NASA TECHNICAL STANDARD	NASA-STD-5020
National Aeronautics and Space Administration Washington, DC 20546-0001	Approved: 03-12-2012

REQUIREMENTS FOR THREADED FASTENING SYSTEMS IN SPACEFLIGHT HARDWARE

MEASUREMENT SYSTEM IDENTIFICATION:
METRIC/SI (ENGLISH)

DOCUMENT HISTORY LOG

Status	Document Revision	Approval Date	Description
Baseline		03-12-2012	Initial Release

FOREWORD

This Standard is published by the National Aeronautics and Space Administration (NASA) to provide uniform engineering and technical requirements for processes, procedures, practices, and methods that have been endorsed as standard for NASA programs and projects, including requirements for selection, application, and design criteria of an item.

This Standard is approved for use by NASA Headquarters and NASA Centers, including Component Facilities and Technical and Service Support Centers.

This Standard establishes criteria for ensuring the integrity of threaded fastening systems in launch and space vehicles and their associated equipment.

Requests for information, corrections, or additions to this Standard should be submitted via "Feedback" in the NASA Standards and Technical Assistance Resource Tool at http://standards.nasa.gov.

Original Signed By: ***03-12-2012***

____________________________ ____________________________

Michael G. Ryschkewitsch Approval Date
NASA Chief Engineer

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

APPENDICES

APPENDICES (Continued)

LIST OF FIGURES

LIST OF TABLES

REQUIREMENTS FOR THREADED FASTENING SYSTEMS IN SPACEFLIGHT HARDWARE

1. SCOPE

1.1 Purpose

The purpose of this Standard is to specify requirements for design and analysis of threaded fastening systems in NASA spaceflight hardware.

1.2 Applicability

This Standard is applicable to all NASA programs and projects. This Standard may not be applicable to ground support equipment. Additional requirements for fracture control, non-metallic structures, non-standard fasteners, and fasteners used in extravehicular activity (EVA) may apply.

This Standard is approved for use by NASA Headquarters and NASA Centers, including Component Facilities and Technical and Service Support Centers, and may be cited in contract, program, and other Agency documents as a technical requirement. This Standard may also apply to the Jet Propulsion Laboratory or to other contractors, grant recipients, or parties to agreements only to the extent specified or referenced in their contracts, grants, or agreements.

Requirements are numbered and indicated by the word "shall." Explanatory or guidance text is indicated in italics beginning in section 4. This document shall not supersede applicable laws and regulations unless a specific exemption has been obtained by the Office of the NASA Chief Engineer.

1.3 Tailoring

Tailoring of this Standard for application to a specific program or project shall be formally documented as part of program or project requirements and approved by the Technical Authority.

2. APPLICABLE DOCUMENTS

2.1 General

The documents listed in this section contain provisions that constitute requirements of this Standard as cited in the text.

2.1.1 The latest issuances of cited documents shall apply unless specific versions are designated.

2.1.2 Non-use of specific versions as designated shall be approved by the responsible Technical Authority.

The applicable documents are accessible via the NASA Standards and Technical Assistance Resource Tool at https://standards.nasa.gov or may be obtained directly from the Standards Developing Organizations or other document distributors.

2.2 Government Documents

NASA

NASA-STD-5001	Structural Design and Test Factors of Safety for Spaceflight Hardware
NASA-STD-5019	Fracture Control Requirements for Spaceflight Hardware
NASA-STD-6008	NASA Fastener Procurement, Receiving Inspection, and Storage Practices for Spaceflight Hardware
NASA-STD-6016	Standard Materials and Processes Requirements for Spacecraft

2.3 Non-Government Document

American Society of Mechanical Engineers (ASME)

ASME B107.300-2010	Torque Instruments

2.4 Order of Precedence

This Standard establishes requirements for design and analysis of threaded fastening systems in NASA spaceflight hardware but does not supersede nor waive established Agency requirements found in other documentation.

2.4.1 Document Conflicts

Conflicts between this Standard and other requirements documents shall be resolved by the responsible Technical Authority.

3. ACRONYMS AND DEFINITIONS

3.1 Acronyms and Abbreviations

%	percent
°F	degrees Fahrenheit
AC	Advisory Circular
AND	Air Force-Navy Design
ASME	The American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
CRES	corrosion resistant steel
EVA	extravehicular activity
FAA	Federal Aviation Administration
FS	factor of safety
HDBK	Handbook
JSC	Johnson Space Center
MEK	methyl ethyl ketone
MS	margin of safety
MSFC	Marshall Space Flight Center
MIL	Military
NAS	National Aerospace Standard
NASA	National Aeronautics and Space Administration
NASM	National Aerospace Standard (Metric)
NSTS	National Space Transportation System
SAE	Society of Automotive Engineers
STD	Standard
VDI	Verein Deutscher Ingenieure (German acronym)

3.2 Definitions

3.2.1 Definitions of Variables

Γ	uncertainty or scatter in initial preload
$\Gamma_{a\text{-}max}$	actual preload variation for maximum preload, as determined from a sample of torque-tension test data
$\Gamma_{a\text{-}min}$	actual preload variation for minimum preload, as determined from a sample of torque-tension test data
$\Gamma_{90/95}$	uncertainty or scatter in initial preload (calculated from torque-tension test data) selected using 90-percent probability and 95-percent confidence intervals
δ_e	elastic deformation of a fastening system at rupture under tensile loading
δ_p	plastic deformation of a fastening system at rupture under tensile loading
ϕ	stiffness factor

μ	coefficient of friction
μ	mean preload for a fastener
μ_{nbp}	mean total preload for a joint
σ	standard deviation of preload for a fastener
σ_{nbp}	standard deviation of total preload for a joint
σ_{pi}	unbiased sample standard deviation for initial preload
τ	shear stress
A	cross-sectional area
A_m	minimum minor-diameter area
A_s	cross-sectional area of the full diameter body
A_t	tensile stress area
a	scaling factor applied to the external loads that causes a zero-margin condition
c_{max}	a factor that accounts for the maximum value of the fastener's controlled installation parameter (e.g., effective torque), as allowed by the specified tolerance
c_{min}	a factor that accounts for the minimum value of the fastener's controlled installation parameter (e.g., effective torque), as allowed by the specified tolerance
D	nominal fastener diameter
e	minimum edge distance measured from center of hole to closest edge of any clamped part other than washers
E_b	elastic modulus of the bolt material
E_c	lowest elastic modulus of clamped parts excluding washers
f_{bu}	design ultimate bending stress based on linear-elastic theory
F_{bu}	allowable ultimate flexural stress (bending)
FF	fitting factor
FS	factor of safety
FS_{sep}	factor of safety for separation
F_{su}	allowable ultimate shear stress
FS_u	ultimate factor of safety
FS_y	yield factor of safety
F_{sy}	yield stress in shear (not a typically encountered material property)
F_{tu}	allowable ultimate tensile stress
F_{ty}	allowable yield tensile stress
j	test number
k_b	stiffness of the bolt
k_c	stiffness of the clamped parts local to the fastener
K_{nom}	nominal nut factor
L	joint grip length less total thickness of washers (see figure 7)
L_{lp}	distance between the planes at which an applied tensile load is introduced to the joint
m	number of torque-tension tests (sample size)
M	moment

M_p	fully plastic moment
MS	margin of safety
MS_{sep}	margin of safety for separation
MS_{slip}	margin of safety for joint slip
MS_u	ultimate margin of safety
MS_y	yield margin of safety
n	load-introduction factor
N	axial force
n_f	number of fasteners in a joint
N_p	fully plastic axial force
p	thread pitch
P'	allowable applied load
P'_{sep}	load that causes separation
$P'_{su\text{-}allow}$	allowable ultimate shear load
P'_{tu}	the applied tensile load that causes the fastener load to exceed the fastening system's allowable ultimate tensile load if rupture occurs before separation
P'_{ty}	the applied tensile load that causes the fastener load to exceed the fastening system's allowable yield tensile load, if yielding occurs before separation
P_f	friction load
P_L	limit load
P_p	preload
P_{pc}	loss of preload from material creep
$P_{p\text{-}max}$	maximum preload
$P_{p\text{-}min}$	minimum preload
P_{pi}	initial preload
$P_{pi\text{-}j}$	initial preload obtained in the j-th torque-tension test
$P_{pi\text{-}max}$	maximum initial preload
$P_{pi\text{-}maxa}$	actual maximum initial preload from a sample of test data
$P_{pi\text{-}min}$	minimum initial preload
$P_{pi\text{-}mina}$	actual minimum initial preload from a sample of test data
$P_{pi\text{-}nom}$	Nominal (sample mean) initial preload for a given effective torque
P_{pr}	short term relaxation of preload
P_{sL}	limit shear load
$P_{sL\text{-}joint}$	the total limit shear load acting on the joint
P_{su}	ultimate design shear load
$P_{su\text{-}allow}$	allowable ultimate shear load
P_t	applied tensile load
P_{tb}	tensile load in a preloaded bolt
$P_{t\text{-}joint}$	total applied tensile load acting on a joint
P_{tL}	limit tensile load
$P_{tL\text{-}joint}$	the total limit tensile load acting on the joint
P_{tu}	ultimate design tensile load

$P_{tu\text{-}allow}$	allowable ultimate tensile load
P_{ty}	yield design tensile load
$P_{ty\text{-}allow}$	allowable yield tensile load
P_u	ultimate design load
P_y	yield design load
$P_{\Delta t\text{-}max}$	maximum increase in preload due to temperature
$P_{\Delta t\text{-}min}$	maximum decrease in preload due to temperature
Q	shear
Q_p	fully plastic shear
r	radius
s	two-sided 90/95 tolerance limit factors for a normal distribution (number of standard deviations from the mean used to calculate a value at 90 percent probability and 95 percent confidence)
$T_{br\text{-}min}$	minimum breakaway torque
$T_{L\text{-}max}$	maximum locking torque
T_{max}	maximum effective torque
T_{min}	minimum effective torque
$T_{s\text{-}max}$	maximum torque specified on the engineering documentation
$T_{s\text{-}min}$	minimum torque specified on the engineering documentation
δ_e	elastic deformation of the fastening system at rupture
δ_p	plastic deformation of the fastening system at rupture

3.2.2 Definitions of Terms

Allowable Applied Load: The maximum permissible applied load, which is derived either from the hardware specification (e.g., for fastening system hardware) or from the allowable stress of the materials.

Allowable Load: The maximum permissible load in a structural part or assembly. As applicable, it can be the specified strength of the hardware (e.g., in a fastener specification); a statistically based, lower-bound, load-carrying capacity derived from test data; or the load derived from the allowable stress of the materials.

Applied Load: Force or moment transferred across a joint; it does not include preload or changes in preload as a result of temperature change. (Note: "Applied load" is also referred to in literature as "external load," "externally applied load," or "service load.")

Body: For purposes of this Standard, the ASME B18.12-2001, Glossary of Terms for Mechanical Fasteners, definition applies: "The unthreaded portion of the shank" of a threaded fastener.

Body Diameter: For purposes of this Standard, the ASME B18.12-2001 definition applies: "The diameter of the unthreaded portion of the shank of a threaded fastener." Note that though it appears in some threaded fastener standards, per ASME B18.12-2001, the term "shank diameter" has no meaning for a threaded fastener, as the shank includes the fillet, neck, collar, body, shoulder, thread, point, etc.

Bolt Grip: For fasteners with a flat bearing surface head, the dimension from the bearing surface under the head to the end of the full diameter body, measured parallel to the axis of the fastener. For fasteners with a conical bearing surface head, the dimension from the top of the head to the end of the full diameter body, measured parallel to the axis of the fastener. Note that these definitions are similar to what some fastener standards use as the definitions for "Grip."

Breakaway Torque: For fastening systems incorporating prevailing torque locking features, this is the torque required to initiate relative motion between mating threads when the self-locking mechanism is fully engaged and the fastener is unseated. For the purpose of this Standard, this torque can be measured in either the tightening or loosening direction.

Design Separation Load: The limit tensile load multiplied by a separation factor of safety.

Detrimental Yielding: Yielding that adversely affects fit, form, function, or integrity of the structure.

Effective Torque: A torque in excess of any running torque.

Factor of Safety: A multiplying factor to be applied to limit loads for purposes of analytical assessment (design factor) or test verification (test factor) of design adequacy in strength or stability.

Failure: Rupture, collapse, excessive deformation, or any other phenomenon resulting in the inability of a structure to sustain specified loads, pressures, and environments or to function as designed.

Fastener: For purposes of this Standard, a fastener is a bolt or a screw that joins two or more parts and transfers load between them.

Fatigue: The cumulative irreversible damage incurred in materials caused by cyclic application of stresses and environments, resulting in degradation of load-carrying capability.

Fitting Factor: A supplemental factor of safety used in analysis of bolted joints to account for uncertainties in load paths and stresses.

Full Diameter Body: For purposes of this Standard, the ASME B18.12-2001 definition applies: "The unthreaded portion of a shank whose diameter is generally within the dimensional limits of the major diameter of the thread." Note that some fastener specifications refer to the full diameter body as the "full cylindrical portion of the shank" or simply as a "shank."

Hardware Developer: Organization directly responsible for the design, manufacture, analysis, test, and safety compliance documentation of the hardware.

Joint Grip: A term normally associated with structural bolting, it is the thickness of material or parts that the fastener is designed to secure when assembled. Note that this definition is similar to the ASME B18.12-2001 definition for "Grip."

Limit Load: The maximum expected applied load, including load transferred across joints as a result of thermally induced loading of the structure; it does not include preload or changes in preload occurring as a result of temperature change.

Locking Feature: A device, chemical substance, or other physical characteristic added by design to one or more elements of a threaded fastening system to resist vibration-induced loosening or to provide retention against complete disengagement of the fastening elements. Preload-induced friction forces between mating joint members, mating threads, or under the head or nut elements of a fastening system inherently resist loosening, however, within this standard the term "locking feature" is used to refer only to the added design items described above.

- Prevailing torque feature: Fastening system design that relies on friction to resist fastener rotation independent of preload (i.e., deformed thread or locking patch).
- Adhesive locking feature: Fastening system design which uses chemical compounds added at the time of assembly (i.e., anaerobic adhesive, epoxies, urethanes, etc.).
- Mechanical locking feature: Fastening system design employing non-friction elements usually involving a "hard stop," (i.e., cotter pins or safety wire).

Locking Torque: See "Running Torque."

Margin of Safety: A measure of a structure's predicted reserve strength in excess of the design criteria. For a preloaded fastener, it represents the percentage that the design load (limit load multiplied by the applicable factor of safety) can increase before the design criteria are no longer satisfied. See Eq. (6-1).

Preload: The tensile force in a bolt and the equivalent compressive force in the joint members when there is no applied load.

Prevailing Torque: See Running Torque.

Running Torque: The torque required to overcome kinetic friction of the mating threads plus the torque required to overcome the locking feature when 100 percent of the locking feature is engaged and the fastener is unseated. This torque can be measured in either a loosening or a tightening direction while the mating threads are in relative motion.

Separation: (Also referred to as "gapping.") The state of no compressive load between mating parts local to the fastener. For a joint designed to maintain a seal, it is further defined as any condition that enables a liquid or gas to penetrate the seal at an unacceptable rate.

Separation-Critical Joint: A joint that fails to function as required if separated.

Separation Load: The minimum applied tensile load that causes separation.

Service Life: All significant loading cycles or events during the period beginning with manufacture of a component and ending with completion of its specified use. Testing, transportation, lift-off, ascent, on-orbit operations, descent, landing, and post-landing events are to be considered.

Service Life Factor (Life Factor): A multiplying factor to be applied to the maximum expected number of load cycles in the service life to determine the design adequacy in fatigue or fracture.

Shank: For purposes of this Standard, the ASME B18.12-2001 definition applies: "That portion of a headed fastener that lies between the head and the extreme point end."

Threaded Fastening System: (Also referred to as "fastening system.") An assembled combination of a fastener, an internally threaded part such as a nut or an insert, and also the region of all parts clamped between them, including washers, compressed by the fastener preload.

Ultimate Design Load: The product of the ultimate factor of safety, the fitting factor, and the limit load. Also referred to as "design ultimate load."

Yield Design Load: The product of the yield factor of safety, the fitting factor, and the limit load. Also referred to as "design yield load."

4. GENERAL REQUIREMENTS

This section specifies general requirements for threaded fastening systems.

4.1 Strength

a. Factors of safety shall be as specified in the applicable standard for the structural assembly (e.g., NASA-STD-5001, Structural Design and Test Factors of Safety for Spaceflight Hardware).

b. A supplemental factor, referred to as a fitting factor (≥ 1.0), shall be applied. *See Appendix A.12 for guidance on selecting fitting factor values.*

Factors of safety are applicable to limit load. Factors of safety do not apply to fastener preload or changes in preload as a result of thermoelastic deformation.

4.1.1 Ultimate Design Loads

All threaded fastening systems shall withstand ultimate design loads in conjunction with the applicable maximum expected range of environmental conditions without rupture. *Section 6.2 provides criteria for ultimate-strength analysis. The ultimate design load, P_u, is calculated as*

$$P_u = FF \cdot FS_u \cdot P_L \tag{4-1}$$

where FF is the fitting factor, FS_u is the ultimate factor of safety, and P_L is the limit load. Additional subscripts 't' and 's' are used to designate tensile and shear loads, respectively (e.g., "P_{tu}" for "ultimate design tensile load" and "P_{sL}" for "limit shear load"). See Appendix A.12 for guidance on use of fitting factors.

4.1.2 Yield Design Loads

All threaded fastening systems shall withstand yield design loads in conjunction with the applicable maximum expected range of environmental conditions without detrimental yielding. *Section 6.3 provides criteria for yield-strength analysis. The yield design load, P_y, is calculated as*

$$P_y = FF \cdot FS_y \cdot P_L \tag{4-2}$$

where FS_y is the yield factor of safety. Additional subscripts 't' and 's' are used to designate tensile and shear loads, respectively (e.g., "P_{ty}" for "yield design tensile load"). See Appendix A.12 for guidance on use of fitting factors.

4.2 Fracture Control and Fatigue Life

All threaded fastening systems shall withstand the service life and service environment without fracture or fatigue failure.

