes.
  - Bearing surface. The flat bearing surfaces of the appropriate machine screws should be perpendicular to the axis of the screw shank within 2°.
- Eccentricity. Machine screw heads, recesses, and slots should generally be on center with the axis of the screw. If eccentricity is suspected from visual examination, refer to the Specification for eccentricity tolerances and measurement methods.
- Under Head Fillets. Machine screws should have a fillet under the head.

  See the Specification for details.
- Length Tolerance. Table 15-21 gives the Nominal Screw Length Tolerance.

Table 15-21			
Nominal Screw Length Tolerance (inch)			
Nominal Screw Size	000 to 00	0 to 12	1/4 inch to 3/4 inch
Up to 1/2 inch, inclusive	-0.01	-0.02	-0.03
Over 1/2 inch to 1 inch, inclusive	-0.02	-0.03	-0.03
Over 1 inch to 2 inches, inclusive	-	-0.06	-0.06
Over 2 inches	-	-0.09	-0.09

- Thread Lengths. The usable thread length for machine screws depends on the diameter and length. The amount of thread required is shown in Table 15-22.

Table 15-22				
Amount of Usable Thread Length				
Product Diameter	Screw Length	Screw Length	Screw Length	Screw Length
	≤ 1 1/8 inches	1 1/8 inches to 2 inches	> 1 1/8 inches	> 2 inches
≤ #5	Full Thread	N/A	1 inch minimum	N/A
> #5	N/A	Full Thread	N/A	1 1/2 inches minimum

- Point or Chamfer. Machine screws should have a plain, sheared end unless otherwise specified.

- Body Diameter. Machine screws should have a minimum body diameter which equals the minimum thread pitch diameter for a Class 2A thread. The maximum body diameter should not exceed the major diameter of a Class 2A thread.

- Marking. The specification does not require product marking.

- Certification. The Specification does not require certification.

- Other. The Specification contains numerous references to dimensions and workmanship which vary by the product type required.

  The Specification does not discuss inspection provisions or rejections.

15.8.4 Recommended Technical Requirements

The adoption of these requirements will enhance the quality, traceability, and reliability of materials intended for nuclear applications. As the Specification was not written exclusively for nuclear applications, the recommendations in this section are not specification requirements. For items marked with "*", refer to Section 15.3.

15.8.4.1 Chemical Analysis

The vendor should provide a heat analysis or product analysis, performed by an approved vendor, for each heat lot or shipping lot. The analysis should report all required elements and residuals.

Table 15-23 gives the number of tests which should be performed in check analyses.

Table 15-23

Number of Samples Required for Check Analyses

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

- Carbon Steel. Carbon steel has no specific composition required, as long as the material is a generally recognized type of carbon steel. The vendor should identify the material grade on the certification which shows the chemical content.

- ANSI 18/8 Stainless Steel. ANSI 18/8 stainless steel should have a chemical composition as provided below. On occasion, elements may vary beyond the required limits. If variations are found, the material should not be used unless accepted by the facility. Materials with a sulfur content above 0.04% should not be welded by the facility.

15.8.4.2 Heat Analysis

The heat analysis, performed by the producing mill, should conform to the heat analysis given in Table 15-24.

Table 15-24	
Criteria for Heat Analysis	
Element	Amount
Chrome	16.00 to 20.00
Nickel	6.00-14.00
Carbon	0.15 maximum
Manganese	2.00 maximum
Silicon	1.00 maximum
Phosphorus	0.045 maximum

15.8.4.3 Product Analysis

Product analyses of the final product may be made by the facility or vendor and should conform to Table 15-25.

Table 15-25	
Criteria for Product Analysis	
Element	Amount
Chrome	15.80 to 20.20
Nickel	5.90-14.10
Carbon	0.155 maximum
Manganese	2.04 maximum
Silicon	1.05 maximum
Phosphorus	0.055 maximum

15.8.4.4 Mechanical Properties

- Final Hardness. All completed carbon steel machine screws and machine screw nuts should meet the hardness requirements of the ANSI B18.6.3 specification.

  Austenitic stainless steel machine screws and machine screw nuts are exempt from all mechanical testing.
- Full Size Testing of Machine Screws and Nuts. All carbon steel machine screws having a diameter larger than #0 and a length of 4 diameters should be capable of meeting the tensile requirements of the ANSI B18.6.3 specification.

All carbon steel machine screw nuts having a diameter larger than #0 should be able to meet the proof load requirements when tested with a threaded mandrel.

Austenitic stainless steel machine screws and machine screw nuts are exempt from full size testing.

- Number of mechanical tests. The number of mechanical tests for each lot is given in Table 15-26.

Table 15-26

Number of Samples Required for Mechanical Tests

Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

- Test methods. Testing should be conducted per ASTM F606.
- Conflicting Results. Full size test results should take precedence over low hardness readings.
- Traceability. Traceability should be maintained by heat lot.
- Marking. All materials should be bagged and tagged for shipment with a label showing the size, description, heat lot or shipping lot, and the customer purchase order number.
- Workmanship. All materials should have a workmanlike finish. For additional requirements refer to Section 15.3.
- Inspection. The purchaser should have full access to the manufacturer facility to inspect the work.

  Dimensional Inspection should be per MIL-STD-105, Level II (See Section 15.3.10.).

  Visual inspection should be 100%.
- Certification. The vendor certification should include the following:
  — Material description — Product type, material, and size
  — Specification and revision per ANSI B18.6.3 (1972)
  — Quantity certified

- Heat number or shipping lot number — indicate which
- Chemical analysis — Results of all tests and the method of analysis. The grade designation should also be reported for carbon steel products.
- Mechanical test results for each test performed
  - Full size axial tensile test for applicable sizes of screws
  - Threaded mandrel proof load for nuts
- Visual inspection report
- Statement of compliance to the approved Quality Program and the Technical and Quality Requirements.