Fracture control and fatigue life requirements for threaded fastening systems are the same as those in the general structural requirements levied by NASA on the hardware developer (e.g., NASA-STD-5019, Fracture Control Requirements for Spaceflight Hardware, and NASA-STD-5001). Note that fracture control requirements may not be levied by NASA for some projects, particularly projects for non-human-rated spaceflight hardware. Typically, verification of service life may be by any of the following methods:

a. Crack growth analysis per NASA-STD-5019.

b. Fatigue analysis using the fatigue analysis factor and service life factor specified in the applicable structural requirements (e.g., NASA-STD-5001).

c. Fatigue testing using the service life factor specified in the applicable structural requirements (e.g., NASA-STD-5001).

Alternatively, with the prior approval of the responsible Technical Authority, the low risk approach described below may be used for fasteners in lieu of fatigue analysis or testing. This approach does not relieve any fracture control requirements. A fastener has low risk of fatigue failure if all of the following are true:

- *The fastener is made of A-286 alloy.*

- *The fastener has UNJ or MJ threads rolled after heat treatment and the limit tensile load multiplied by any factors applicable to fatigue analysis does not exceed 50 percent of the fastener's allowable ultimate tensile load, <u>or</u>*

- *The fastener has UNR threads rolled after heat treatment and the limit tensile load multiplied by any factors applicable to fatigue analysis does not exceed 40 percent of the fastener's allowable ultimate tensile load.*

- *The lowest elastic modulus for the material of any clamped part excluding washers is greater than one-third of the elastic modulus for the bolt material. (See discussion in section A.5.)*

- *The minimum edge distance (measured from center of hole to closest edge of any clamped part other than washers) is at least 1.5 times the bolt diameter. (See discussion in section A.5.)*

- *The limit tensile load does not exceed the minimum preload calculated using Eqs. 6-3 and 6-5a.*

- *When threads are in the shear plane, either a dedicated shear transfer device, such as a shear pin (dowel pin), is present or a positive margin of safety for joint slip is shown per section 6.4.*

See Appendix C for justification of the low-risk classification described above.

4.3 Joint Separation

a. Mechanical joints using threaded fastening system hardware shall withstand the design separation load in conjunction with applicable maximum or minimum temperatures without separation, using the minimum separation factor of safety, FS_{sep}, as defined in figure 1, Logic Flow for Minimum Separation Factor of Safety. *Section 6.5 provides criteria for separation analysis.*

b. The joint separation factor of safety shall be multiplied by a fitting factor.

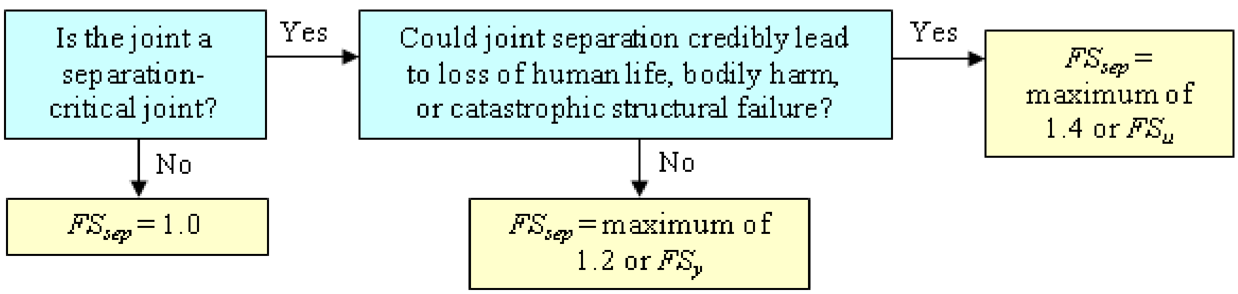

Figure 1—Logic Flow for Minimum Separation Factor of Safety

4.4 Locking Features

a. Regardless of the magnitude of preload, each threaded fastening system in spaceflight hardware shall incorporate a minimum of one locking feature that does not depend upon preload to function.

Devices such as jam nuts may not be suitable locking features to satisfy this requirement if they require preload to function effectively, or if a change in preload could compromise the locking performance. A redundant locking feature may be advisable for some joints as described in Federal Aviation Administration (FAA) Advisory Circular (AC) 20-71, "Dual Locking Devices on Fasteners," December 8, 1970.

b. Locking features shall be verifiable per section 7.6.

c. A mechanical locking feature shall be used on any bolt subject to rotation in operation.

A bolt is considered to be subject to rotation if it serves as an axis of rotation producing relative motion between the bolt and one or more components attached to the bolt. Friction-type locking features include prevailing torque and adhesive locking features which rely on friction to resist loosening or provide retention. On a bolt subject to rotation in operation, the relative motion between parts may produce forces capable of overcoming the locking friction, so friction-type locking features are not suitable except when they are used in conjunction with a mechanical locking feature such as safety wire. For reference, consult FAA AC 20-71, "Dual Locking Devices on Fasteners," December 8, 1970, and FAA AC 23.607-1, "Self-Locking Nuts on Bolts Subject to Rotation," August 24, 1984.

4.5 Fastening System Control Plan

Each hardware developer shall submit a Fastening System Control Plan to the responsible Technical Authority at the Preliminary Requirements Review or equivalent program or project milestone review that:

a. Shows how the requirements in this Standard are to be satisfied.

b. Includes any organization-specific requirements and criteria for design, analysis, fastener installation, and verification.

c. Captures or refers to organization-specific processes for ensuring quality and integrity.

5. DESIGN REQUIREMENTS AND CONSIDERATIONS

This section specifies design requirements for threaded fastening systems.

5.1 Materials

Materials used in threaded fastening systems shall comply with NASA-STD-6016, Standard Materials and Processes Requirements for Spacecraft.

Material issues of particular importance for threaded fastening systems include galling, corrosion, stress-corrosion cracking, hydrogen embrittlement, creep, outgassing, and effects of atomic oxygen. These issues and others are addressed in NASA-STD-6016.

5.2 Specification of Lubricants, Coatings, and Sealants

All lubricants, coatings, and sealants used in threaded fastening systems, as well as their application processes, shall be specified and controlled on the engineering documentation per NASA-STD-6016.

Lubrication of fasteners prior to assembly is often desirable and sometimes necessary to prevent galling or to reduce preload variation when using torque control. For a given applied torque, *use of lubricants, coatings, or sealants can increase the preload developed in a mechanical joint over that of an unlubricated joint. The surfaces on which lubricants, coatings, or sealants are applied, or the surfaces to which they migrate, influence preload as well. Lubricants also reduce wear, inhibit fretting, and improve ease of installation and removal of fasteners.*

Lubricants used in sensitive applications, such as fluid wetted components, optical systems, and some external spacecraft surfaces, can result in system contamination and performance degradation. Overuse of lubricants should be avoided, even in non-sensitive applications.

5.3 Thread Form Compatibility

Mating threaded parts shall have compatible thread forms.

The following are examples of incompatible thread combinations that can result in dimensional interference and may lead to seizure, inadequate strength, inadequate preload for a specified torque value, or inability to disassemble.

- Fine threads mated to coarse threads.
- Inch-based threads mated to metric threads.
- Mated threads with the same pitch and different nominal diameters.
- External UNJ threads mated to UN internal threads.
- External MJ threads mated to M internal threads.

5.4 Use of Washers and Chamfered Bolt Holes

a. The engineering documentation shall specify types, locations, and allowable quantities of washers.

b. Clearance shall be provided for the head-to-shank fillet radius as needed to avoid interference with the clamped parts through the use of chamfered holes or countersunk washers installed under bolt heads with the countersink facing the bolt head (see figure 2, Dimensional Considerations in Selecting Fastening Hardware). *Flush-head screws should be installed without washers under the head.*

5.5 Locking Features

a. Locking features and their installation processes, including verification methods, shall be specified in the engineering documentation.

b. When using locking adhesives, whether as thread-locking compounds or staking materials, installation processes shall be developed and validated prior to implementation to ensure adhesives cure, adhere, and function as expected.

It should be recognized that some locking features are not a means of maintaining preload but rather are safety devices principally intended to resist rotational loosening and prevent loss of fasteners. Examples of the selection of locking features are provided in Appendix B.

5.6 Thread Engagement, Dimensions, and Tolerances

Dimensions, tolerances, and fastening system hardware shall be specified in the engineering documentation to control the features and issues described in the following subsections.

All parts of the threaded fastening system should be dimensionally compatible.

5.6.1 Fastener Length Selection for Thread Engagement

When the threaded fastening system incorporates a prevailing torque locking feature, the fastener length shall be sufficient for fully formed threads to engage the locking feature.

To ensure that all internal threads are fully engaged, the length of each fastener used with a nut, nut plate, or insert should be selected to extend a distance of at least twice the thread pitch, p, past the outboard end of the nut, nut plate, or insert. Figure 2 points out common methods of addressing dimensional issues in hardware selection for a fastener used with a nut.

To avoid thread stripping as the critical failure mode, thread engagement in an internally threaded part other than a nut, nut plate, or insert should be selected to ensure the minimum number of engaged complete threads such that the fastener would fail in tension before threads would strip. It is desirable to have the fastener fail in tension before the threads strip to facilitate ease of repair and to increase the likelihood that any failed hardware may be detected. In addition, fastener tensile failures typically exhibit more plastic deformation prior to rupture, which allows loads to redistribute between fasteners before failure occurs in any one threaded fastening system.

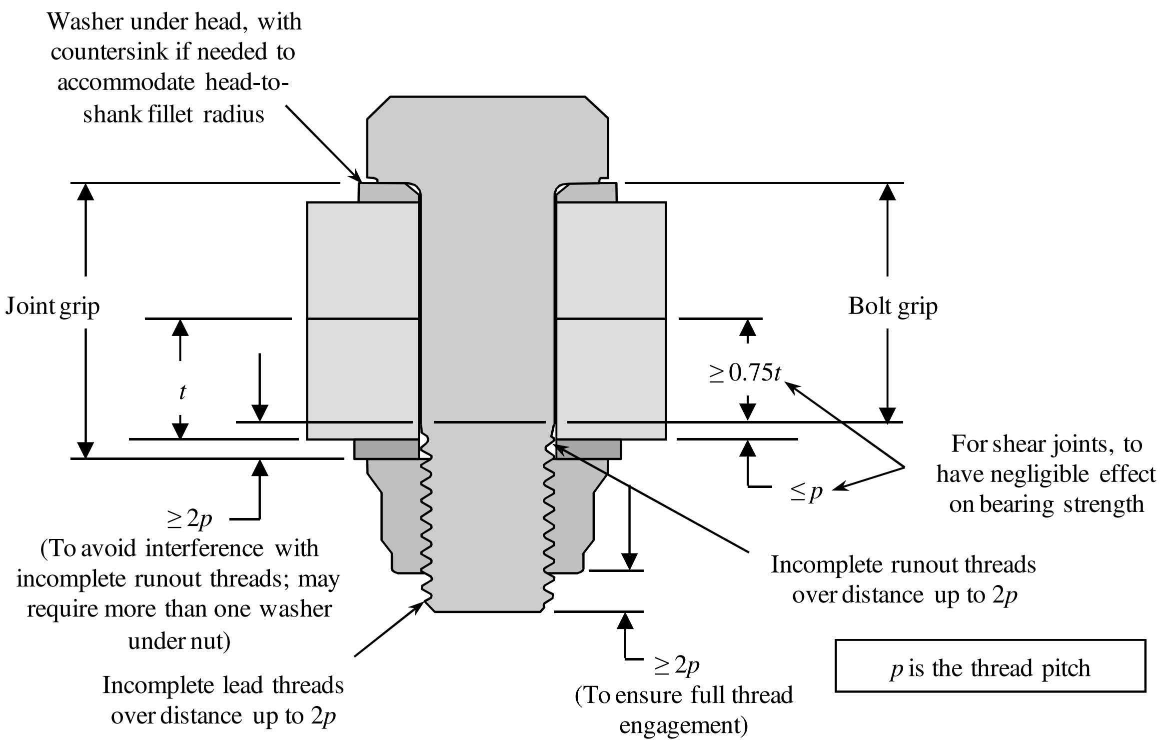

Figure 2—Dimensional Considerations in Selecting Fastening Hardware

5.6.2 Bolt Grip Selection to Prevent Interference

a. For a fastener with a full diameter body, the bolt grip and the number and type of washers shall be selected to ensure the internal threads do not encroach on the incomplete runout threads of the fastener.

Where complete threads transition to the unthreaded full diameter body, incomplete runout threads exist over a distance up to twice the pitch on most commercial aerospace fasteners. If the nut or the insert engages the incomplete runout threads, interference may occur and the intended preload may not be achieved.

b. Fasteners threaded into blind holes shall be selected to prevent contacting the bottom of the hole or interfering with incomplete internal threads.

Where blind holes are tapped, incomplete internal threads are present at the bottom of the hole.

5.7 Fastener Installation Specification and Control

The engineering drawings and assembly procedures shall specify and control installation methods and parameters for achieving preload.
Common methods of achieving a desired preload during installation are controlling the installation torque, controlling the turn angle of a nut or bolt head after the fastening system is

seated, measuring the fastener's change in length, and using strain gages on the fastener body. These methods are listed in the order of increasing labor, cost, and accuracy. The most economical method, torque control, is the least accurate for controlling preload.

5.8 Installation Torque Specification and Control

The requirements in this section apply when preload is generated by torque control.

a. The engineering documentation shall specify the installation torque range or specify an applicable standard that defines the installation torque range.

b. The engineering documentation shall clearly identify when the installation torque is the torque above running torque.

When engineering documentation does not specify "above running torque," the specified torque is understood to be the final torque indicated by the torque instrument. Running torque does not generate preload. Specifying torque above a measured running torque provides tighter control of preload. Refer to section 6.1.

The engineering documentation should specify whether torque is to be applied to the fastener head or the nut because the relationship between torque and preload is different depending upon which part is torqued. Whenever possible, torque should be applied to the nut. When torque is applied to the fastener head, the fastening system should be designed to ensure no binding occurs between any parts of the rotating shank and its clearance hole.

When it is important for joints with multiple fasteners to have an evenly distributed clamp load, an appropriate torque sequence along with incremental torquing should be specified on the engineering drawing or hardware-specific installation procedure. Example procedures for multi-fastener preloading can be found in MIL-HDBK-60, Threaded Fasteners-Tightening to Proper Tension, section 5.7: Flange Joint Bolt Tensioning.

6. CRITERIA FOR ANALYSIS OF THREADED FASTENING SYSTEMS

This section applies to analysis of a threaded fastening system based on the assumption that the applied load acting on that fastening system is known or is adequately predicted. The analysis performed to determine how applied loads distribute between fasteners is outside the scope of this document. Appendix A provides technical rationale for selected analysis criteria in this section. The analysis criteria herein apply only to fasteners made of ductile metallic materials. Analysis of threaded fasteners made of brittle or nonmetallic materials should be based on test data.

As applicable, analysis results for ultimate strength (section 6.2), yield strength (section 6.3), joint slip (section 6.4), and joint separation (section 6.5) should be reported with a margin of *safety, MS, per the general Eq. (6-1). The margin of safety indicates how much the applied load can increase before the criteria are no longer satisfied.*

$$MS = \frac{P'}{FF \cdot FS \cdot P_L} - 1 \tag{6-1}$$

where P' is the allowable applied load and FS is the factor of safety.

All margins of safety shall be greater than or equal to zero ($MS \geq 0$).

6.1 Nominal, Maximum, and Minimum Preloads

a. Analysis of threaded fastening systems shall address maximum and minimum preloads, accounting for the preload variation associated with the method of developing initial preload, potential relaxation, creep, and the effects of maximum and minimum expected temperatures, as shown in table 1, Analytical Adjustments of Maximum and Minimum Preloads.

Maximum and minimum preloads are calculated as

$$P_{p-\max} = P_{pi-\max} + P_{\Delta t-\max} \tag{6-2}$$

$$P_{p-min} = P_{pi-min} - P_{pr} - P_{pc} - P_{\Delta t-min} \tag{6-3}$$

where $P_{pi\text{-}max}$ *and* $P_{pi\text{-}min}$ *are the maximum and minimum initial preloads, respectively; the other variables are defined in table 1.*

Table 1—Analytical Adjustments of Maximum and Minimum Preloads

Description	Symbol	Adjustment used when calculating maximum preload	Adjustment used when calculating minimum preload
Short-term relaxation of preload (embedment of imperfectly matched surfaces); see Appendix A.3	P_{pr}	Zero	5% of minimum initial preload for joints with all-metallic clamped parts, test-derived value for joints with any non-metallic parts or coatings
Loss of preload from material creep	P_{pc}	Zero	Calculated maximum expected preload loss from creep, if applicable
Change of preload with temperature (differential coefficients of thermal expansion and temperature dependence of elastic moduli)*	$P_{\Delta t\text{-}max}$, $P_{\Delta t\text{-}min}$	Calculated maximum increase in preload, $P_{\Delta t\text{-}max}$, for maximum or minimum expected temperatures	Calculated maximum decrease (as a positive number) in preload, $P_{\Delta t\text{-}min}$, for maximum or minimum expected temperatures

* An effective technique to minimize the change of preload with temperature is described in the SAE AIR 1754A "Washer, Thermal Compensating, Metric Series."

Maximum initial preload for strength and fatigue analyses is calculated as

$$P_{pi-\max} = c_{\max}(1+\Gamma)P_{pi-nom} \tag{6-4}$$

where $P_{pi\text{-}nom}$ is the nominal (mean) preload applicable to installation, Γ is the preload variation, and $c_{\max}$ is a factor that accounts for the maximum value of the controlled installation parameter, as allowed by the specified tolerance. For example, if torque control is used and the effective torque is specified as 40 ± 2 N·m, then $c_{\max}$ = (40 + 2)/40 = 1.05.

For use in separation analysis of separation-critical joints and for fatigue analysis, minimum initial preload is calculated as

$$P_{pi-\min} = c_{\min}(1-\Gamma)P_{pi-nom} \tag{6-5a}$$

where $c_{\min}$ is a factor that accounts for the minimum value of the controlled installation parameter, as allowed by the specified tolerance. For example, if torque control is used and the effective torque is specified as 40 ± 2 N·m, then $c_{\min}$ = (40 – 2)/40 = 0.95.

For use in joint-slip analysis and separation analysis of joints that are not separation-critical, minimum initial preload is calculated as

$$P_{pi-\min} = c_{\min}\left(1 - \frac{\Gamma}{\sqrt{n_f}}\right) P_{pi-nom} \qquad \text{(6-5b)}$$

where n_f is the number of fasteners in the joint. See Appendix A.2 for rationale.

In addition to the adjustments noted in table 1, loss of preload may occur if the joint experiences cyclic slip. To minimize any such loss, the joint should be designed in any one of the following ways:

(1) Not to slip (see section 6.4).

(2) With minimal clearance fits achieved by specifying precision fasteners having a closely controlled or "close tolerance" full diameter body in conjunction with precision holes per National Aerospace Standard (NAS) 618.

(3) With shear pins (dowel pins) or other dedicated shear-transfer devices.

b. Calculation of the nominal (mean) initial preload, P_{pi-nom}, shall be substantiated by tests of a minimum of six sets of the fastening system hardware to determine the relationship between initial preload and the parameter controlled during installation (torque, turn-of-nut, turn-angle, or bolt stretch). *Other requirements pertaining to these tests depend on the parameter controlled during installation:*

(1) Torque control (*see NASM 1312-15, Fastener Test Methods, Method 15, Torque-Tension for guidance for torque-tension testing).*

A. The fastening system hardware shall be of materials that are the same as used in the flight assembly.

B. The fastening system hardware shall be installed using the same lubricants and lubrication process as used on the flight assembly.

C. *At least three tests (torque and removal) should be performed on each of the six sets of fastening system hardware for a total of eighteen tests, unless reuse is prohibited in the flight assembly.*

(2) Turn-of-nut, turn-angle, or bolt-stretch control

A. The fastening system hardware shall be of the same specification (part number) as the flight assembly fasteners.

B. The installation process shall be the same as for the flight assembly.

c. For separation-critical joints, the preload variation, Γ, used to calculate the minimum initial preload shall be derived from test data at a statistical basis of 90 percent probability and 95 percent confidence (two-sided distribution). *Test requirements depend on the parameter controlled during installation:*

(1) Torque control. The tests performed to determine Γ shall meet the requirements of paragraph 6.1.b.1 and should conform to either of the following approaches:

A. Lot-specific testing (testing the actual procurement lot of fastening system hardware that will be used for spaceflight), with each fastener installed at least three times unless reuse is prohibited for the flight assembly.

B. Testing equal numbers of sets of fastening system hardware of the same specification as the flight-assembly hardware, from each of at least three procurement lots, with each fastener installed at least three times unless reuse is prohibited for the flight assembly.

(2) Turn-of-nut, turn-angle, or bolt-stretch control. The tests performed to determine Γ shall meet the requirements of paragraph 6.1.b.2.

d. The preload variation, Γ, used to calculate maximum initial preload and to calculate minimum initial preload for joints that are not separation-critical shall be as follows, depending on the parameter controlled during installation:

(1) Torque control. Either of the following approaches may be used:

A. In absence of applicable data showing otherwise, Γ = 25 percent for fasteners lubricated at assembly, and Γ = 35 percent for non-lubricated and as-received. If the actual extreme variation found in the tests performed per paragraph 6.1.b exceeds the Γ values stated above, then the actual extreme variation should be used or methods for reducing the preload variation should be pursued. A fastener is considered to be lubricated if a solid-film, grease, or liquid lubricant is applied to the threads and to the turning bearing surfaces of the nut or the bolt head (or to the washer under the nut or the bolt head).

B. Γ to be obtained from test data at a statistical basis of 90 percent probability and 95 percent confidence (two-sided distribution), with the tests conforming to the requirements of paragraph 6.1.b.1.

(2) Turn-of-nut or turn-angle control. Either of the following approaches may be used:

A. Γ = 25 percent or the actual variation from the sample mean found in the tests performed per paragraph 6.1.b., whichever is greater.

B. Γ to be obtained from test data at a statistical basis of 90 percent probability and 95 percent confidence (two-sided distribution), with the tests conforming to the requirements of paragraph 6.1.b.1.

(3) Bolt stretch control methods such as ultrasonic measurement, bolt elongation measurement, and instrumented bolts. Either of the following approaches may be used:

A. Γ = 10 percent or the actual variation from the sample mean found in the tests performed per paragraph 6.1.b.2, whichever is greater.

B. Γ to be obtained from test data at a statistical basis of 90 percent probability and 95 percent confidence (two-sided distribution), with the tests conforming to the requirements of paragraph 6.1.b.2.

Appendix A.2 shows an example of how to calculate minimum and maximum preloads when preload is generated with torque control.

6.2 Strength Under Ultimate Design Loads

Analysis for ultimate design loads shall address potential rupture in all elements of the threaded fastening system, including the fastener, the internally threaded part, such as a nut or an insert, and the clamped parts.

Each internally threaded part used for design should have either a specified and controlled allowable tensile load, or an allowable tensile load derived from dedicated testing. Many threaded inserts have two allowable loads, one directly provided in the insert specification (or the procurement specification) and one calculated for pull-out based on data typically provided in the insert specification. The lower value should be used for strength analysis.

For inserts installed in nonhomogeneous or nonmetallic materials or in sandwich panels, allowable pull-out loads should be derived from test. An insert's allowable pull-out load depends on the material in which the insert is installed (parent material). The most common failure mode associated with pull-out is shear of internal threads in the parent material. The specifications (or procurement specifications) for most threaded inserts define how the allowable pull-out load is calculated, such as by multiplying a specified minimum shear engagement area by the allowable ultimate shear stress of the parent material. Such an allowable pull-out load applies when the insert is installed in a solid, homogenous material.

6.2.1 Ultimate Strength Analysis for Tensile Loading

Ultimate strength analysis of a fastening system under applied tensile loading shall be performed as follows:

a. Do not include preload when separation occurs before rupture—see section 6.2.1.1.

b. Include preload when rupture occurs before separation—see section 6.2.1.2.

Whether separation would occur before rupture can be determined based on test or analysis. Refer to Appendix A.5 for guidance.

6.2.1.1 Ultimate Tensile Strength Analysis for Separated Joints

If separation occurs before rupture, the ultimate margin of safety, MS_u, for tensile loading is

$$MS_u = \frac{P_{tu-allow}}{FF \cdot FS_u \cdot P_{tL}} - 1 \qquad (6\text{-}6)$$

where $P_{tu\text{-}allow}$ is the allowable ultimate tensile load for the fastener, the internally threaded part, or the threaded interface, whichever is lower, and P_{tL} is the limit tensile load.

6.2.1.2 Inclusion of Preload in Ultimate Tensile-Strength Analysis with Linear Theory

If it is not clear whether rupture occurs before separation, linear theory should be used to assess ultimate tensile strength of the fastening system. Appendix A.6 provides further detail on this approach. With such theory, the tensile load in a preloaded bolt, P_{tb}, increases proportionally with the applied tensile load per Eq. (6-7) until either any part in the fastening system ruptures or separation occurs.

$$P_{tb} = P_p + n\phi P_t \tag{6-7}$$

where P_p is the preload, P_t is the applied tensile load, n is the load-introduction factor which depends on the joint design and accounts for where the load is applied to the joint, and ϕ is the stiffness factor. The stiffness factor is

$$\phi = \frac{k_b}{k_b + k_c} \tag{6-8}$$

where k_b is the stiffness of the bolt and k_c is the stiffness of the clamped parts local to the fastener. See Appendix A.4 for discussion of the load-introduction factor.

Based on Eq. (6-7) and the assumption of maximum preload, $P_{p\text{-}max}$, the applied tensile load that causes the bolt load to exceed the allowable ultimate tensile load for the fastening system, P'_{tu}, is

$$P'_{tu} = \frac{1}{n\phi}\left(P_{tu-allow} - P_{p-max}\right) \tag{6-9}$$

and the linearly projected load that causes separation when at maximum preload is

$$P'_{sep} = \frac{P_{p-max}}{1-n\phi} \tag{6-10}$$

If P'_{sep} is less than P'_{tu}, linear theory predicts that separation would occur before rupture, and the ultimate margin of safety for tensile loading is calculated per Eq. (6-6).

If P'_{sep} is greater than P'_{tu}, linear theory predicts that rupture would occur before separation, and the ultimate margin of safety for tensile loading is

$$MS_u = \frac{P'_{tu}}{FF \cdot FS_u \cdot P_{tL}} - 1 \tag{6-11}$$

6.2.2 Ultimate Strength Analysis for Shear Loading

a. Ultimate strength analysis of a fastening system under applied shear loading shall be based on the logic flow in figure 3, Ultimate Strength Analysis of Fasteners Under Shear Loading.

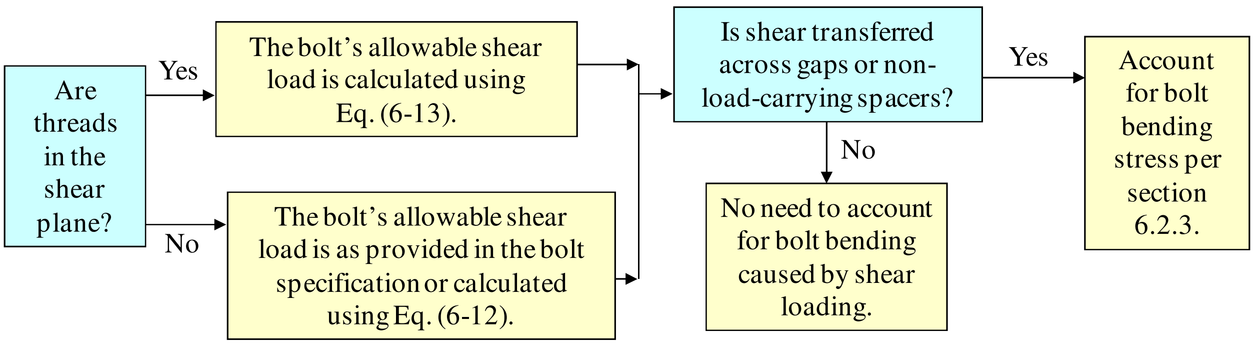

Figure 3—Ultimate Strength Analysis of Fasteners Under Shear Loading

b. Ultimate strength analysis of bolts under shear loading shall be based on the assumption that no shear load is carried by friction between the faying surfaces.

Ultimate strength of the fastener should not be dependent on preload and friction. Once a bolt begins to yield under the combination of preload and applied load, preload relaxes (see Appendix A.7). Therefore, it is not conservative to assume that friction carries any of the shear load when assessing ultimate strength.

The allowable ultimate shear load for a fastener depends on whether the threads are in the shear plane. If threads are not in the shear plane, the allowable ultimate shear load per shear plane is half the value given in the fastener specification for double-shear joints, when applicable, or is calculated by

$$P_{su-allow} = F_{su} \pi D^2 / 4 \quad (6\text{-}12)$$

where F_{su} is the allowable ultimate shear stress for the fastener. If threads are in the shear plane, the allowable ultimate shear load for a fastener is calculated by

$$P_{su-allow} = F_{su} A_m \quad (6\text{-}13)$$

where A_m is the minimum minor-diameter area for the fastener threads. The ultimate margin of safety for shear loading of a fastener is

$$MS_u = \frac{P_{su-allow}}{FF \cdot FS_u \cdot P_{sL}} - 1 \quad (6\text{-}14)$$

where P_{sL} is the limit shear load acting on the shear plane.

6.2.3 Ultimate Strength Analysis for Interaction of Tension, Shear, and Bending

For fasteners under simultaneous applied tensile and shear loads, along with any applicable bending, analysis shall account for interaction of the combined loading. *Preload may be omitted in the interaction check.*

6.2.3.1 Shear Acting Without Threads in the Shear Plane

For the case in which the full-diameter body is in the shear plane rather than the threads, satisfying either of the criteria given by Eqs. (6-15) and (6-16) is acceptable. The criterion given by Eq. (6-15) should be satisfied when not accounting for plastic bending, or the criterion given by Eq. (6-16) should be satisfied when accounting for plastic bending.

$$\left(\frac{P_{su}}{P_{su-\text{allow}}}\right)^{2.5} + \left(\frac{P_{tu}}{P_{tu-\text{allow}}} + \frac{f_{bu}}{F_{tu}}\right)^{1.5} \leq 1 \quad (6\text{-}15)$$

$$\left(\frac{P_{su}}{P_{su-\text{allow}}}\right)^{2.5} + \left(\frac{P_{tu}}{P_{tu-\text{allow}}}\right)^{1.5} + \left(\frac{f_{bu}}{F_{bu}}\right) \leq 1 \quad (6\text{-}16)$$

where $P_{su-allow}$ is calculated using Eq. (6-12), P_{su} is the ultimate design shear load per shear plane, f_{bu} is the design ultimate bending stress based on linear-elastic theory, F_{tu} is the allowable ultimate tensile stress, and F_{bu} is the allowable ultimate flexural stress (bending), which applies for ductile materials as a result of plastic bending. Use of F_{bu}, which depends on the material and the cross section, accounts for the additional moment-carrying capacity of a section in bending over and above the capacity calculated based on linear-elastic theory.

Inclusion of the bending stress term in the criteria given in Eqs. (6-15) and (6-16) is considered to be conservative. These criteria, with omission of the bending stress term, are based on tests of A-286 3/8-24 (NAS1956C14) fasteners performed at NASA Marshall Space Flight Center (MSFC) in 2010. Shear-induced bending stress does not need to be included in Eqs. (6-15) and (6-16) unless there are gaps or shims between the faying surfaces.

6.2.3.2 Shear Acting with Threads in the Shear Plane

For the case in which the threads are in the shear plane, as is typically the case for a bolt going into a tapped hole or a threaded insert, satisfying either of the criteria given by Eqs. (6-17) and (6-18) is acceptable. The criterion given by Eq. (6-17) should be satisfied when not accounting *for plastic bending, or the criterion given by Eq. (6-18) should be satisfied when accounting for plastic bending.*

$$\left(\frac{P_{su}}{P_{su-\text{allow}}}\right)^{1.2}+\left(\frac{P_{tu}}{P_{tu-\text{allow}}}+\frac{f_{bu}}{F_{tu}}\right)^{2}\leq 1 \tag{6-17}$$

$$\left(\frac{P_{su}}{P_{su-\text{allow}}}\right)^{1.2}+\left(\frac{P_{tu}}{P_{tu-\text{allow}}}\right)^{2}+\left(\frac{f_{bu}}{F_{bu}}\right)\leq 1 \tag{6-18}$$

where $P_{su\text{-}allow}$ is calculated using Eq. (6-13). The shear capability of a fastener is greatest when the full diameter body is in the shear plane and is reduced when the threads are in the shear plane due to cross-sectional area reduction.

Note the difference in exponents used in Eqs. (6-17) and (6-18) (threads in shear plane) as compared with those used in Eqs. (6-15) and (6-16) (full diameter body in shear plane). Tensile and shear stresses peak at the same cross section when the threads are in the shear plane; when the full diameter body is in the shear plane, the tensile and shear stresses do not peak at the same cross section. Inclusion of the bending stress term in the criteria given in Eqs. (6-17) and (6-18) is considered to be conservative. These criteria, with omission of the bending stress term, are based on tests of A-286 3/8-24 (NAS1956C14) fasteners performed at MSFC in 2010. Shear-induced bending stress does not need to be included in Eqs. (6-17) and (6-18) unless there are gaps or shims between the faying surfaces.

6.3 Strength Under Yield Design Loads

If one or more of the following applies:

- Fastener yielding causes the joint to separate under an applied tensile load that is less than the design separation load,
- Fastener yielding causes the joint to suffer detrimental slip under an applied shear load that is less than the applicable design shear load,
- Some other design-specific reason exists for why fastener yielding is detrimental (e.g., any fastener yielding that adversely affects the form, fit, or function of the design),

then fastener yielding is detrimental and analysis shall show the fastener's total tensile load, when accounting for maximum preload and the yield design tensile load, does not exceed the allowable yield tensile load, as defined below. *If fastener yielding is not detrimental, yield-strength analysis for the fastener is not required.*

In a typical joint with preload generated by torque control, fastener yielding under combination of preload and applied load is not detrimental for separation (see first bullet above), given the separation analysis criteria specified in section 6.5. Rationale and exceptions for the above *statement, as well as a discussion of the effects of fastener yielding on joint slip (see second bullet above), are provided in Appendix A.9.*

If yielding is detrimental, a fastener's allowable yield tensile load is

$$P_{ty-\text{allow}} = \left(\frac{F_{ty}}{F_{tu}}\right) P_{tu-\text{allow}} \tag{6-19}$$

where F_{ty} *is the allowable yield tensile stress.*

Based on Eq. (6-7), with maximum preload, $P_{p\text{-}max}$*, the applied tensile load that causes the bolt load to exceed the allowable yield tensile load is*

$$P'_{ty} = \frac{1}{n\varphi}\left(P_{ty-\text{allow}} - P_{p-\max}\right) \tag{6-20}$$

If P'_{sep} *(Eq. (6-10)) is less than* P'_{ty}*, separation occurs before yield, and the yield margin of safety for tensile loading is*

$$MS_y = \frac{P_{ty-\text{allow}}}{FF \cdot FS_y \cdot P_{tL}} - 1 \tag{6-21}$$

If P'_{ty} *is less than* P'_{sep}*, yield occurs before separation, and the yield margin of safety for tensile loading is*

$$MS_y = \frac{P'_{ty}}{FF \cdot FS_y \cdot P_{tL}} - 1 \tag{6-22}$$

Yield-strength analysis for the fastener is not required for shear or the shear component of combined loading.

6.4 Friction as a Load Path for Shear Loading: Joint-Slip Analysis

Friction may be included as a shear-load path for the following types of analysis for joints using preloaded fasteners:

- *Yield-strength analysis of the joint members (e.g., bearing).*
- *Analysis for ensuring a structural assembly maintains required alignment of mechanical interfaces.*
- *Fatigue or fracture mechanics analysis.*

When friction is used as a load path for shear loading, the analysis should show no slippage at the applicable design load (see Appendix A.10 for guidance on joint-slip analysis). If the margin of safety for joint slip is negative, then friction cannot be used as a load path for shear loading.

Note that section 6.2.2 requires positive margins of safety for ultimate design loads without reliance on friction to ensure high structural reliability regarding catastrophic failure.

The coefficient of friction for joint-slip analysis shall be no greater than the following unless otherwise substantiated by test:

- *0.2 for uncoated, non-lubricated metal surfaces that are cleaned by a qualified process and visibly clean at and after assembly.*
- *0.1 for all other surfaces. This category includes nonmetallic (coated or uncoated) surfaces and metallic surfaces that are coated with any substance, including lubricant, paint, and conversion coating.*

Higher values may be used with program- or project-approved testing in a relevant environment.