15.8.5 Summary of Changes to Materials Specifications

15.8.5.1 Summary of Changes for ANSI B18.6.3

ANSI B18.6.3 did not change between 1972 and 1986.

15.9 Summary Specification for ANSI/ASME B18.3 and ANSI B18.6.2 Carbon, Alloy, and Stainless Steel Headless Set Screws

15.9.1 Scope

This summary specification is a condensed version of specification requirements and dimensional information from ANSI B18.6.3 and ANSI B18.6.2 which cover various types of set screws for general purpose applications.

Alloy hexagon socket set screws are manufactured to a Class 3A thread fit from a high harden alloy steel or 18/8 stainless steel. The hexagon socket distributes the wrenching torque over six flats of the socket. They are recommended for all critical applications and stock.

The intent of this summary specification is to assist the facility in determining the significant specification requirements pertinent to procurement and receipt inspection. Refer to the Specifications for details on plating and coating requirements.

This summary specification includes the following information:

- Unplated carbon steel, alloy steel, and austenitic stainless steel headless set screws

- Sizes from #0 through 1 1/2 inches nominal diameter where dimensions are published by ANSI

- Requirements contained in the 1982 Edition of the ANSI/ASME B18.3 and the 1972 Edition of ANSI B18.6.2.

  See the Specifications for other product types and materials.

15.9.2 Ordering Information

The facility purchase order should include the following information:

- Specification. The facility should specify the applicable ANSI specification as follows:
  - ANSI/ASME B18.3: Hexagon socket; set screws
  - ANSI B18.6.2: Slotted; set screws.

- Quantity. Number of pieces required

- Product Type. The facility should specify the recess type and point from the following:
  - Recess Type: Hexagon socket; slotted
  - Point: Flat, cup, oval, cone, half dog, dog (slotted only)

- Material
  - Carbon steel (slotted only)
  - Alloy steel (hexagon socket only)
  - AISI 18/8 or AISI Type 384 austenitic stainless steel

  Other materials may be available on a special order basis only.

- Dimensions. Nominal diameter, threads per inch (i.e., UNC or UNF), thread fit (e.g., Class 2A for slotted or 3A for hexagon socket), and length. The manufacturer may choose to provide UNRC or UNRF in lieu of UNC or UNF, respectively.

  The length of set screws is the overall distance measured parallel to the axis of the screw, from the top of the screw to the end of the point.

- Specification Year. The latest specification year for hexagon sockets and slotted set screws will be furnished unless otherwise specified.

- Recommended Requirements. Invoke the Recommended Technical Requirements in this section and the Technical and Quality Requirements in Section 15.3.

- Hardness. The Specification requires hardness testing. The facility may also invoke the supplementary hardness requirements from Section 15.3.

- Certification. Invoke the certification requirements of this section.

- Supplementary Requirements. Special plating, coating, material, and/or dimensions should be noted.

15.9.3 Pertinent Specification Requirements

The following are summaries of the requirements contained in the latest ANSI B18.6.3 (1982) and ANSI B18.6.2 (1972) Specification revisions. This summary does not include most mill processing requirements and information considered inconsequential to the facility.

- Heat Treatment. Carbon and alloy steel set screws require heat treatment. The Specifications do not provide required times or temperatures. alloy set screws should be heat treated by austenitizing, followed by a liquid quench an subsequent tempering. Carbon steel set screws should be heat treated per the manufacturer normal procedures.

  Austenitic stainless steel set screws do not require heat treatment.

- Chemical composition. Alloy set screws should contain either chrome, nickel, molybdenum, or vanadium in sufficient quantities to ensure that the required hardness is met. There are no specific percentages established for the alloying elements required.

  Carbon steel set screws have no required chemical content. Austenitic stainless steel set screws may be manufactured from ANSI 18/8 stainless steel or AISI 384 stainless steel. There are no published requirements for 18/8. Table 15-27 gives the heat analysis for AISI 384 stainless steel.

Table 15-27

Criteria for Heat Analysis

Element	Amount
Carbon	0.08 maximum
Manganese	1.00 max.
Silicon	1.00 maximum
Phosphorus	0.045 maximum
Sulfur	0.030 maximum
Chrome	15.00-17.00
Nickel	17.00-19.00

- Mechanical properties. There are no specific mechanical properties required for carbon steel or austenitic stainless steel set screws.

  Alloy set screws should have a final Rockwell "C" hardness of 45 to 53.

- Dimensions. Not all of the sizes listed in the ANSI Specifications are production items. The facility should consult the manufacturer or vendor to determine availability of specific sizes.

Set screws recesses and points should generally be on center with the axis of the screw. If eccentricity is suspected from visual examination, refer to the Specifications for eccentricity tolerances and measurement methods.

The tolerance on length is given in Table 15-28.

Table 15-28

Tolerance on Length

Nominal Screw Length	Slotted	Hex Socket
≤ 5/8 inches	-0.03	±0.01
Over 5/8 inch to 1 inch	-0.03	±0.02
Over 1 inch to 2 inches	-0.06	±0.02
Over 2 inches to 5 inches	-0.09	±0.03
> 6 inches	-0.09	±0.06

- Marking. The Specifications do not require product marking.
- Certification. The Specifications do not require certification.
- Other. Slotted headless set screws should be free from burrs, seams, laps, loose scale and any other defects affecting their serviceability.

  The ANSI B18.3 Specification for hexagon socket set screws contain numerous indirect references to workmanship.

  The Specifications do not discuss inspection provisions or rejection.

15.9.4 Recommended Technical Requirements

The adoption of these requirements will enhance the quality, traceability, and reliability of materials intended for nuclear applications. As the Specification was not written exclusively for nuclear applications, the recommendations in this section are not specification requirements. For items marked with *, refer to Section 15.3.

15.9.4.1 Chemical Analysis

The vendor should provide a heat analysis or product analysis, performed by an approved vendor, for each heat lot or shipping lot. The analysis should report all required elements and residuals.

Table 15-29 gives the number of tests required in product analyses.

Table 15-29	
Number of Samples Required for Product Analyses	
Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

- Carbon Steel. Has no specific composition required, as long as the material is a generally recognized type of carbon steel. The vendor should identify the material grade on the certification which shows the chemical content.
- Alloy steel has no specific composition required. The vendor should provide the AISI alloy number or its equivalent, and the allowed range of alloying elements on the certification which reports the chemical content. Product analysis tolerances should be determined from ASTM A484.
- AISI 18/8 Stainless Steel should have a chemical composition as provided below. On occasion, elements may vary beyond the required limits. If variations are found, the material should not be used unless accepted by the facility. Materials with a sulfur content above .04 should not be welded by the facility.
- The heat analysis for 18/8, as performed by the producing mill should conform to the criteria in Table 15-30

Table 15-30	
Criteria for Heat Analysis	
Element	Amount
Carbon	0.15 maximum
Manganese	2.00 max.
Silicon	1.00 maximum
Phosphorus	0.0456 maximum
Chrome	16.00-20.00
Nickel	6.00-14.00

- Product analyses of the final product of 18/8 material may be made by the facility or vendor and should conform to the criteria in Table 15-31.

Table 15-31	
Criteria for Product Analysis for 18/8 Material	
Element	Amount
Carbon	0.155 maximum
Manganese	2.04 max.
Silicon	1.05 maximum
Phosphorus	0.055 maximum
Chrome	15.80-20.20
Nickel	5.90-14.10

Product analyses of the final product of AISI 384 material may be made by the facility or vendor and should conform to the criteria in Table 15-32.

Table 15-32	
Criteria for Product Analysis for AISI 384 Material	
Element	Amount
Carbon	0.09 maximum
Manganese	1.04 max.
Silicon	1.05 maximum
Phosphorus	0.055 maximum
Sulfur	0.035 maximum
Chrome	14.85-17.20
Nickel	16.85-19.15

15.9.4.2 Traceability

Traceability should be maintained by heat lot.

15.9.4.3 Mechanical Properties

Carbon steel set screws should have a final hardness as agreed on between the manufacturer and the facility.

Austenitic stainless steel set screws are exempt from all mechanical testing.

The number of hardness tests for each lot of carbon steel and alloy steel set screws is given in Table 15-33.

Table 15-33	
Number of Samples Required for Hardness Tests	
Lot Size	Sample Size
1-800	1
801-8000	2
8001-22,000	3
Over 22,000	5

Testing should be conducted per ASTM F606.

15.9.4.4 Marking

All materials should be bagged and tagged for shipment with a label showing the size, description, heat lot or shipping lot, and the customer purchase order number.

15.9.4.5 Workmanship

All materials should have a workmanlike finish. For additional requirements refer to Section 15.3.

15.9.4.6 Inspection

The purchaser should have full access to the manufacture facility to inspect the work.

Dimensional Inspection should be per MIL-STD-105, Level II (see Section 15.3.10).

Visual inspection should be 100%.

15.9.4.7 Rejection

The results of tests, inspections, or reviews performed by the facility may constitute a basis for rejection.

15.9.4.8 Certification

The vendor certification should include the following:

- Material description — Product type, material, and size
- Specification and revision ANSI B18.3 (1982) or ANSI B18.6.2 (1972)

- Quantity certified
- Heat number or shipping lot number — Indicate which
- Chemical analysis — Results of all tests and the method of analysis. The grade designation should also be reported for carbon steel and alloy steel set screws. The allowed ranges of alloying elements should also be reported for alloy set screws.
- Hardness results for each test performed (carbon and alloy set screws only)
- Visual inspection report
- Statement of compliance to the approved Quality Program.

15.9.5 Summary of Changes to Materials Specifications

15.9.5.1 Summary of Changes for ANSI B18.3 and ANSI B18.6.2

ANSI B18.3 was revised in 1972 and 1976. As the Specification covers many additional products other than set screws, the revisions did not affect set screws. ANSI B18.6.3 did not change between 1972 and 1986.

16.0 Receipt Inspection and General Precautions

16.1 Introduction

This chapter discusses guidelines on receipt inspection including statistical sample size and various inspection attributes such as visual and dimensional inspection and hardness testing. In addition, general considerations or precautions are highlighted for hardness tests, documentation, segregation and storage, replacement material and material received with equipment, manufacture of bolting material from bar stock, disposition of nonconforming material, and reuse of material.

16.2 Sample Size for Inspection

Lot size is defined as the number of items of the same size, nominal diameter, material type, property class, and material heat number received under a single purchase order. In addition, the items are identified by the manufacturer as being from one production lot and from one heat treated batch.

A random sample is a sample of items in a lot which are representative of the quality of the entire lot. Each item is selected at random from different locations in the lot. The sample size is the recommended number of items inspected from each lot, as given in Table 16-1. Other sampling methods are acceptable provided they adequately represent the lot and are statistically significant.

Table 16-1

Recommended Sample Size and Maximum Number of Failures for Various Size Lots

Lot Size	Sample Size	Maximum Acceptable Failures
1 to 50	8	0
51 to 90	13	0
91 to 150	20	0
151 to 280	32	1
281 to 500	50	1
501 to 1,200	80	2
1,201 to 3,200	125	3
3,201 to 10,000	200	5
10,001 to 35,000	315	7
35,001 and over	500	10

16.3 Visual and Dimensional Inspection

The visual inspection performed on receipt or reinstallation of fasteners includes:

- Cleanliness. Fastener samples should be free of moisture, foreign materials, and oxides or rust. Clean the fastener to permit proper inspection and installation, if necessary.
- Accurate dimensions. Inspect fastener samples for conformance to applicable ANSI specifications as well as to the purchase order.
- Physical damage and discontinuities. Examine fastener samples for physical damage and discontinuities. For bolts, this examination includes the threads, shaft, and head. For nuts, the threads, edges, and flats should be examined. Examples of physical damage include nicks, gouges, dents, and scratches. Examples of discontinuities include laps, seams, cracks, bursts, folds, voids, and marks.