Relying on friction to carry shear across a joint is a departure from past NASA policy. This departure is intentional for the purpose of encouraging good design practice. In structural assemblies subject to cyclic loads or vibration, such as aerospace structures, joints using threaded fasteners are most dependable if shear is transferred by friction or by close-fit or interference-fit shear pins (dowel pins) rather than by the fasteners themselves. When shear pins (dowel pins) are not practical, the fasteners should be highly preloaded, and care should be taken to clean the faying surfaces and prevent contamination from oils or other lubricants in order to maximize the coefficient of friction.

If the joint members slip within the limits of the fasteners' clearance holes,

- *A nonlinearity is introduced within the assembly, which can compromise the accuracy of dynamic loads analysis and predicted load distribution.*
- *A damaging shock can be introduced when the fasteners contact the sides of the clearance holes.*
- *The first fasteners within a pattern to contact the sides of the clearance holes are potentially overloaded.*
- *The structural assembly may suffer detrimental misalignment.*
- *If the slip is cyclic, the fasteners may lose preload despite the presence of locking features.*
- *If the slip is cyclic and the fastener threads are in the shear plane, as is typically the case for fasteners threading into tapped holes or threaded inserts, the fastener may fail in fatigue.*

If friction is inadequate for preventing slip at limit load and no shear pin (dowel pin) is present, the severity of the above effects should be reduced by ensuring minimal clearance fits, achieved by specifying precision fasteners having a closely controlled or "close tolerance" full diameter body in conjunction with precision holes per NAS 618, Fastener – Recommended Shank, Hole, and Head-to-shank Fillet Radius Limits For. Fasteners installed into tapped holes or threaded inserts typically require relatively large-clearance fits between fasteners and their holes to account for location tolerances. As such, shear pins (dowel pins) should be used unless friction is adequate for preventing slip at limit load.

6.5 Joint Separation Analysis

a. Analysis shall show no separation for each threaded fastening system that is subject to applied tensile loading, with the assumption of minimum preload (see Appendix A.11).

The margin of safety for separation, MS_{sep}, is

$$MS_{sep} = \frac{P_{p-min}}{FF \cdot FS_{sep} \cdot P_{tL}} - 1 \qquad (6\text{-}23)$$

where FS_{sep} is the separation factor of safety per section 4.3.

b. For a joint that maintains a seal (e.g., to maintain pressure or contain a fluid), analysis shall show that the seal meets its requirements at the design separation load when assuming minimum preload for all fasteners in the joint. Eq. (6-23) is not adequate for this situation; the analysis approach depends on the design of the joint and the seal.

7. QUALITY ASSURANCE

This section specifies requirements for ensuring quality of threaded fastening systems. Essential parts of a quality-assurance program include in-house processes and controls for ensuring quality of raw materials, fastening system hardware, design engineering, analysis, manufacturing, assembly, and test. Every aspect that can affect the joint's ability to meet the requirements specified herein should be controlled. Examples of process controls are standard parts lists, design criteria, checks and balances in the design process, inspections, effective use of testing, and standards or procedures for fastener installation.

7.1 As-Built Documentation

Records documenting the as-built procedures and configuration of each assembly shall be created, and maintained by the hardware developer and provided for review upon request by the responsible Technical Authority.

7.2 Training

Personnel installing fastening system hardware shall be trained per program, project, or organization-specific quality process.

7.3 Installation Tools and Instruments

a. Tools and instruments used to install fastening system hardware shall be used within their design and calibration ranges.

b. Torque instruments shall conform to ASME B107.300-2010, Torque Instruments, or a NASA-approved alternative. Torque instruments should be chosen so the torque (running or final assembly) being measured or controlled is between 20 and 90 percent of the instruments' full-scale torque.

7.4 Calibration of Installation Tools and Instruments

a. Installation tools and instruments shall be under calibration control and be verified to be in calibration prior to use.

b. If a calibrated tool or instrument is dropped, struck, or otherwise damaged or suspected of being out of calibration, the calibration shall be re-verified before further use.

7.5 Threaded Fastening System Hardware Inspection

Threaded fastening system hardware shall be inspected prior to installation to verify that part number(s), cleanliness, and orientation are in accordance with the engineering documentation.

7.6 Locking Feature Verification

a. Mechanical locking features, such as cotter pins, safety wire, and safety cable, shall be verified by visual inspection after installation. Proper installation processes are located in specifications, such as National Aerospace Standard NASM 33540, Safety Wiring, Safety Cabling, Cotter Pinning, General Practices for and Society of Automotive Engineers (SAE) International AS567, Safety Cable, Safety Wire, Key Washers, and Cotter Pins for Propulsion Systems, General Practices for Use of (Revision J).

b. Prevailing torque features, such as deformed thread features, pellets, strips, or patches, shall be verified by torque measurement during the installation process.

c. Adhesive locking features dependent upon substrate and/or configuration for cure, such as anaerobic liquid locking compounds, shall be verified by torque measurements on witness coupons that are representative of and processed with the hardware being verified.

d. All other adhesive locking features shall be verified using cure samples processed at the time of application/processing.

Example verification methods for each of the locking feature types listed above are located in various sections of Appendix B.

7.7 Fastening System Hardware Procurement, Inspection, and Storage

Procurement, receiving inspection, and storage of threaded fastening system hardware, with the exception of the clamped parts, shall meet the requirements of NASA-STD-6008, NASA Fastener Procurement, Receiving Inspection, and Storage Practices for Spaceflight Hardware.

APPENDIX A

EXPLANATION AND JUSTIFICATION OF FASTENER ANALYSIS CRITERIA

A.1 Purpose of Appendix A

The purpose of Appendix A is to explain and justify fastener analysis criteria.

A.2 Accounting for Preload Variation when Installing Fasteners with Torque Control (Supplement to Section 6.1)

This section is intended to clarify the criteria in section 6.1 and to provide additional guidance regarding calculation of maximum and minimum preloads when using torque control.

The relationship between preload and torque can be characterized by a statistical distribution. Multiple tests have shown that this distribution is usually normal (Gaussian), and the mean and standard deviation can be inferred from the test data with varying degrees of confidence. Thus, for a given input torque, there is a normal probability distribution for the resulting preload having a mean and a standard deviation.

The relationship between the nominal (mean) initial preload, $P_{pi\text{-}nom}$, and the effective torque, T, can be expressed as

$$P_{pi-nom} = \frac{T}{K_{nom} D} \tag{A.2-1}$$

where K_{nom} is the nominal (mean) nut factor and D is the nominal bolt diameter.

Maximum initial preload is calculated by modifying Eq. (6-4) as

$$P_{pi-\max} = \frac{(1+\Gamma)T_{\max}}{K_{nom} D} \tag{A.2-2}$$

where Γ is the preload variation, and T_{max} is the maximum effective torque. For use in separation analysis of separation-critical joints, minimum initial preload is calculated by modifying Eq. (6-5a) as

$$P_{pi-\min} = \frac{(1-\Gamma)T_{\min}}{K_{nom}D} \tag{A.2-3a}$$

where T_{min} is the minimum effective torque. For use in separation analysis of joints that are not separation-critical and joint-slip analysis, minimum initial preload is calculated by modifying Eq. (6-5b) as

$$P_{pi-\min} = \left(1 - \frac{\Gamma}{\sqrt{n_f}}\right)\frac{T_{\min}}{K_{nom}D} \tag{A.2-3b}$$

The maximum and minimum effective torques are calculated as follows:

- If the installation torque is specified as some value or range "above running torque," or if a locking feature that does not introduce a running torque is used,

$$T_{\max} = T_{s-\max} \tag{A.2-4}$$

$$T_{\min} = T_{s-\min} \tag{A.2-5}$$

where T_{s-max} and T_{s-min} are the maximum and minimum specified torques, respectively. For example, if the torque specification is "60 ± 3 in·lb above running torque," $T_{max} = T_{s-max} = 63$ in·lb and $T_{min} = T_{s-min} = 57$ in·lb.

- Otherwise, the maximum and minimum effective torques are calculated by subtracting the extreme values of running torque from the specified torques:

$$T_{\max} = T_{s-\max} - T_{br-\min} \tag{A.2-6}$$

$$T_{\min} = T_{s-\min} - T_{L-\max} \tag{A.2-7}$$

where T_{br-min} is the minimum breakaway torque specified for the locking feature (can be conservatively set equal to zero) and T_{L-max} is the maximum locking torque (running torque) specified for the locking feature.

Section 6.1b herein requires that calculation of the nominal initial preload, P_{pi-nom}, be substantiated by test. Because the torque-preload relationship is not truly linear, as Eq. (A.2-1) suggests, the most accurate method of determining P_{pi-nom} for a given effective torque, T, is to compute the sample mean (average) preload measured at that effective torque during the tests:

$$P_{pi-nom} = \frac{1}{m}\sum_{j=1}^{m} P_{pi-j} \tag{A.2-8}$$

where j is the test number, $P_{pi\text{-}j}$ is the initial preload obtained in the j-th test, and m is the number of tests. The nominal nut factor is then calculated by rearranging Eq. (A.2-1):

$$K_{nom} = \frac{T}{DP_{pi-nom}} \tag{A.2-9}$$

Section 6.1.d herein says that, when using torque control and without applicable data showing otherwise, the preload variation, Γ, for use in Eqs. (A.2-2) and (A.2-3) (in all analyses except separation analysis of separation-critical joints) shall be 25 percent for fasteners lubricated at assembly and 35 percent for non-lubricated and as-received fasteners[1]. Section 6.1.d goes on to say "*If the actual extreme variation found in the tests performed per paragraph 6.1.b exceeds the Γ values stated above, then the actual extreme variation should be used or methods for reducing the preload variation should be pursued.*" The "actual extreme variation" referred to here is calculated from test data as two values, $\Gamma_{a\text{-}max}$ and $\Gamma_{a\text{-}min}$:

$$\Gamma_{a-\max} = \frac{P_{pi-\max a}}{P_{pi-nom}} - 1 \tag{A.2-10}$$

or

$$\Gamma_{a-\min} = 1 - \frac{P_{pi-\min a}}{P_{pi-nom}} \tag{A.2-11}$$

where $P_{pi\text{-}maxa}$ and $P_{pi\text{-}mina}$ are the actual maximum and minimum initial preloads measured in the torque-preload tests. Eq. (A.2-10) applies to the establishment of Γ when calculating maximum preload, and Eq. (A.2-11) applies when calculating minimum preload. When calculating maximum preload, Γ is then the greater of $\Gamma_{a\text{-}max}$ and the default value (0.25 lubricated, 0.35 non-lubricated). When calculating minimum preload, Γ is then the greater of $\Gamma_{a\text{-}min}$ and the default value (0.25 lubricated, 0.35 non-lubricated).

Per section 6.1.c, calculation of minimum initial preload for separation analysis of separation-critical joints requires Γ to be derived from test data as the value determined to be at a statistical basis of 90 percent probability and 95 percent confidence (two-sided distribution). This value, $\Gamma_{90/95}$, may also be used per section 6.1.d to calculate maximum preload or to calculate minimum preload for joints that are not separation-critical. To determine the $\Gamma_{90/95}$ value, first calculate the unbiased sample standard deviation, σ_{pi}, from the collected sample of initial preloads for a given torque:

[1] Historic NASA criteria have specified that when using torque control of preload, lubricated bolts have a preload uncertainty (variation), Γ, of ±25 percent from a typical (nominal or mean) value, and non-lubricated bolts have a preload uncertainty of ±35 percent from a typical value. It historically has been *assumed* that these preload uncertainties envelop some large, but undefined, percentage of the probability distribution.

$$\sigma_{pi} = \sqrt{\frac{1}{m-1}\sum_{j=1}^{m}\left(P_{pi-j} - P_{pi-nom}\right)^2} \qquad \text{(A.2-12)}$$

The preload variation at 90 percent probability and 95 percent confidence is then

$$\Gamma_{90/95} = \frac{s\sigma_{pi}}{P_{pi-nom}} \qquad \text{(A.2-13)}$$

where *s* is the number of standard deviations from the mean applicable to 90 percent probability and 95 percent confidence; *s* varies with sample size, *m*, as given in table 2, Two-Sided 90/95 Tolerance Limit Factors for a Normal Distribution (Odeh and Owen, 1980).

Table 2—Two-Sided 90/95 Tolerance Limit Factors for a Normal Distribution (Odeh and Owen, 1980)

m	*s*	*m*	*s*	*m*	*s*	*m*	*s*
2	31.092	12	2.670	22	2.272	40	2.055
3	8.306	13	2.601	23	2.251	45	2.024
4	5.368	14	2.542	24	2.232	50	1.999
5	4.291	15	2.492	25	2.215	55	1.978
6	3.733	16	2.449	26	2.199	60	1.960
7	3.390	17	2.410	27	2.184	65	1.944
8	3.156	18	2.376	28	2.170	70	1.931
9	2.986	19	2.346	29	2.157	80	1.908
10	2.856	20	2.319	30	2.145	90	1.890
11	2.754	21	2.294	35	2.094	100	1.875

As an example, consider a torque-preload test of 10 sets of fastening system hardware, with each cycled 3 times in the as-received condition. The effective torque, *T*, is 450 in·lb, and the nominal bolt diameter, *D*, is 0.375 in. The measured preloads are listed in table 3, Sample Data from 30 Torque-Tension Tests (Actual Test Data).

Table 3—Sample Data from 30 Torque-Tension Tests (Actual Test Data)

	Bolt 1	Bolt 2	Bolt 3	Bolt 4	Bolt 5	Bolt 6	Bolt 7	Bolt 8	Bolt 9	Bolt 10
Test 1	7760	4010	3587	4635	4920	3797	4462	4940	6220	4780
Test 2	7229	4310	3900	4322	4475	3812	4132	4219	6100	4685
Test 3	6080	4291	3054	4804	3910	3303	4090	4550	6062	5180

From Eqs. (A.2-8) through (A.2-13), with n = 30 and s = 2.145,

$$P_{pi\text{-}nom} = 4{,}721 \text{ lb}$$

$$K_{nom} = \frac{450}{0.375(4721)} = 0.254$$

$$\Gamma_{a-\max} = \frac{7760}{4721} - 1 = 0.644$$

$$\Gamma_{a-\min} = 1 - \frac{3054}{4721} = 0.353$$

To calculate $P_{pi\text{-}max}$, Γ = 0.644 (higher of $\Gamma_{a\text{-}max}$ and 0.35).

Alternatively, it is acceptable to use $\Gamma = \Gamma_{90/95}$, calculated as follows:

$$\sigma_{pi} = \sqrt{\frac{1}{m-1}\sum_{j=1}^{m}\left(P_{pi-j} - P_{pi-nom}\right)^2} = 1{,}097\ lb$$

$$\Gamma_{90/95} = \frac{2.145(1097)}{4721} = 0.498$$

To calculate $P_{pi\text{-}min}$, if the joint is not separation-critical, Γ = 0.353 (higher of $\Gamma_{a\text{-}min}$ and 0.35), which is less penalizing than the alternative of using $\Gamma = \Gamma_{90/95}$ = 0.498.

Note that, in the above example, there is unusually high variation in the initial preloads obtained for the given torque. In such a case, it is advisable to investigate the cause of such high variation in Γ and take appropriate steps such as modifying the installation process by use of lubrication or selecting different fastening system hardware.

A.2.1 Rationale for Eqs. (6-5b) and (A.2-3b)

When performing slip analysis, the concern related to preload is not the variation in preload for a single fastener, it is the variation in total preload for the joint. Because the preload for any single bolt in the joint has a probability distribution for a given torque, then the total preload for the bolt pattern has a probability distribution. The total preload equals the mean preload for the bolts in the pattern multiplied by the number of fasteners, so the probability distribution for total preload is the same as the probability distribution for the mean preload for the bolts in the pattern. This distribution is referred to as the *distribution of the means*, and it is characterized using the *Central Limit Theorem* as having a mean equal to the mean of the population (the mean of the distribution for any single bolt) and a standard deviation equal to the standard deviation of the population divided by the square root of the number of bolts in the bolt pattern. So, for a joint containing a number of bolts, n_f, the probability distribution for the total preload for the n-bolt pattern has a mean, $\mu_{nbp} = \mu$, and a standard deviation, $\sigma_{nbp} = \sigma/\sqrt{n_f}$. It can be assumed that the distribution of the means is normal.[2]

When performing separation analysis, part of the concern is separation of the entire joint, and part of the concern is local separation near a single fastener. The minimum preload used for separation analysis of separation-critical joints must be derived by test at the statistical probability and confidence specified in section 6.1. For joints that are not separation-critical, it is acknowledged that the consequence of local separation in a joint with multiple fasteners is not as severe as the consequence of rupture or detrimental yielding, so it is acceptable to base the separation analysis on a minimum preload that is progressively less conservative as the number of fasteners in the pattern increases. Although the *Central Limit Theorem* does not apply to preload at a single fastener (it applies only to total preload for a bolt pattern and thus separation analysis for the overall joint), it is acceptable for separation analysis at the highest-loaded fastener in a non-separation-critical joint to use a minimum preload calculated the same way in which it is calculated for slip analysis, as noted above and specified in section 6.1.
Separation under cyclic applied load, whether for the entire joint or local to a fastener, is undesirable as it causes more fatigue damage in the fastener. Fatigue analysis, however, must account for the full range in preload with uncertainty as specified in section 6.1. Thus, for non-separation-critical joints with multiple fasteners, fatigue analysis must account for potential separation near the highest-loaded fastener in a pattern even if a positive margin of safety for separation is shown.

---

[2] The *Central Limit Theorem* states that the distribution of the means approaches a normal distribution as the size of the sample used to calculate the mean approaches infinity, even if the distribution of the population is non-normal. For practical purposes, the distribution of the means is near normal when the sample size is greater than or equal to 30 (Spiegel, 1961, p. 144). The distribution of the means "tends to be normally distributed even for relatively small *n* [sample size] as long as no single element or small group of elements has a dominating variance and the element distributions do not deviate extremely from a normal distribution" (Hahn and Shapiro, 1967, p. 73). The standard deviation of the distribution of the means is also known as the "standard error."

A.3 Short-Term Relaxation of Preload (Supplement to Section 6.1)

Table 1 provides an adjustment for short-term relaxation of preload. Such relaxation is a well-documented effect attributed to embedment, which results from localized yielding at interfaces. When calculating minimum preload, it is acceptable in most cases to reduce the minimum initial preload by 5 percent for short-term relaxation. Higher losses of preload have been reported as a result of short-term relaxation due to stack up of washers, smaller-diameter washers, low-yield strength materials, etc. In such cases, even for all-metal threaded fastening systems, larger reductions may be necessary.