  If any sample lot with unacceptable physical damage, inspection of the entire lot may be necessary to separate the damaged fasteners. Another option is to return the entire lot to the manufacturer in accordance with the purchase contract.

  All inspections should be conducted by qualified personnel who have an understanding of possible discontinuities. The inspector should accept or reject the fastener based on ASTM or ASME standard specifications for surface discontinuities on bolts and nuts. Each facility is responsible for developing more stringent rejection criteria if necessary, as well as for training personnel performing the detailed evaluation of the discontinuity. This activity may fall under the provisions of NX-2580 and SNT-TC-1A-1975. If the discontinuity is unacceptable, the entire lot should be returned to the manufacturer.

16.4 Hardness Test

The hardness test performed on receipt or reinstallation of fasteners should be performed by qualified personnel or by an experienced laboratory. The test method (e.g., Brinell, Rockwell, etc.) should be in accordance with ASTM standards.

A lower test load, a smaller indentor, or a different hardness test can be used. Hardness conversion tables may be used to compare actual hardness values to those required in the material specification.

For bolts, the hardness test is most often performed on the side or top of the bolt head. However, the test may be performed on the threaded end of the bolt. For nuts, the hardness test is usually performed on the top or bottom face of the nut. The side of the nut is also acceptable.

In order to accept the lot, each fastener in a sample must have a hardness within the range given in the material specification. If any hardness reading is not within the hardness range, additional hardness tests on the cross section of the hardest and softest fasteners are required. Do this only if tension testing is not required.

For bolts, additional tests are performed on transverse sections through the threaded portion of the bolt at a point one fourth the nominal diameter from the bolt axis. The tested section is one bolt diameter from the end of the bolt.

For nuts, additional hardness tests are performed on transverse sections parallel to the face of the nut through the threads at the mid-point between the inside and outside diameters. The tested section is at least two threads from the flat face of the nut.

The lot may be rejected for failure to comply with the hardness requirements given in the material specification or in the purchase order. Sometimes hardness test results are suspect. In these cases, accept the fastener if the tension test result is acceptable.

16.5 Documentation

Maintain documentation of the inspection and test results, including physical damage, discontinuities, and hardness test results. This documentation supports the activities performed at receipt or reinstallation.

16.6 Segregation and Storage Requirements

16.6.1 One Inch Nominal Diameter and Smaller Bolts

All bolting of one inch and less nominal diameter, regardless of the classification, may be stored together provided they are segregated by certification requirement, material specification, grade, type, size, etc.

16.6.2 Bolts Larger than One Inch Nominal Diameter

Bolts greater than one inch nominal diameter should be segregated by certification requirement, material specification, grade, type, size, heat treated condition, etc., by the following classifications:

- All ASME Section III, Class 1 bolting material
- All ASME Section III, Class 2, 3, or MC bolting material
- All ASME Section III, Subsection NF component support bolting material for Class 1 components
- All ASME Section III, Subsection NF component support bolting material for Class 2, 3, or MC components requiring impact tests
- All remaining ASME Section III, Subsection NF component support bolting material for Class 2, 3, or MC components

- All non-ASME Section III bolting material should be segregated when additional requirements, such as nondestructive examination or impact testing are specified.

16.6.3 Cleanliness and Storage

Before storing bolting material, the storage area must ensure the cleanliness of the bolting material and prevent corrosion. Austenitic stainless steel bolting material should be cleaned in accordance with the facility standard.

16.7 Replacement Bolting Material

All requirements for replacement bolting material should be equal to or more stringent than the those for the original material, unless the original requirements have been modified by approved design changes. Replacement bolting material includes material purchased to replace material no longer usable, joints assembled by the supplier, and structural steel assemblies.

For operating plants and those who are under the jurisdiction of ASME Section XI, the following options are available for the procurement contract:

- Fasteners may be specified by manufacturer or supplier part number which applies to the original procurement contract.

- If the fasteners are standard, procure using a standard description, the appropriate material specification, and the latest Nuclear Regulatory Commission (NRC) approved ASME Code requirements.

- Procure fasteners as described above, except that an earlier edition of the ASME Code is used. The edition of the Code must **not** be earlier than the Code applied to the original component. If the fastener was procured under non-ASME Code requirements, a later edition of the Code can be used, provided all of the original requirements are met.

16.8 Bolting Material Received with Equipment

Loose bolting material, received as part of vendor-supplied equipment or prefabricated structural steel assemblies, should be identified to ensure matching to the correct equipment. If certification, identification, and nondestructive examination requirements are met, the bolting material should be stored in accordance with Section 16.2.

16.9 Manufacture of Bolting Material from Code-Approved Bar Stock

The following sections give guidelines for manufacturing bolting material from Code-approved bar stock.

16.9.1 Material

Use material that meets the requirements of applicable ASME SA or ASTM A bolting material specifications. If bolting is needed in ASME Section III applications, refer to this Section for upgrade and classification requirements prior to using the bolting material.

16.9.2 Machining

Machine bolting material as directed on approved owner or vendor drawings.

16.9.3 Examination

Examine bolting material as required by the applicable material specification or code requirements. As a minimum, examine:

- Unthreaded rods
- Bars
- Pins that function as component support parts and are held at each end with a locking device, such as cotter pins
- Pins that function by deforming the ends to provide an alternate locking device.

16.9.4 Testing

Perform chemical and mechanical testing of bolting material as required by applicable bolting material specifications or standards. Test reports should include the number of test samples and the lot size for each group of test samples.

A qualified laboratory should perform necessary tests to supplement the documentation requirements received with the material. The results of all tests should be documented and maintained in the files at the construction site.