A.4 Use of a Load-Introduction Factor (Supplement to Section 6.2.1.2)

The load-introduction factor, n, as used in Eq. (6-7), varies between 0 and 1 depending on the joint geometry and the point of application of the external tensile load as outlined in Verein Deutscher Ingenieure (VDI) 2230, "Systematic Calculation of High Duty Bolted Joints, Joints with One Cylindrical Bolt," and "Effect of Separating Load Eccentricity on the Clamp Load Loss in a Bolted Joint Using a Strain Hardening Model" (Nassar, S.A., and others, ASME Journal of Pressure Vessel Technology manuscript accepted for publication #PVT-09-1060). As external load is applied to the joint and the tensile load in the fastener increases, the stress and deformation distribution in the clamped materials becomes highly uneven and more complex. The bending stiffness of the clamped plates significantly increases the effective joint stiffness from its nominal value k_c. As external load is applied, the bolt load increases and the corresponding joint compressive load decreases between mating parts resulting in a reduction in k_c. The materials directly under the washer and near the washer surface compress further to introduce additional load into the bolt. The stiffness of this material is in the bolt's load path, as another spring in series with bolt stiffness. For many applications, n is significantly lower than 0.5. Setting $n = 1$ in Eq. (6-7) results in a significantly higher calculated increase in bolt load than actually occurs.

Historically, National Space Transportation System (NSTS) 08307, Criteria for Preloaded Bolts, the load-introduction factor has been defined solely based on loading-plane locations but ignores the effect of dimension b, as shown in figure 4, Dimensions Used to Calculate Load-Introduction Factor. Note that this approach is conservative for bolt analysis but its use would be unsafe for separation analysis. With such an approach, the load-introduction factor is defined as

$$n = \frac{L_{lp}}{L} \tag{A.4-1}$$

where L and L_{lp} are as defined in figure 4 for three types of joints. The dimension L_{lp} is the distance between the planes at which an applied tensile load is introduced to the joint. Each plane is at the middle of each clamped member.

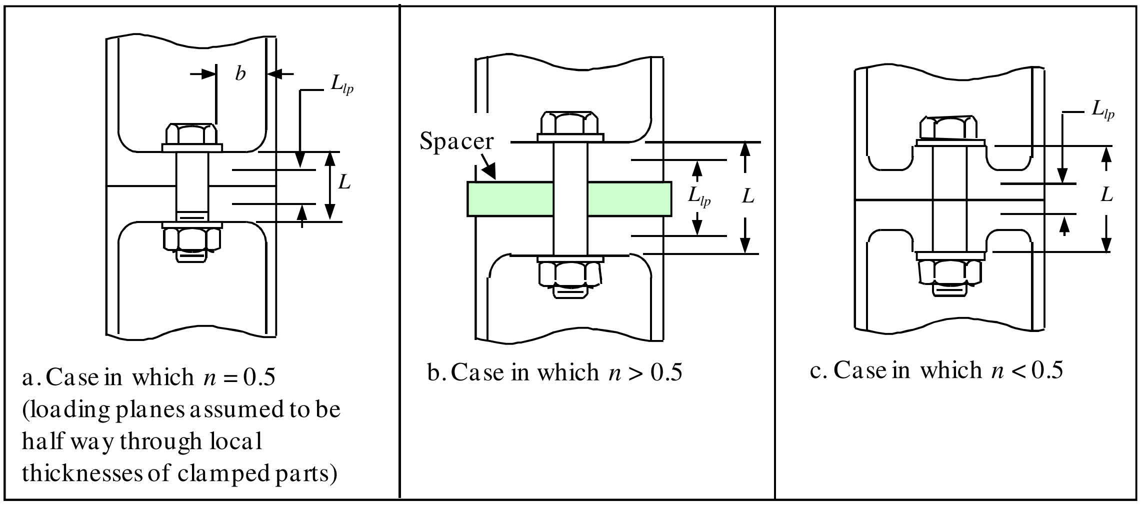

Figure 4—Dimensions Used to Calculate Load-Introduction Factor

A.5 Bolt Analysis: Separation Before Rupture (Supplement to Section 6.2.1)

Whether separation would occur before rupture can be determined based on test, analysis, or the logic flow in figure 5, Determining Whether a Joint Separates Before Rupture when Loaded Solely in Tension. If analysis is used, it should be based on the assumption of maximum preload. (See section 6.2.1.2.)

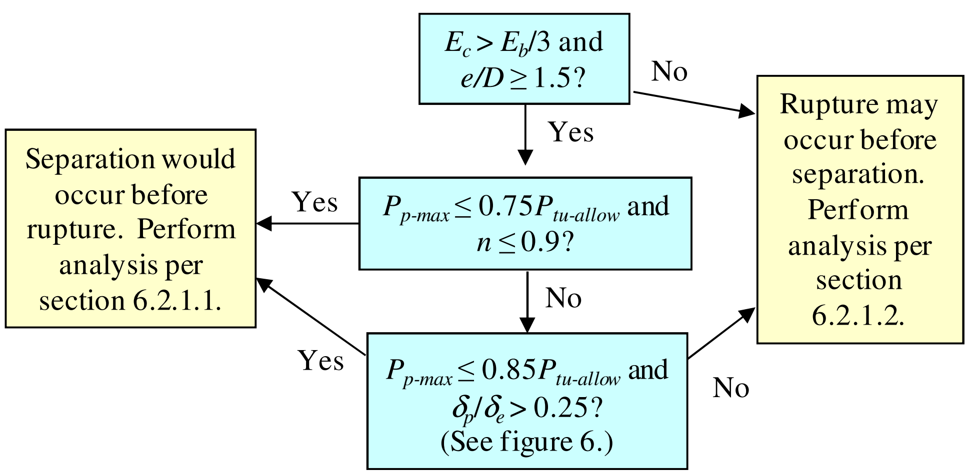

Figure 5—Determining Whether a Joint Separates Before Rupture When Loaded Solely in Tension

In figure 5, E_c is the lowest elastic modulus of clamped parts excluding washers, E_b is the elastic modulus of the bolt material, e is the minimum edge distance (measured from center of hole to closest edge of any clamped part other than washers), δ_p is the plastic deformation of the fastening system at rupture (see figure 6, How to Quantify Ductility of a Fastening System from a Tension Test), and δ_e is the elastic deformation of the fastening system at rupture (see figure 6).

Figure 6 shows a typical load-deflection curve for a bolt tested to failure in tension. The ratio of δ_e to δ_p depends on the fastener and the mating internally threaded part. To determine δ_e and δ_p, the bolt should be tested with an internally threaded part that is of the same specification as is used in the flight assembly in case the hardware combination leads to thread stripping as the mode of failure.

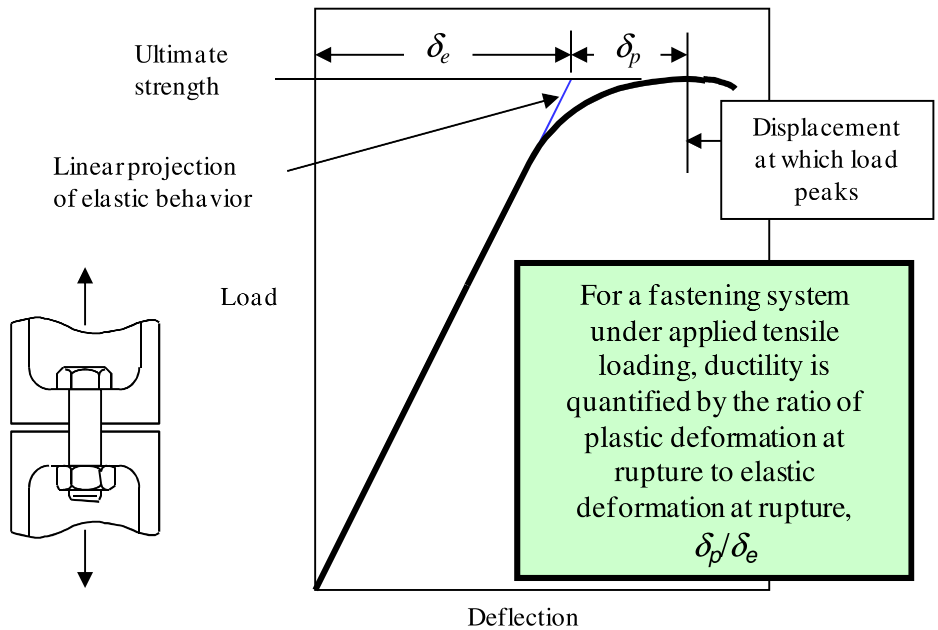

Figure 6—How to Quantify Ductility of a Fastening System from a Tension Test

Figure 5 allows engineers to quickly recognize designs that clearly would separate under applied tensile load before the fastening system ruptures. If the joint would separate before rupture, the tensile bolt load at rupture would be equal to the tensile applied load and there is no need to include preload or to calculate stiffness of the bolt and stiffness of the clamped material when assessing ultimate strength.

The criteria in figure 5 were derived from analysis of a joint with a single A-286 bolt and clamped parts made of aluminum alloy. For this combination, the elastic modulus of the clamped material, E_c, is approximately one-third of the elastic modulus of the bolt material, E_b. For any material combination in which E_c is significantly less than $E_b/3$, the total load in a preloaded bolt increases by a larger percentage of applied tensile load than is calculated in this study and the conclusions made here regarding separation before rupture may not apply.

The joint assessed in this study is shown in figure 7, Preloaded Joint in Which the Bolt Load Increases by a Relatively High Percentage of Applied Tensile Load. The geometry and dimensions were selected to maximize $n\phi$, the percentage of applied load by which the bolt load increases prior to separation, with the constraints that $e/D \geq 1.5$ and $E_c > E_b/3$. This joint has a large load-introduction factor, n, which can be calculated as L_{lp}/L, as shown in figure 4. The joint is shown without washers, as the presence of washers decreases $n\phi$.

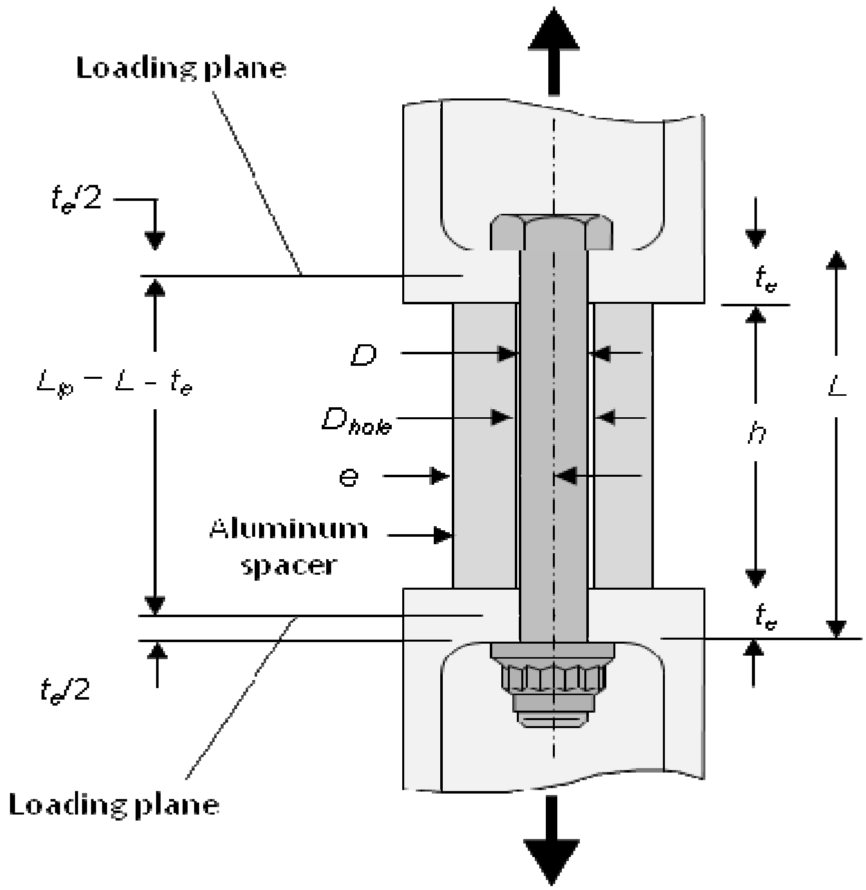

Figure 7—Preloaded Joint in Which the Bolt Load Increases by a Relatively High Percentage of Applied Tensile Load

Nonlinear finite element analysis of the joint illustrated in figure 7—with $E_c = E_b/3$, $e/D = 1.5$, and $n = L_{lp} / L = 0.9$—shows that $n\phi < 0.25$.

The conclusion is that, for any practical design with $E_c < E_b/3$, $e/D \geq 1.5$, and $n \leq 0.9$, the bolt load increases by no more than 25 percent of the applied tensile load prior to separation.

As shown in figure 8, Separation Before Rupture for a Brittle Threaded Fastening System, when the bolt load increases by 25 percent of the applied tensile load, the joint would separate prior to rupture as long as the preload is not greater than 75 percent of the bolt's ultimate tensile strength. This is the case even if the fastening system exhibits perfectly brittle failure.

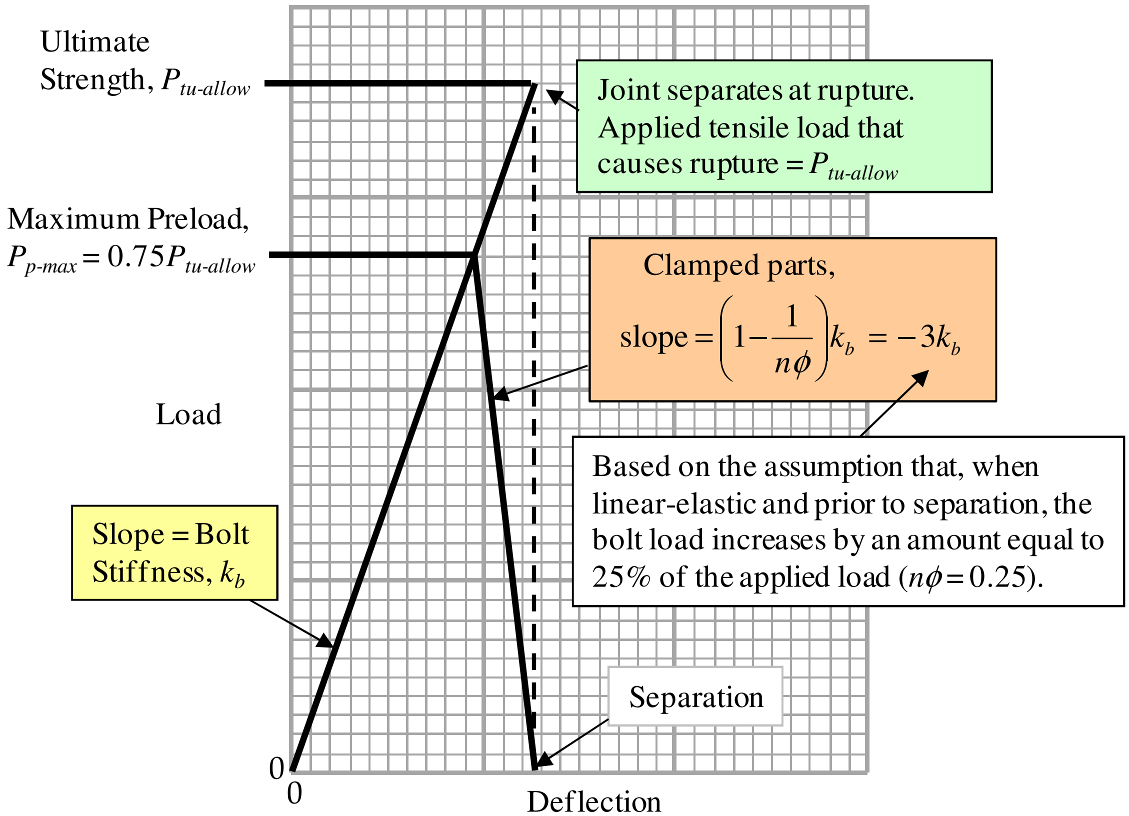

Figure 8—Separation Before Rupture for a Brittle Threaded Fastening System

Fastening-system hardware made of metal alloys exhibit some degree of ductility prior to rupture. Ductility causes a joint to separate at a lower applied load than would be predicted with linear theory. Consider a joint in which the bolt load increases by 25 percent of the applied tensile load when linear elastic, with preload at 85 percent of the bolt's ultimate tensile strength. For such a joint, separation would occur prior to rupture if the plastic deformation, δ_p, were at least 13.3 percent of the elastic deformation, δ_e, as shown in figure 9, Separation Before Rupture for a Ductile Threaded Fastening System.

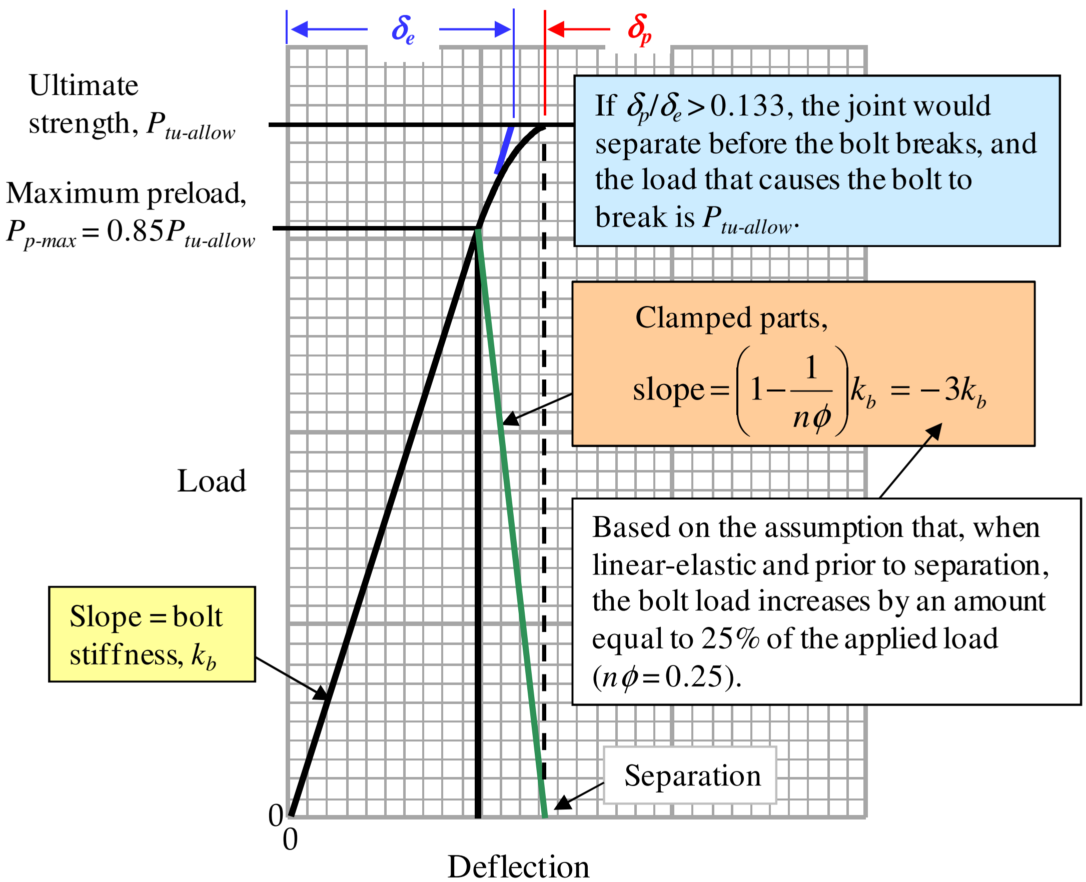

Figure 9—Separation Before Rupture for a Ductile Threaded Fastening System

Figure 5 states the criterion that $\delta_p/\delta_e > 0.25$ in order to draw the conclusion that a joint would separate before the fastening system ruptures when the maximum preload is no greater than 85 percent of the fastening system's allowable ultimate tensile load. The difference between $\delta_p/\delta_e = 0.25$ and $\delta_p/\delta_e = 0.133$ (justified in figure 9) is intentional to account for potential variation in plastic strain between the actual flight hardware and the hardware tested to rupture when determining δ_p/δ_e.

A.6 Ultimate Margin of Safety for Tensile Loading with Linear Theory (Supplement to Section 6.2.1.2)

Figures 10 through 12 illustrate the linear mechanics of a preloaded joint under applied tensile load, in support of section 6.2.1.2.

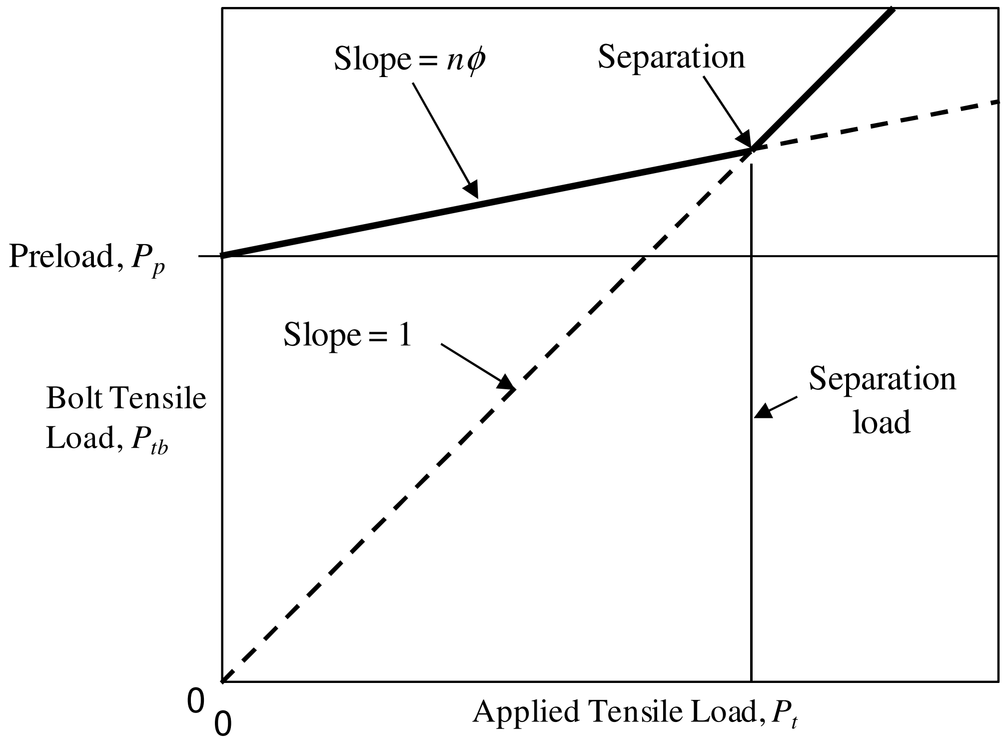

Figure 10—Bolt Tensile Load versus Applied Tensile Load

Eq. (6-7) applies until either the bolt ruptures or the joint separates. If the joint separates, the bolt load equals the applied load (see figure 10, Bolt Tensile Load versus Applied Tensile Load).

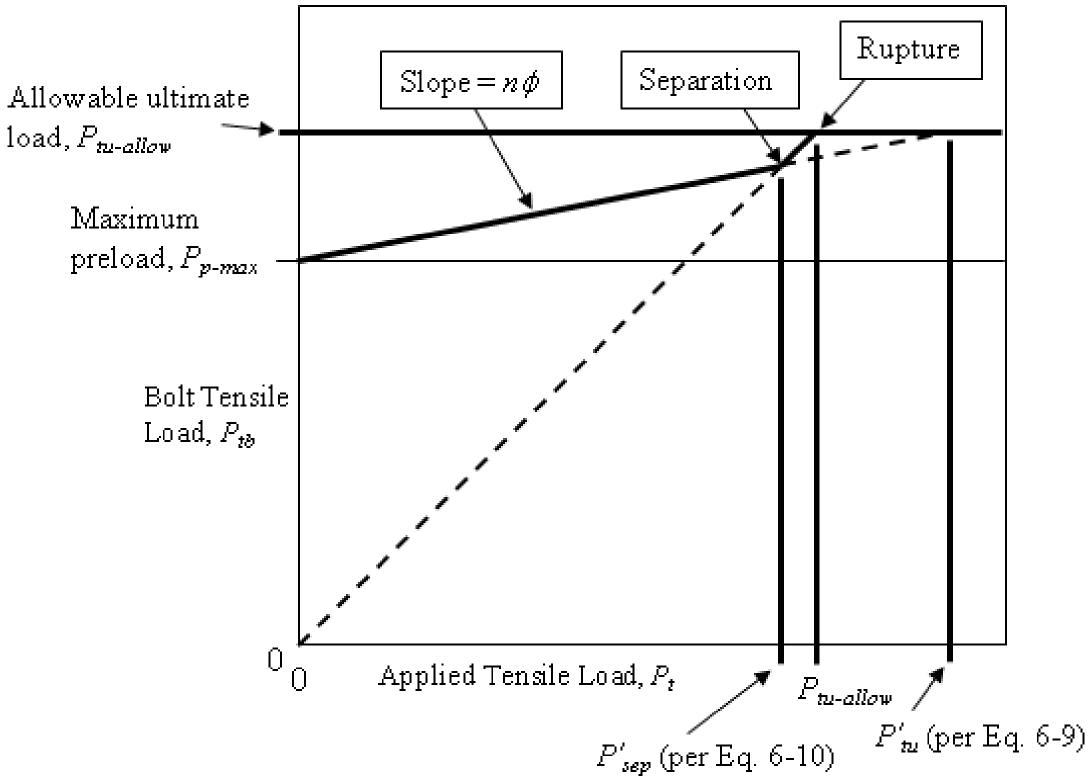

Figure 11—Separation Before Rupture with Linear Theory

If separation would occur before rupture, Eq. (6-9) does not apply and the margin of safety is given by Eq. (6-6) (see figure 11, Separation Before Rupture with Linear Theory).

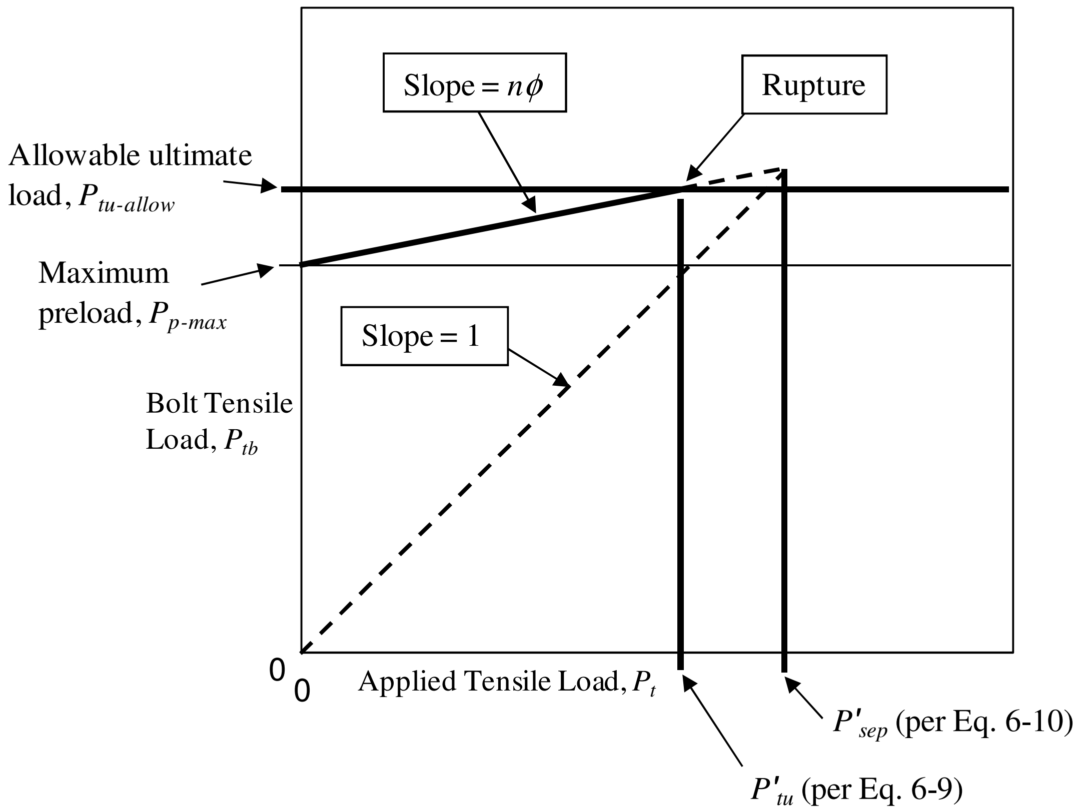

Figure 12—Rupture Before Separation with Linear Theory

If rupture would occur before separation, the margin of safety is given by Eq. (6-11) (see figure 12, Rupture Before Separation with Linear Theory).

A.7 Omission of Preload in Shear and Interaction Analyses (Supplement to Section 6.2.2)

Any fastener loaded in shear should have a full diameter body and the bolt grip should be selected to ensure the full diameter body is in the shear plane. Specified or allowable shear loads for such fasteners are based on shear tests in which the full diameter body is in the shear plane.

When a preloaded fastener is subjected to shear loading, tensile and shear stresses exist simultaneously in the shank. However, if the shear load is increased until the fastener ruptures, the fastener material first yields and the preload relaxes. As a result, the preload does not contribute to failure.

NASA tests performed in 2009 substantiated the above hypothesis for preloaded fasteners loaded in shear only with no threads in the shear plane. Double-shear specimens with steel plates and A-286 3/8-24 (NAS1956C28) bolts were tested to failure, with the full diameter body in the shear plane. Five bolts were tested without preload and five were tested with preload, all fasteners were from the same procurement lot. Those tested with preload were lubricated with Castrol Braycote® 602EF grease and then torqued to 122 N-m (90 ft-lb), which caused yielding in the fasteners. Prior to assembly, to minimize friction as a shear-load path, the mating surfaces of the steel plates were lubricated with molybdenum disulfide dry-film lubricant. A fresh set of steel plates was used for each test. The average rupture loads for the two configurations tested showed a difference of less than 2 percent.

Similar results were found in tests of A-286 3/8-24 (NAS1956C14) fasteners under simultaneous tensile and shear loading, conducted in 2010 at MSFC. These tests were performed with and without threads in the shear plane. The tests were performed with and without preload; the tests showed preload had an insignificant effect on the failure loads.

The conclusion from these tests is that preload can be disregarded when assessing ultimate strength for a fastener under shear loading or under simultaneously applied tensile and shear loading.

A.8 Theoretical Treatment of Interaction Equations (Supplement to Section 6.2.3)

The interaction equations for ultimate-strength analysis in section 6.2.3 are based on empirical test data rather than a theoretical derivation. However, similar theoretical interaction equations can be derived when shear, tension, and bending across a cross-section are considered. This section presents a classical treatment for interaction of shear, tension, and bending. The equations in this section are not intended to replace the interaction equations in section 6.2.3.

Consider a fastener whose shear plane is in the x-z plane and whose tensile direction is along the z-axis. The fastener is loaded in shear only, developing a shear stress τ. The stress tensor associated with the state of stress is given by

$$\sigma = \begin{bmatrix} 0 & 0 & \tau \\ 0 & 0 & 0 \\ \tau & 0 & 0 \end{bmatrix} \tag{A.8-1}$$

where σ is used to denote the stress tensor (not to be mistaken with a single stress component σ). The principal stresses, given by the eigen values of the stress tensor, are

$$S_1 = 0,\ S_2 = \tau,\ S_3 = -\tau \tag{A.8-2}$$

Using the von Mises yield criterion, one can establish a relationship between the yield stress in tension, F_{ty}, and the yield stress in shear, F_{sy}. The von Mises criterion is

$$(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2 \le 2F_{ty}^2 \tag{A.8-3}$$

Using Eqs. (A.8-2) and (A.8-3) yields

$$F_{sy} = \frac{F_{ty}}{\sqrt{3}} \tag{A.8-4}$$

It should be noted that F_{sy} is not a typically encountered material property. The usual property encountered is the ultimate shear stress F_{su}. Often Eq. (A.8-4) is assumed to hold at both yield and ultimate conditions, such that

$$F_{su} = \frac{F_{tu}}{\sqrt{3}} \tag{A.8-5}$$

While Eq. (A.8-5) does not have a firm theoretical basis to it, a survey of commonly used materials for fasteners shows that the relationship is typically affirmed.

Now consider a fastener with a shear τ and tension σ at a point in the cross section. The stress tensor for this state of stress is given by

$$\sigma = \begin{bmatrix} 0 & 0 & \tau \\ 0 & 0 & 0 \\ \tau & 0 & \sigma \end{bmatrix} \tag{A.8-6}$$

and the principal stresses are given by

$$S_1 = \frac{\sigma + \sqrt{\sigma^2 + 4\tau^2}}{2}, \quad S_2 = \frac{\sigma - \sqrt{\sigma^2 + 4\tau^2}}{2}, \quad S_3 = 0 \tag{A.8-7}$$

Using Eqs. (A.8-3), (A.8-4), and (A.8-7) yields

$$\left(\frac{\sigma}{F_{ty}}\right)^2 + \left(\frac{\tau}{F_{sy}}\right)^2 \leq 1 \tag{A.8-8}$$

If Eq. (A. 8-8) is extrapolated to the ultimate stress condition (which is theoretically incorrect); bending stress is accounted for such that

$$\sigma = \frac{P_{tu}}{A} + f_{bu} \tag{A.8-9}$$

where A is the cross-sectional area; and using $P_{su} = \tau A$, $P_{su\text{-}allow} = F_{su}A$, and $P_{tu\text{-}allow} = F_{tu}A$; one has

$$\left(\frac{P_{su}}{P_{su-allow}}\right)^2 + \left(\frac{P_{tu}}{P_{tu-allow}} + \frac{f_{bu}}{F_{tu}}\right)^2 \leq 1 \tag{A.8-10}$$

Eq. (A.8-10) gives an approximation at the ultimate stress condition for a fastener loaded in shear, bending, and tension in the same cross section.

If the bolt is allowed to yield at ultimate design load, one must accommodate the potential for additional load. For simplicity, the case of combined tension and bending for a circular cross section with radius r is examined. It is assumed to be an elastic, perfectly plastic material with tensile yield stress of F_{ty}. If a pure axial load is applied such that the entire section has yielded, the fully plastic axial force N_p can be computed. Likewise, if the cross section has a pure moment applied such that the upper half is fully yielded to F_{ty} and the lower half is fully yielded to $-F_{ty}$, the fully plastic moment, M_p, can be computed. These values are computed by the integrals

$$N_p = 2\int_{y=0}^{y=r} \int_{x=-(r^2-y^2)^{1/2}}^{x=(r^2-y^2)^{1/2}} F_{ty}\,dxdy = \pi F_{ty} r^2 \tag{A.8-11}$$

and

$$M_p = 2\int_{y=0}^{y=r} \int_{x=-(r^2-y^2)^{1/2}}^{x=(r^2-y^2)^{1/2}} F_{ty}\,ydxdy = \frac{4}{3} F_{ty} r^3 \tag{A.8-12}$$

Now consider a section with combined tension and bending at collapse such that the stress distribution is as shown in figure 13, Stress Distribution at Collapse due to Combined Bending and Tension.