16.9.5 Marking

Markings should be in accordance with Chapter 15, Section 15.3.4 of this manual.

16.10 Recommended Disposition of Nonconforming Material

16.10.1 Improperly Marked Material

Bolting material that is not properly marked should be rejected or placed in "nonconforming" or "hold" status. Dispose of the material according to approved procedures.

16.10.2 Inadequately Certified Material

Bolting material received without proper certification may be used for ASME applications, provided the following requirements are met:

- Additional tests necessary to meet the material specification and Code requirements are performed by the facility or an approved laboratory.
- Results of all tests, examinations, and treatments are documented. The material should be placed on "hold" status pending the acceptance of test results.

Material which does not conform to acceptance criteria should be disposed of according approved procedures.

The use of inadequately certified material may be permitted for non-ASME use if the material is properly identified and the application does not require traceability.

16.11 Precautions

16.11.1 Bolting and Nut Materials

Evaluate the potential for stress corrosion cracking for bolting materials with a minimum specified ultimate tensile strength greater than 150 ksi or a maximum actual ultimate tensile strength greater than 170 ksi. The designer should evaluate the adequacy of the material. Other considerations include:

- SA or A564, Type 630 material may be used if it is aged and heat treated to a minimum of 1100°F.
- SA or A563, Grade C nuts may be used if heat treated and have a minimum hardness of 248 HB. If these nuts are not available, substitute SA or A194, Grade 2H nuts or SA or A563, Grade DH or DH3 nuts with a hardness range of 248 HB to 352 HB.
- Welding, including tack welding, on bolts and nuts requires a qualified procedure meeting the requirements of ASME Section IX and the applicable portion of the governing ASME or ANSI Code. No welding should be performed on the loaded portion of bolts.
- Welding of high strength low alloy bolting material (i.e., A193, Grade B7, etc.) should not be done.

16.11.2 Reuse of Bolting Material

Bolting material which is damaged, deformed, or otherwise affected during installation or service should not be reused unless the reuse is supported by an engineering evaluation and analysis. The designer of the bolting material should determine if reused material can perform in accordance with applicable design requirements and specifications.

Retightening of previously tightened bolts loosened during the tightening of adjacent bolts is not considered reuse. Bolting material that should not be reused under any circumstances include:

- Galvanized bolts and nuts
- A490 bolts
- Any bolt or nut tightened by the turn of nut method.

17.0 REFERENCES

17.0 REFERENCES

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Appendix A

Glossary

Actual Fit is the relation existing between two mating parts with respect to the amount of clearance or interference which is present when they are assembled.

Allowance is an intentional difference between the maximum material limits of mating parts. It is the minimum clearance (positive allowance) or maximum interference (negative allowance) between such parts.

Alloy Steel is a steel to which other elements in addition to carbon have been added to obtain improved mechanical or physical properties, such as higher strength at elevated temperatures, toughness, etc.

Angularity is the angle between the faces of the nut and the bolt head and the thread or shank axes should be perpendicular. If the angle between the face and the axis is, for example, 87° or 93°, the fastener is said to have an angularity of 3°.

Annealing is heat treatment of a fastener to make it soft, that is, free of hardness caused by cold working or previous heat treatment and quenching.

Anodizing is the formation of an oxide film on the surface by means of an anodic treatment. This is commonly used on aluminum.

Barrier Protection is a term relating to coating of a fastener and applies if the coating (paint) isolates the fastener from the environment.

Basic Size is that size derived by the application of allowances and tolerances.

Bearing Surface is the supporting surface of a fastener with respect to the part which it fastens.

Body of a threaded fastener is the unthreaded portion of the shank.

Bolt Gage is an ultrasonic instrument used to measure the stress or strain in bolts or other fasteners by measuring change in elongation due to preload and stretch of the bolt.

Bolt is a threaded fastener designed to be used with a nut.

Brittle behavior in a bolt is characterized by breaking when stretched only a small amount past its yield point.

Cap Screw is a threaded fastener made to closely controlled dimensions having a head of wrench, slotted, recessed or socket type.

Carbon Steel is a steel containing less than 0.8% carbon and trace amounts of sulfur, phosphorous, silicon, or manganese.

Case Hardened fastener is one of ferrous metal having undergone a surface heat treatment which has resulted in the surface being harder than the core.

Chamfer Angle is measured from the normal to the axis of the fastener and is generally specified in conjunction with either a length or a diameter.

Clamping Forces are those equal and opposite forces which exist at the interface between two joint members created by tightening the bolts, but is not always equal to the combined tension in the bolts. Such problems as hole interference, for example, can create a difference between clamping force and bolt load.

Class of Threads specifies the degree of tolerance and allowance.

Commercial Fastener one that is manufactured to published standards and stocked by manufacturers or distributors. The material, dimensions and finish conforms to a quality level generally recognized by manufacturers and users as commercial quality.

Constant Life Diagram is a plot of experimentally derived fatigue-life data; and is perhaps the most complex and complete of those diagrams used to represent such data.

Corrosion Cell is a natural "battery" formed when two metals having different electrical potentials (an Anode and a Cathode) are connected together in the presence of a liquid (the Electrolyte).

Countersink is an internal chamfer.

Creep is the slow plastic deformation of a body under heavy loads that occurs at high temperature.

Cross Drilled fasteners have one or more holes in the head or shank at right angles to, and normally intersecting, the axis of the fastener.

Cut Threads are produced by removing material from the surface with a form cutting tool.

Decarburization is a diffusion controlled metallurgical phenomena resulting in the carbon content of the surface being lower than the carbon content of the core.

Design Size is that from which the limits of size are derived by application of tolerances. When there is no allowance, the design size is the same as the basic size.

Direct Tension Indicator is used primarily in the structural steel industry, to indicate that a certain minimum amount of tension has been developed in the fastener during assembly.

Ductile bolt is one that can be stretched well past its yield point before breaking.