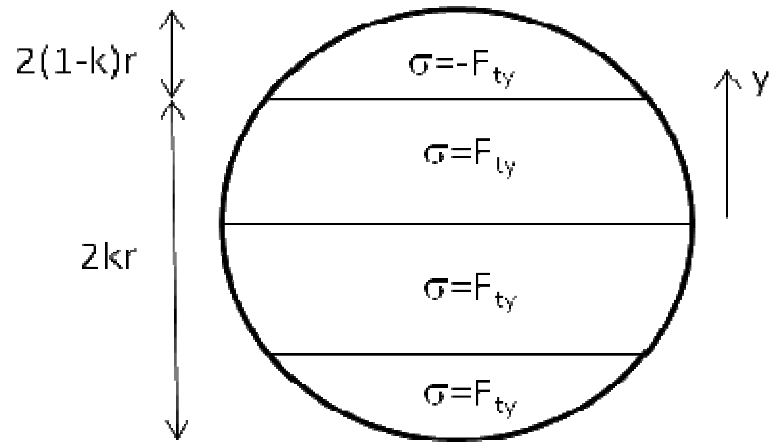

Figure 13—Stress Distribution at Collapse Due to Combined Bending and Tension

One can compute the axial force and moment due to the stress distribution shown in figure 13 as

$$N = 2\int_{y=0}^{y=-r+2kr} \int_{x=-\left(r^2-y^2\right)^{1/2}}^{x=\left(r^2-y^2\right)^{1/2}} F_{ty}\, y dx dy = 2F_{ty}\left[2r^2\left(2k-1\right)\sqrt{k\left(1-k\right)} + r^2 \sin^{-1}\left(2k-1\right)\right] \tag{A.8-13}$$

and

$$M = 2\int_{y=-r+2kr}^{y=r} \int_{x=-\left(r^2-y^2\right)^{1/2}}^{x=\left(r^2-y^2\right)^{1/2}} F_{ty}\, y dx dy = \frac{32}{3} F_{ty} r^3 \left[k\left(1-k\right)\right]^{3/2} \tag{A.8-14}$$

If either Eq. (A.8-13) or Eq. (A.8-14) is solved for *k*, an interaction equation could be written relating the allowable axial load and moment at collapse. The closed-form solution is shown by Eq. (A.8-15) (see Stronge & Yu, "Dynamic Models for Structural Plasticity").

$$\left\{\cos\left[\frac{\pi N}{2N_p} - \left(\frac{M}{M_p}\right)^{1/3} \sqrt{1-\left(\frac{M}{M_p}\right)^{2/3}}\right]\right\} - \left|\frac{M}{M_p}\right| = 0 \tag{A.8-15}$$

By comparison, if a rectangular cross section is considered, a simpler expression relating moment and axial load at collapse is found such that

$$\frac{M}{M_p}+\left(\frac{N}{N_p}\right)^2=1 \tag{A.8-16}$$

A comparison between Eqs. (A.8-15) and (A.8-16) is shown in figure 14, Comparison of Interaction Between Circular and Rectangular Cross Sections at Collapse. As can be seen from the figure, there is little difference between the two closed-form solutions. For simplicity, it can be assumed that the interaction equation given by Eq. (A.8-16) can be applied to circular cross sections indicative of fasteners.

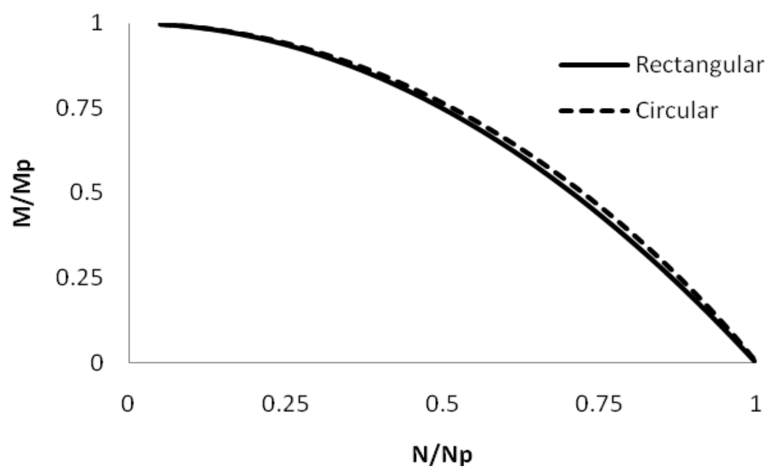

Figure 14—Comparison of Interaction Between Circular and Rectangular Cross Sections at Collapse

To account for shear, a rectangular cross section is used. This simplifies the problem and allows for a simple closed-form solution. For a rectangular cross section, the fully plastic axial force and fully plastic moment are given by

$$N_p = F_{ty}bh \qquad M_p = \frac{F_{ty}bh^2}{4} \tag{A.8-17}$$

From Eq. (A.8-8) the tension-carrying capability of the cross section is reduced due to an applied shear. Eq. (A.8-8) can be rewritten in terms of shear Q and fully plastic shear Q_p such that

$$\left(\frac{\hat{F}_{ty}}{F_{ty}}\right)^2=1-\left(\frac{Q}{Q_p}\right)^2 \quad \text{where} \quad \frac{Q}{Q_p}=\frac{\tau A}{F_{sy}A} \tag{A.8-18}$$

and $\hat{F}_{ty}$ is the reduced allowable load due to the shear load. Under combined shear, tension, and bending, the peak tensile stress is limited to $\hat{F}_{ty}$. Using the same methodology that led to Eqs. (A.8-13) and (A.8-14), the axial force and moment acting on the cross section due to $\hat{F}_{ty}$ are

$$N = \hat{F}_{ty} bh(2k-1) \qquad M = \hat{F}_{ty} bh^2(1-k)k \tag{A.8-19}$$

Solving for *k* and combining Eq. (A.8-17) to (A.8-19) at collapse yields

$$\left(\frac{Q}{Q_p}\right)^2 + \left(\frac{N}{N_p}\right)^2 + \frac{M}{M_p}\sqrt{1-\left(\frac{Q}{Q_p}\right)^2} = 1 \tag{A.8-20}$$

Eq. (A.8-20) can be simplified and rewritten as

$$\left(\frac{P_{su}}{P_{su-allow}}\right)^2 + \left(\frac{P_{tu}}{P_{tu-allow}}\right)^2 + \frac{f_{bu}}{F_{bu}} \le 1 \tag{A.8-21}$$

In Eq. (A.8-21), the more theoretically appropriate F_{tu} has been replaced with the term F_{bu}. It is not uncommon for practitioners to implement a higher allowable based on plastic bending rather than the tensile allowable. Doing so is less conservative, though this approach is accepted due to the conservatism that led to the development of the interaction equation, namely the use of a linear elastic, perfectly plastic material response.

A.9 Determining if Fastener Yielding is Detrimental for Separation or Joint Slip (Supplement to Section 6.3)

In a typical joint with preload generated by torque control, fastener yielding under combination of preload and applied load is not detrimental for separation. Rationale:

a. As shown in figure 15, Effect of Fastener Yielding on the Separation Load, fastener yielding causes the separation load to be lower than is predicted based on linearly calculated bolt stiffness and clamp stiffness. However, section 6.5 requires that the design separation load not exceed minimum preload, which is less than the actual separation load shown in the figure. Thus, fastener yielding that occurs under a given applied load does not affect separation analysis for that load.

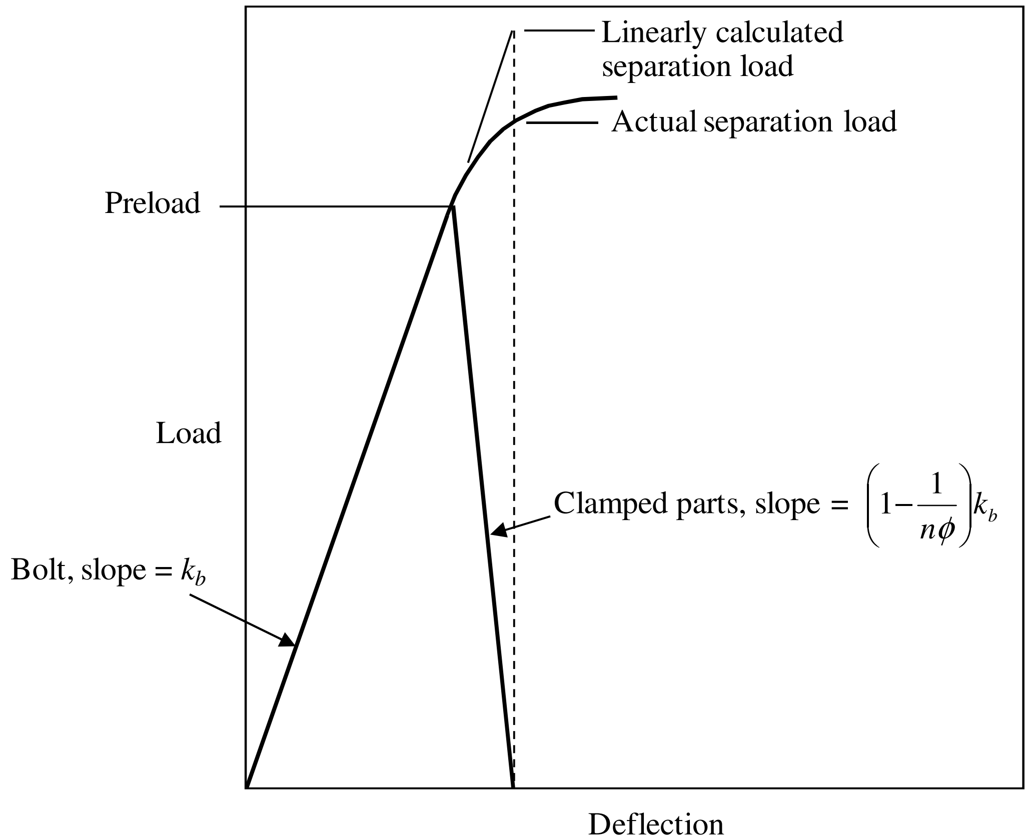

Figure 15—Effect of Fastener Yielding on the Separation Load

b. Figure 16, Loss of Preload Resulting from Fastener Yielding, shows an example of the loss in preload associated with yielding under an applied load equal to the separation load. Because of strain hardening, reapplication of the same load does not cause further yielding, and the separation load is unchanged. Thus, fastener yielding that occurs under an applied load that does not cause separation is not detrimental for separation under repeated loading.

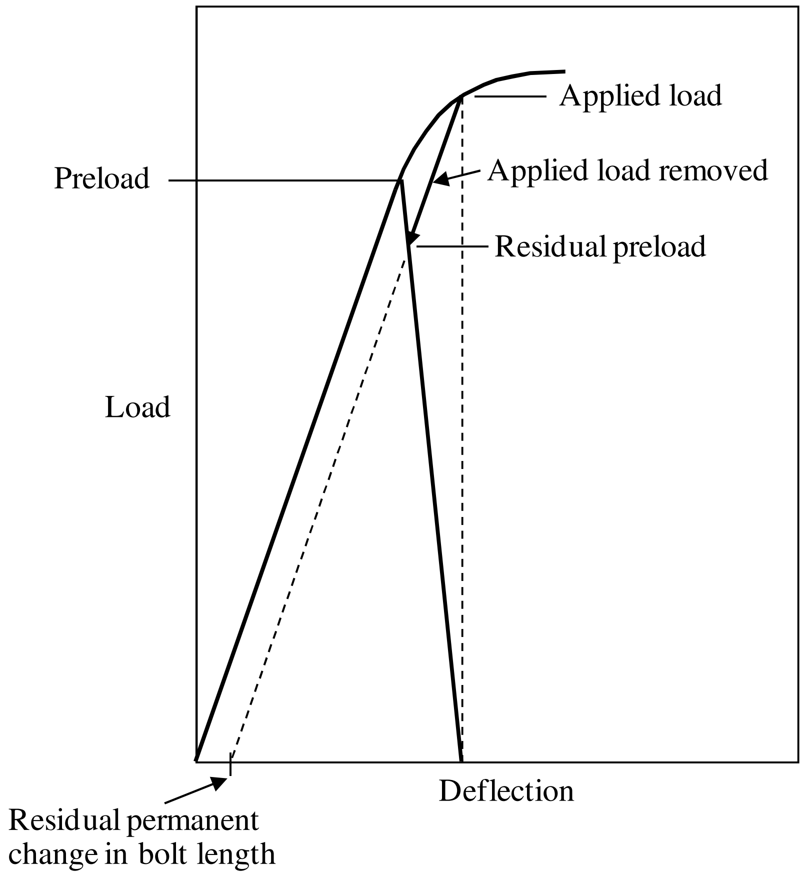

Figure 16—Loss of Preload Resulting from Fastener Yielding

c. Figure 17, Fastener Yielding Under an Applied Load that Exceeds the Separation Load, shows a situation in which yielding occurs under an applied load that separates the joint. (This situation is possible if the yield factor of safety is greater than the separation factor of safety, in which case the design criteria in section 6 allow the design yield load to exceed the minimum preload.) Once the applied load is removed, the residual preload is considerably less than the original preload, and the separation load also is below the original preload.

Such yielding can be detrimental for separation if the new separation load is below the minimum preload calculated with Eq. (6-5). This may be the case if preload is accurately controlled through a method such as ultrasonically measuring the bolt's change in length. In such a situation, a positive margin of safety for fastener yielding using Eq. (6-21) ensures the fastener will not yield after separation under the design yield load.

With torque control, there is a wide range between the maximum and minimum preloads calculated per section 6.1. Section 6.5 specifies a criterion that the design separation load not exceed the calculated minimum preload. If this criterion is satisfied, the design yield load, using typical yield factors of safety for flight hardware, would not exceed the maximum preload and would not cause separation. This premise can be verified by calculation of a yield margin of safety using Eq. (6-21), if desired.

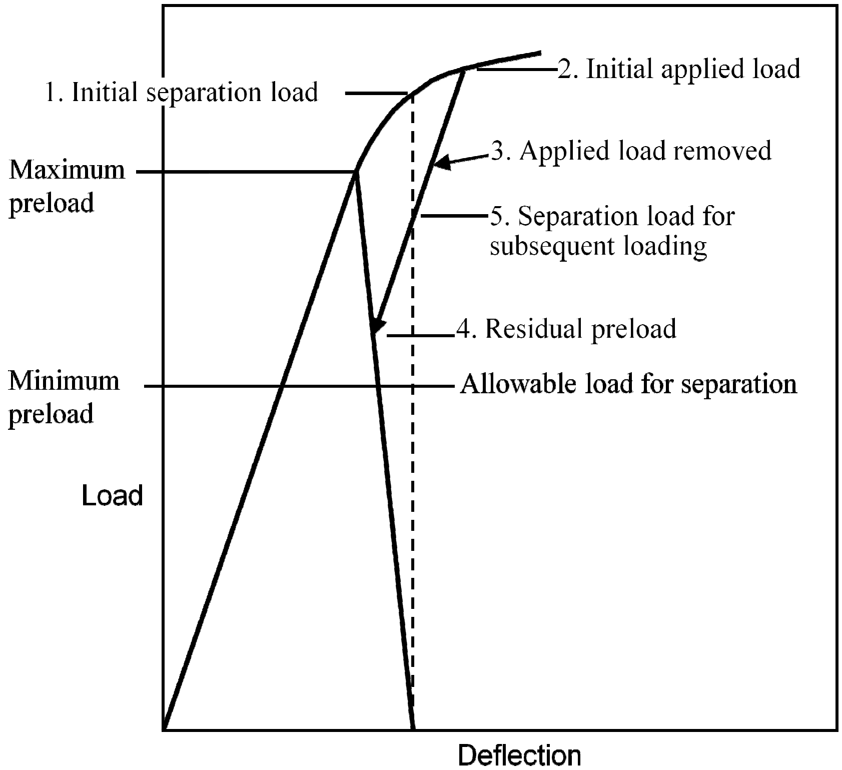

Figure 17—Fastener Yielding Under an Applied Load that Exceeds the Separation Load

A possible exception to the above conclusions arises if preload changes significantly under temperature extremes. For example, if preload increases with temperature and yielding occurs at the maximum design temperature, and then the joint later reaches its minimum design temperature, the residual preload may be inadequate to prevent separation under the co-existing design separation load. A recommended criterion for avoiding this situation is to ensure the maximum preload calculated with Eq. (6-4) does not exceed the fastener's allowable yield tensile load.

The guidance for determining whether fastener yielding is detrimental for separation also applies to the issue of whether fastener yielding is detrimental for joint slip. In addition, consideration must be given to the situation in which the fastener yields under one loading event, such as ground test or liftoff, followed by the design yield shear load in a later event, such as maximum air loads during launch. When relying on friction for an alignment-critical joint, it may be wise to assume fastener yielding is detrimental and satisfy the criteria given in section 6.3.

A.10 Margin of Safety for Joint Slip (Supplement to Section 6.4)

To be consistent with separation analysis (see Appendix A.11), slip analysis is based on the assumption that the clamp force in a preloaded joint is reduced by 100 percent of the applied tensile load. When a joint is concentrically loaded with shear (load passes through bolt-pattern centroid) and simultaneously loaded in tension, using fasteners of the same type and with the same nominal preload, the friction load, P_f, that can be developed is

$$P_f = \mu \left(n_f P_p - P_{t-joint} \right) \tag{A.10-1}$$

where $P_{t\text{-}joint}$ is the total applied tensile load acting on the joint.

Calculation of a margin of safety for slip—which must account for minimum preload, limit loads, and a factor of safety—is not straightforward in such a case. Based on the assumption that, if the applied tensile and shear loads were to increase, they would do so proportionally, one can solve for the proportionality constant, *a*, that causes a zero-margin condition:

$$\frac{\mu \left\lfloor n_f P_{p-min} - a \cdot FS \cdot P_{tL-joint} \right\rfloor}{a \cdot FS \cdot P_{sL-joint}} - 1 = 0 \tag{A.10-2}$$

where μ is the coefficient of friction, *FS* is the appropriate factor of safety for the failure mode affected by slip, and $P_{sL\text{-}joint}$ is the total limit shear load acting on the joint, and $P_{tL\text{-}joint}$ is the total limit tensile load acting on the joint. Solving for *a* gives

$$a = \frac{n_f \mu P_{p-min}}{FS \cdot \left(P_{sL-joint} + \mu P_{tl-joint} \right)} \tag{A.10-3}$$

Because *a* is the scaling factor applied to the external loads that causes a zero-margin condition, the margin of safety for slip is

$$MS_{slip} = a - 1 = \frac{n_f \mu P_{p-\min}}{FS \cdot \left(P_{sL-\text{joint}} + \mu P_{tL-\text{joint}} \right)} - 1 \tag{A.10-4}$$

When no external tensile load is applied, Eq. (A.10-4) reduces to

$$MS_{\text{slip}} = \frac{n_f \mu P_{p-\min}}{FS \cdot P_{sL-joint}} - 1 \tag{A.10-5}$$

Eqs. (A.10-4) and (A.10-5) only apply to a joint that is concentrically loaded in tension and shear, has equal nominal preload for all fasteners, and equivalent fastener sizes. When any of these conditions are violated, these equations cannot be used, and an analysis considering these effects must be performed. An example when Eqs. (A.10-4) and (A.10-5) would not apply is eccentrically loaded joint where the shear produces a resultant torque about the bolt-pattern centroid. A finite element analysis or test can be used to verify that the joint does not slip under such a condition. Another acceptable approach is to assess slip individually at each fastener location as follows:

$$MS_{slip} = \frac{\mu P_{p-\min}}{FF \cdot FS \cdot \left(P_{sL} + \mu P_{tL}\right)} - 1 \tag{A.10-6}$$

where P_{sL} is the limit shear load for the fastener and P_{tL} is the limit tensile load for the fastener. Note, that with this approach, a non-negative margin of safety for slip must be shown for each fastener, given that the limit shear and tensile loads may not peak on the same fastener.