Eccentric Load on a fastener or groups of fasteners occurs when a resultant of that load does not pass through the center of the group of fasteners (eccentric shear load) or does not coincide with the bolt axis (eccentric tensile load).

Effective Length of a bolt is the grip length plus some portion of the bolt (often one-half of the thickness of the nuts) which lies within the nut(s) plus some portion (often one-half the thickness) of the head.

Effective Radius of nut or bolt head or threads is the distance between the geometric center of the part and the circle of points through which the resultant contact forces between mating parts pass.

Effective Thread includes the complete thread and that portion of the incomplete thread having fully formed roots, but having crests not fully formed.

Elastic Interactions occur when a bolt, as it is tightened, partially compresses joint members closest to it. Subsequent tightening of adjacent bolts further compresses the joint in this region. This allows the first bolt to relax slightly. Bolts tightened on the opposite side of the joint, might increase preload in some of the initial bolts tightened on the near side. Shifts and changes in the elastic energy stored in individual bolts, during assembly, are called elastic interactions.

Electro-Galvanizing is the process of coating metal with zinc by electroplating.

Electrodes are the two metallic bodies in a battery or corrosion cell which give up electrons (the Anode) or which attract them (the Cathode).

Electrolyte is the liquid medium with which two electrodes of a battery or corrosion cell are wetted.

Embedment is localized plastic deformation in heavily loaded fasteners which allows one part to sink into, such as by crushing asperities, a softer or more heavily loaded second part, etc. Bolt threads embed themselves in nut threads, etc. Nuts embed themselves in joint surfaces.

Endurance Limit is that completely reversing stress limit below which a bolt or joint member will have an essentially infinite life under cyclic fatigue loads. The mean stress on the bolts here is zero.

Equation, Long Form Torque, is a theoretical equation based on rigid body mechanics that relates the torque applied to a bolt to the preload created in it. Parameters include fastener geometry and the coefficient of friction between mating surfaces.

Equation, Short Form Torque, is an empirical equation relating torque applied to the bolt to the preload created in it. It depends mainly on an experimentally derived Nut Factor.

Extensometer is any instrument that measures the change in length of a part as the part is loaded (see Bolt Gage).

External Load Forces are those exerted on fastener and/or joint members by such external factors as weight, wind, inertia, vibration, temperature expansion, pressure, etc.

Extruding is the process of reducing the size of some feature or diameter by forcing it through a die.

Failure of the Bolt implies that the bolt has broken or the threads have stripped.

Failure of the Joint implies the joint fails to behave as intended by the designer. Failure can be caused or accompanied by broken or lost bolts, but can also mean joint slip or leakage from a gasketed joint even if all bolts still remain whole and in place. Common reasons for joint failure include vibration loosening, poor assembly practices, improper design, unexpected service loads or conditions, etc.

Fastener is a mechanical device for holding two or more bodies in fixed positions with respect to each other.

Fillet is the transition region between bolt head and shank, or between other changes in diameter.

Finish is commonly applied to the condition of the surface of a fastener as a result of chemical or organic treatment subsequent to fabrication.

Fit is the general term used to signify the range of tightness which may result from the application of a specific combination of allowances and tolerances in the design of mating parts. Three general types of fit include—clearance, transition and interference.

Flange Rotation is angular distortion of a flange under the influence of bolt and reaction forces.

Flank is the surface connecting the crest with the root. Theoretically, the intersection of the flank with an axial plane is a straight line.

Flash Plating is a very thin deposit of metal on a fastener usually on the order of 0.00005 to 0.00015 in. in thickness.

Forging is the process of forming a product by hammering or pressing.

Form is the profile of the thread in an axial plane for a length of one pitch.

Full or Nominal Diameter Body is that diameter that is generally within the dimensional limits of the major diameter of the thread.

Galling is an extreme form of adhesive wear, in which large chunks of one part stick to the mating part (during sliding contact).

Galvanic Protective coating on a fastener is more anodic than the fastener and will, therefore, be destroyed first in-service instead of the fastener.

Galvanizing is the process of coating metal with zinc generally by hot dipping

Gasket Factors are experimentally derived "constants" used to define the behavior of a gasket and/or the assembly and in-service conditions required for acceptable behavior. These factors are not design recommendations, but, instead, define the behavior of the gasket.

Gasket Stress is that stress exerted on the gasket by the joint members.

Grinding is the process of removing material from the surface by the cutting action of a bonded abrasive wheel.

Grip Length is the combined thickness of all members in the joint clamped together by the bolt and nut, including washers, gaskets.

Grip is the thickness of material or parts which the fastener is designed to secure when assembled.

Ground Thread is a thread finished on the flanks by a grinding operation.

Head Diameter is the diameter at the largest periphery of the head.

Head Height is the overall distance, for a flat bearing surface head, measured parallel to the fastener axis, from the extreme top to the bearing surface.

Head Length is the distance along the longest axis of the head, measured in a plane perpendicular to the axis of the fastener.

Head Width is the distance across opposite flats of hexagon, square or twelve-point heads measured in a plane perpendicular to the fastener axis. For rectangular or irregular shaped heads, the head width is the distance along the narrowest axis of the head measured in a like manner.

Head of a fastener is the enlarged shape on one end of a headed fastener to provide a bearing surface.

Header is a specialized form of horizontal press in which the head is formed.

Heading is a manufacturing process involving the use of a header. The process may or may not involve upsetting or extruding. A part made from wire below the recrystallization temperature is said to be cold headed, whereas parts made from wire above the recrystallization temperature are said to be hot headed.

Hexagon Head has a flat or indented top surface, six flat sides and a flat bearing surface.

Hexagon Washer Head is a washer head upon which a hexagon head is formed.

High Strength Fastener has high tensile and shear strengths attained through combinations of materials, work-hardening and heat treatment.

Hydrogen Embrittlement may occur by entrapment of hydrogen in a bolt by poor electroplating practices. It can promote stress cracking, bolts can fail, suddenly and unexpectedly, under normal operating loads.

Impact Wrench is an air- or electric-powered wrench in which multiple blows from tiny hammers are used to produce output torque to tighten or loosen fasteners .

Inclusions are particles of nonmetallic impurities contained in material.

Incomplete Thread, also known as the vanish or washout thread. On straight threads, it is that portion at the end having roots not fully formed by the lead or chamfer on threading tools.

Infinite Life Diagram is a simple plot of experimentally derived fatigue life data, showing conditions required for infinite life.

Initial Preload is the tension created in a single bolt as it is tightened. It can subsequently be modified by additional assembly operations and/or by in-service loads and conditions.

Internally Relieved Body is a body which has an axial hole drilled through a portion of the body.

Joint Diagrams are mathematical plots which illustrate the forces on and deflections of fasteners and joint members.

Laps are surface defects caused by folding over of fins or sharp corners into the surface of the material.

Lead is the distance a threaded fastener travels axially in one complete revolution.

Length of Thread Engagement of two mating threads is the distance between the extreme points of contact on the pitch cylinders or cones, measured parallel to the axis.

Length of a headed fastener is the distance from the intersection of the largest diameter of the head with the bearing surface to the extreme point, measured in a line parallel to the axis of the fastener.

Lock Nut is a device that provides extra resistance to vibration loosening (beyond that produced by proper preload), either by providing some form of prevailing torque, or, in free-spinning lock nuts, by deforming, cramping, and/or biting into mating parts when fully tightened .

Lockbolt bears a superficial resemblance to a bolt, but which engages a collar (instead of a nut) with annular grooves (instead of threads) . The collar is swaged over the grooves on the male fastener to develop preload .

Machining is the process of forming the surface by cutting away material.

Major Diameter is that diameter at the crest of an external thread or the root of an internal thread.

Material Velocity of sound in a body (e.g., a bolt) is a term used in the ultrasonic measurement of bolt stress or strain.

Maximum Material Condition (MMC) is the maximum amount of material permitted by the tolerance shown for a specific feature of a fastener.

Mean Value is the average value of a number of data points, computed by dividing the sum of all data by the number of data points.

Minor Diameter is that diameter of the coaxial cylinder at the boundary of the root of an external thread or the crest of an internal thread.

Nominal Diameter is roughly equal to the diameter of the body, or the outer diameter of the threads.

Nominal Size is the designation used for the purpose of general identification. Tolerances will apply to the nominal size.

Nonlinear Behavior in a fastener or joint system occurs when the relationship between the preload and/or external load on the joint and deformation of the parts is nonlinear.

Nonstandard Fastener is one that differs in size, length, material markings or finish from established and published standards.

Nut Factor is an experimental constant used to evaluate or describe the ratio between the torque applied to a fastener and the preload achieved.

Nut Thickness is the overall distance from the top of the nut to the bearing surface, measured parallel to the axis of the nut.

Nut Width is the distance across opposite flats of hexagon, square or twelve-point nuts.

Oiled is a term denoting the application of a suitable corrosion retarding oil to a fastener.

Passivating is done to improve the corrosion resistance of the surface of a fastener. The process includes for stainless steels dissolution of ferrous particles and surface impurities by chemical means (normally a nitric acid dip) and producing a passive film on the surface.

Pitch Diameter is the diameter of the coaxial cylinder which passes through the thread forms such that the width of the groove (at the cylinder) equals one-half of the pitch.

Pitch is the nominal distance between two adjacent thread roots or crests.

Preload Accuracy is a measure of the precision with which a given tool or procedure creates preload in a bolt when the bolt is tightened.

Preload is the tension created in a threaded fastener when the nut is tightened.

Prevailing Torque is that required to run a nut down against the joint when some obstruction, such as a plastic insert in the threads, or a noncircular thread, or other, has been introduced to help the fastener resist vibration loosening.

Proof Load is a specified test load which a fastener must withstand without any indication of failure.

Proof Test is any specified test required for a fastener to indicate that it is suitable for the purpose intended.

Prying is the magnification of an eccentric tensile by pseudolever action.

Raised-face Flange is one which contacts its mating joint member only in the region in which the gasket is located. The flanges do not contact each other at the bolt circle.

Recessed Head has a specially formed indentation or recess centered in its top surface.

Reference Dimension has no tolerance and is used for information purposes only.

Relaxation results in the loss of clamping force due to loss of tension, in a bolt and joint as a result of embedment, vibration loosening, gasket creep, differential thermal expansion, etc.

Residual Preload remains in an unloaded bolted joint after relaxation.

Rolled Threads are formed by plastically deforming the surface of the blank rather than by cutting operation the thread into the blank. The use of rolled threads may increase fatigue life (and possibly stress corrosion cracking) and thread strength. Size limitations and economics may preclude its use on larger sizes.

Stress Corrosion Cracking (SCC) is a common form of material degradation in which an electrolyte growth of a crack in a highly stressed and susceptible bolt material. Only a tiny quantity of electrolyte need be present, at the tip or face of the crack.

Scale is an oxide of iron sometimes formed on the surfaces of hot headed or forged fasteners.

Screw Threaded Fastener is one designed to be used in a tapped or untapped (e.g., wood screw) hole, but not with a nut.

Self-Loosening is the process by which a supposedly tightened fastener becomes loose, as a result of vibration, thermal cycles, shock, or anything else which cause transverse slip between joint members and/or male and female threads.

Shank is that portion of a bolt which lies under the head.

Shear Stress Area is the effective area at a diameter equal to the maximum minor diameter of the mating internal thread for external threads,. For internal threads it is the effective area of a diameter equal to the major diameter of the mating external thread.

Slug Wrench is a box wrench with an anvil on the end of the handle. Torque is produced by striking the anvil with a sledge hammer.

Socket Head has a flat chamfered top surface with a smooth or knurled cylindrical side surface and a flat bearing surface. A hexagon or spline (formerly known as "fluted") socket is usually formed in the center of the top surface.

Special Fastener differs in any respect from recognized standards.

Spherical Washer has an upper surface that is semispherical. It is normally used with a nut whose contact face is also semispherical. It's use reduces bending stress in a bolt or stud, by allowing some self-alignment and/or some compensation for nonparallel joint surfaces.

Spring Constant is the ratio between the force exerted on a spring (or a bolt) and the accompanying deflection. Has the dimensions of force per unit change in length (e.g., Ib/in.). It is somewhat opposite to stiffness.

Stainless Steel is a corrosion resistant alloy steel which usually contains a minimum of 12 percent chromium.

Standard Fastener is a fastener which conforms in all respects to recognized standards.

Stock Fastener is a fastener that is commercially stocked in quantity by manufacturers or distributors of fasteners.

Strain or Work Hardening is the increase in hardness, and hence strength, resulting from plastic deformation at a temperature below the recrystallization range.

Strength of Bolt can refer either to ultimate strength, proof load endurance limit, or yield strength.

Stress Area is the effective cross-sectional area of the threaded section of a fastener. It is used to compute average stress levels in that section and is based on the mean of pitch and minor diameters.

Stress Cracking involves a number of failure modes, each having a component of stress and chemical actions, including - hydrogen embrittlement, stress corrosion cracking, stress embrittlement, and hydrogen-assisted stress corrosion.

Stress Factor is a calibration constant used in ultrasonic measurement of bolt stress or strain. It is the ratio between the change in ultrasonic transit time caused by the change in length of the fastener, under load, to the total change in transit time (which is also affected by a change in the stress level).

Stress Relaxation is the slow decrease in stress level within a bolt which is heavily loaded under constant deflection conditions. It is similar to creep, which is a slow change in geometry under constant stress conditions.

Stud is a headless threaded fastener, threaded on both ends, with an unthreaded body in the middle section, or it may be threaded from end to end.

Temper is the state of a metal or alloy involving its microstructure and mechanical properties, it may vary from the annealed temper (soft) to spring temper (hard).

Temperature Factor is a calibration constant that accounts for the effects of thermal expansion and the temperature-induced change in the velocity of sound It is used in ultrasonic measurement of bolt stress or strain.

Tensile Stress Area is the assumed area of an external threaded fastener used for the purpose of computing the tensile strength of the part.

Tensioner is a hydraulic tool used to tighten a fastener by stretching it rather than by applying a substantial torque to the nut. After stretching a bolt or stud, a nut is run down against the joint with a slight torque, the tensioner is subsequently disengaged from the fastener. The nut will hold the stretch produced by the tensioner.

Thread Axis is the axis of its pitch cylinder.

Thread Form is the cross-sectional shape of the threads, defining thread angle, root, and crest profiles, etc.

Thread Length is that portion of the fastener which contains threads cut or rolled to full depth.

Thread Run-out is that portion of the threads which are not cut or rolled full depth, but which provide the transition between full-depth threads and the body or head .

Thread, Complete Length is that having full form at both crest and root. Where there is a chamfer at the start of the thread not exceeding two pitches in length, it is included within the length of the complete thread. The thread length on the drawing shall be the gaging length or the length of threads having full form, i.e., the partial threads shall be outside or beyond the length specified.

Tightness Parameter is dimensionless and defines the mass leakage of a gasket as a function of contained pressure and a contained fluid constant.

Tightness is the ability of a bolted joint to contain a fluid.

Tolerance is the total permissible variation of a parameter (e.g. length, diameter, etc.).

Torque Monitor is a system that controls torque by monitoring the tool during use, but does not control the tool or the torque produced.

Torque Multiplier is a gearbox used to increase the torque produced by a small hand wrench. The multiplier drives the fastener with a higher torque and a slower speed than, than input torque and speed.

Torque Pack is a geared wrench that multiplies input torque and provides a read-out of output torque. In effect, a combination of a Torque wrench and a multiplier.

Torque is the twisting moment, product of force and wrench length, applied to a nut or bolt.

Torque Wrench is a manual wrench incorporating a gauge or other device to measure and display the amount of torque being delivered to fastener.

Total Thread includes the complete or effective thread and the incomplete thread.

Toughness relates to the ability of a material to absorb energy without fracturing.

Transducer is a device that converts one form of energy into another (e.g. conversion of electrical to acoustic energy by an ultrasonic transducer used by the Bolt Gage).

Turn-of-nut describes general rotation of the nut (or bolt head) as the fastener is tightened. Defines a tightening procedure in which a fastener is first tightened with a preselected torque, and is then tightened further by giving the nut an additional, measured, turn.

Ultimate Strength is the maximum tensile strength a material can support prior to rupture.

Ultrasonic Extensometer is an electronic instrument used to measure the change in length of a fastener ultrasonically for purposes of calculating strain or stress (preload).

Unfinished Fasteners are made the same basic dimensions as finished fasteners, they have relatively wider tolerances than finished fasteners with all surfaces in their formed condition.

Washer Face is a circular boss on the bearing surface of a bolt or nut.

Width Across Corners of hexagon, square or rectangular shaped fasteners is the distance measured perpendicular to the axis of the fastener from the intersection of two sides to the opposite side of the fastener

Width Across Flats of hexagon or square heads is the distance measured perpendicular to the fastener axis across opposite sides of the fastener

Work Hardening is the increase in hardness and strength produced when a body is loaded past its yield point.

Working Load is the tension in a bolt in use; produced by a combination of residual preload and any external load.

Wrenching Head has provision for driving or holding by means of a wrench, it may be external and internal.

Yield Strength is that stress level which will create permanent deformation of a preselected, amount in a body.

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TR-104213

Printed with soya inks on recycled paper (50% recycled fiber, including 10% postconsu waste) in the United States of America