A.11 Margin of Safety for Bolted Joint Separation (Supplement to Section 6.5)

The numerator in Eq. (6-23) is P_{p-min}. Although a bolted joint can withstand an applied tensile load somewhat higher than the preload before separation occurs, it is difficult to conservatively calculate the true separation load. This is true because most methods of analysis used to calculate load sharing in a preloaded bolted joint, using bolt stiffness, clamped-material stiffness, and loading-plane factor, have been derived from test data with the intention of conservatively predicting the total bolt load. Any such method that conservatively predicts bolt load over predicts the separation load. In addition, if a preloaded bolt begins to yield under applied tensile load, separation occurs at a lower applied load than linear theory predicts.

In recognition of the above considerations, it was decided that the separation load should be set equal to minimum preload.

A.12 Use of a Fitting Factor

This section provides guidelines for use of fitting factors. Other approaches may be used if acceptable to the program or project.

A supplemental factor of safety called a "fitting factor" is included in the strength and separation analysis of each part of the bolted joint to account for uncertainties in load paths and stresses. For purposes of analysis, joint members integral to a larger structural member are considered part of the bolted joint up to the point where their section properties become typical of the structural member away from the bolted joint.

Ultimate strength analysis of bolted joints should include a fitting factor of at least 1.15 as a multiplier of the required ultimate factor of safety.

Yield strength analysis of bolted joints whose performance is particularly sensitive to local yielding should include a fitting factor of at least 1.15 as a multiplier of the required yield factor of safety.

Analysis of bolted joints in which there is little sensitivity to uncertainty in load paths and stresses may include a fitting factor of 1.0 as a multiplier of the appropriate required factor of safety for strength analysis when one of the following applies:

- The bolted joint's strength is verified in an ultimate load test in which actual load paths and stresses are simulated in the fitting and surrounding structure.

- The bolted joint's load paths and stresses have been determined with detailed finite element analysis using modeling practices that have been correlated with tests of similar bolted joints in which actual load paths and stresses were simulated and measured.

- The bolted joint contains redundant load paths (e.g., multiple fasteners) and sufficient ductility to allow the load to redistribute before failure.

Separation analysis of joints that are separation-critical should include a fitting factor of at least 1.15 as a multiplier of the required separation factor of safety.

Separation analysis may include a fitting factor of 1.0 as a multiplier of the appropriate required factor of safety when the following applies:

- The bolted joint is not separation-critical.

- The bolted joint is separation-critical but the joint's functionality is verified in an ultimate load test in which actual load paths and stresses are simulated in the fitting and surrounding structure.

- The bolted joint is separation-critical but the joint's load paths and stresses have been determined with detailed finite element analysis using modeling practices that have been correlated with tests of similar bolted joints in which actual load paths and stresses were simulated and measured.

APPENDIX B

BEST PRACTICES FOR LOCKING FEATURES

B.1 Purpose of Appendix B

The purpose of Appendix B is to present best practices for locking features.

B.2 Background and Introduction

Threaded fasteners are often used in dynamic environments including vibration, shock, thermal cycles, and pressure cycles. As a result, threaded fasteners often experience dynamic tensile, shear, and moment loads. A properly designed preloaded joint will have sufficient preload (taking into account torque-tension uncertainty, embedment, etc.) such that the dynamic loads have little or no effect on joint integrity. However, if unexpected dynamic loads or temperature cycles are encountered that are sufficient to cause cyclic slip (localized or complete; see Pai and Hess (2002)) at the fastener thread or bearing surfaces, then fastener loosening, preload loss, and component/fastener loss can occur.

The friction at fastener thread and bearing surfaces in a properly designed preloaded joint is the mechanism that prevents slip in a preloaded joint. This friction is proportional to preload. As a result, preload is often referred to as a primary locking feature in a preloaded joint.

In an effort to eliminate or reduce loosening and possible failure in the event of unexpected dynamic loads, temperature excursions or design errors; or to provide locking in fasteners without preload (e.g., fasteners in some electrical applications); additional locking mechanisms or features are often utilized or required. These include the following: (a) mechanical locking devices, such as cotter pins and safety wire; (b) prevailing torque devices such as fasteners with a deformed portion of threads or a polymer patch; (c) adhesives such as anaerobic adhesive; and (d) free spinning locking devices such as fasteners with serrated bearing surfaces or lock washers. Such devices or features are often considered secondary locking features when used in a preloaded joint with preload providing primary locking.

Experience has shown these locking features have the following characteristics:

a. Mechanical locking features, such as cotter pins and safety wire, allow some rotation until abutment, which results in some loss of preload.

b. In severe vibration and shock environments, the main feature of prevailing torque locking devices is to prevent complete separation and loss of fasteners (i.e., provide foreign object debris protection), not to maintain preload.

c. With adequate cure, adhesives perform comparatively well in maintaining preload even in severe vibration and shock environments.

d. Free spinning locking devices, such as fasteners with serrated bearing surfaces, perform comparatively well, but their function is dependent on preload. Because of this dependence on preload, free spinning locking devices cannot be used as a secondary locking feature if preload is used for primary locking. Loss of preload removes both locking features. In addition, serrations can damage joint bearing surfaces. Free spinning split lock and tooth lock washers provide minimal, if any, locking.

e. Figure 18 provides basic guidance for locking feature selection. The following sections of this appendix provide more information and guidance on locking features and devices including appropriate specifications, best practices, limitations, and installation.

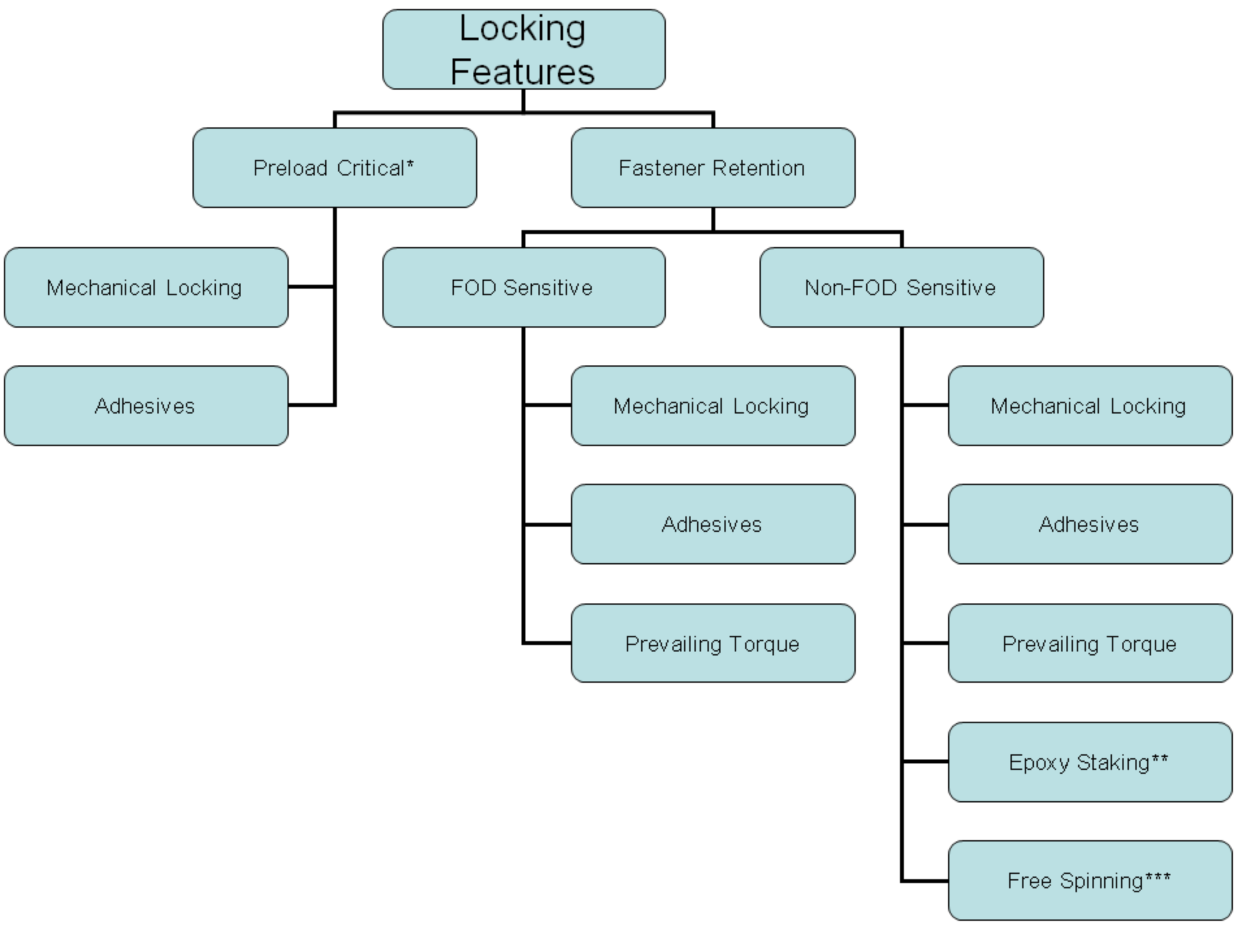

Figure 18—Locking Feature Selection Flowchart

*	Mechanical and adhesive locking features are the best options for preload-critical joints but should not be relied upon to maintain preload. Proper joint design is necessary to maintain preload in dynamic environments.

**	Epoxy staking is most often used for fasteners no larger than #8 or M4. Guidelines for the use of epoxy staking are given in Appendix B.5.1.

***	Locking features, such as jam nuts, may not meet the requirements in section 4.4 if they require preload to function effectively, or if a change in preload could compromise the locking performance.

B.3 Best Practices for Mechanical Locking Features

Mechanical locking features include, but are not limited to cotter pins, safety wire, and safety cable. These parts are covered by industry standards, such as NASM 33540.

Mechanical locking features provide a reliable locking method for joints but come with drawbacks and design details that must be considered. Most mechanical locking features require additional parts which may increase the design's complexity and weight. Sharp edge hazards from mechanical locking features are not acceptable for designs in crew translation paths for manned spaceflight vehicles or crew actuated joints (EVA requirements are outlined in JSC-28918, EVA Design Requirements and Considerations). Mechanical locking features posing sharp edge hazards may be mitigated using encapsulation methods with room-temperature vulcanizing silicones or epoxy; however, materials must be compatible with their use environment.

Mechanical locking features such as cotter pins and safety wire allow some rotation until abutment. This can lead to loss of preload. As an example, Light (1983) presents data showing 30-percent loss of preload under severe vibration conditions from this rotation.

B.4 Best Practices for Prevailing Torque Locking Features

B.4.1 General

a. Compared to fasteners with no secondary locking feature, prevailing torque locking devices can prevent complete separation of a fastening system after preload is lost. Due to the interference introduced by prevailing torque locking devices, they will help reduce the rate at which preload is lost.

b. In severe vibration and shock environments, the main feature of prevailing torque locking devices is to prevent complete separation and loss of fasteners, not to maintain preload.

c. Designs planning to re-use inserts are advised to have the locking feature on the male side to ease replacement and inspection.

B.4.2 Prevailing Torque Locking Devices

B.4.2.1 Self-Locking Nuts

a. Self-locking nuts are nuts with a locking feature resulting from a deformed (e.g., to an elliptical shape), undersized portion of the threads or incorporating nonmetallic (polymeric) insert. The locking feature can also be provided by a non-metallic (such as nylon) element that protrudes from the nut internal threads. The locking feature can be verified by measuring running torque.

b. NASM 25027, Nut, Self-Locking, 250 Degrees F, 450 Degrees F, and 800 Degrees F; and NAS 3350, Nuts, Self-Locking, 450 Degrees F and 800 Degrees F, High Quality, are the *performance* specifications for self-locking nuts. These include maximum locking (prevailing) torque and minimum breakaway torque values. NASM 25027 covers metal nuts with non-metal and metal locking features. NAS 3350 covers nuts with all all-metal locking features.

c. The unseated cycle life of 15 cycles required for qualification by NASM 25027 and NAS 3350 should not be assumed for seated cycle life. No seated cycle life requirement exists. Seating the bolt will affect the locking torque and the number of locking cycles. The extent of the effect is dependent upon materials of construction, lubrication, and preload.

d. Usage limitations are provided in NASM 33588, Nut, Self-Locking, Aircraft, Reliability and Maintainability Usage Requirements For. Examples include the following:

(1) Not using self-locking nuts on an axis of rotation.
(2) Ensuring external thread extends at least two threads through nut.
(3) Not using silver-plated nuts with silver-plated bolts. Temperature limitations for nonmetallic locking features are provided in NASM 25027.

e. Examples of self-locking nuts are as follows:

(1) NAS 1291, Nut, Self-Locking, Hexagon – Low Height, Light Weight.
(2) NASM 21043, Nut, Self-Locking, 800 Degrees F, Reduced Hexagon, Reduced Height, Ring Base, Corrosion Resistant Steel.
(3) NASM 21045, Nut, Self-Locking Hexagon – Regular Height, 450 Degrees F, 125 ksi Ftu.
(4) NASM 21046, Nut, Self-Locking, Hexagon-Regular Height, 800 Degrees F, 125 ksi Ftu.

B.4.2.2 Self-Locking Nutplates

a. Self-locking nutplates use the same locking features as self-locking nuts.

b. Examples of self-locking nutplates are NAS 1068, Nut, Self-Locking Plate, Two Lug, Low Height, C'bored, Floating, Reduced Rivet Spacing; and NAS 1773, Nut, Self-Locking, Plate, Two Lug, Low Height, C´Bored, Floating, 160 ksi, 450 Degrees F, 800 Degrees F; and NAS 1774, Nut, Self-Locking, Plate, One Lug, Low Height, C´Bored, Floating, 160 ksi, 450 Degrees F, 800 Degrees F.

B.4.2.3 Self-Locking Bolts with Nonmetallic Locking Feature

a. Self-locking bolts are bolts with a locking feature resulting from a polymer plug, pellet, strip, or patch.

b. Vibra-TITE® is pre-applied and dried before installation with no bonding between mating threads and is therefore not considered an adhesive. Performance characteristics per MIL-DTL-18420 could be used.

c. MIL-DTL-18240, Fastener Element, Self-Locking, Threaded Fastener, 250° F Maximum, is the specification for locking bolts up to 121 °C (250 °F) (temperature limit due to polymer locking material). This specification includes maximum and minimum running torque values.

d. A seated cycle life of five cycles is required for qualification by MIL-DTL-18240.

e. Usage limitations are given in NAS 15981 and are similar to those given for locknuts in NASM 33588.

f. Examples of fasteners that can be purchased as self-locking are as follows:

(1) NAS 6703–NAS 6720.
(2) NAS 8100–NAS 8106.
(3) NAS 1189.
(4) NAS 1351.

B.4.2.4 Self-Locking Helical-Coil Inserts

a. Self-locking helical-coil inserts are coils of wire (e.g., diamond-shaped wire). They are installed in a tapped parent metal hole with a special tool and are slightly oversized to create a frictional interlocking. The locking feature is provided by a deformed or undersized portion of the coil (e.g., hex shape for one thread) that creates a running torque.

b. NASM 8846, Inserts, Screw-Thread, Helical Coil, is the specification for self-locking helical coils. This includes maximum locking (prevailing) torque and minimum breakaway torque values.

c. The unseated cycle life of 15 cycles required for qualification by NASM 8846 should not be assumed for seated cycle life. No seated cycle life requirement exists. Seating the bolt will affect the locking torque and the number of locking cycles. The extent of the effect is dependent upon materials of construction, lubrication, and preload.

d. HeliCoil® Technical Bulletin 68-2 provides comprehensive data on pullout strength for a range of available insert lengths and parent material shear strength.

e. It is common practice to “wet install” helical-coils in aluminum structure to prevent galvanic corrosion between a corrosion-resistant steel (CRES) insert and an aluminum structure. Primers, such as those conforming to MIL-PRF-23377, Primer Coatings: Epoxy, High-Solids, should be applied to the insert/base metal interface. Excess primer may migrate to areas contacting the installed fastener and affect torque.

f. In general, helical-coils are easier to replace than key locked inserts. However, if not installed correctly, helical-coils have the potential to back out when subjected to high vibrations.

B.4.2.5 Self- Locking Key Locked-In Inserts

a. Key locked-in inserts have integrally assembled keys on the outer diameter that stake (i.e., mechanically lock) the inserts inside a parent metal hole. The locking feature in a self-locking key locked-in insert is a deformed or undersized portion of the internal threads that creates a running torque.

b. MIL-I-45914, Insert, Screw Thread – Locked In, Key Locked, General Specification For, is the specification for key locked-in inserts.

c. The unseated cycle life of 15 cycles required for qualification by MIL-I-45914 should not be assumed for seated cycle life. No seated cycle life requirement exists. Seating the bolt will affect the locking torque and the number of locking cycles. The extent of the effect is dependent upon materials of construction, lubrication, and preload.

d. Heavy-duty and extra-heavy-duty key locked-in inserts provide more pullout resistance than thin-wall locked-in inserts and helical-coil inserts of the same length. MIL-I-45914 provides data on pullout required for qualification.

e. Key locked-in inserts are solid walled so that application of corrosion prevention primer is not as problematic as with helical-coil inserts.

f. Typical examples of key locked-in inserts are MS 51830–MS 51832.

B.4.2.6 Self-Locking Thin-Wall Locked-In Inserts

a. Thin-wall, locked-in inserts are solid inserts that have a cylindrical shoulder at the top which is swaged into the parent metal with a special swaging tool. Swaging locks the insert into the hole and inhibits torque-out. The locking feature in self-locking thin-wall locked-in inserts is a deformed or undersized portion of the internal threads that creates a running torque on the fastener.

b. MIL-I-45932, Insert, Screw Thread, Thin Wall, Locked In, General Specification For, is the specification for thin-wall locked-in inserts.

c. The unseated cycle life of 15 cycles required for qualification by MIL-I-45932 should not be assumed for seated cycle life. No seated cycle life requirement exists. Seating the bolt will affect the locking torque and the number of locking cycles. The extent of the effect is dependent upon materials of construction, lubrication, and preload.

d. Thin-wall inserts are useful in joints that require less edge distance compared to key locked-in inserts which require larger edge distance to prevent shear tear-out.

B.4.3 Inspect

a. Parts are to be inspected (see section 7.5).

b. Prevailing torque locking devices are generally used “as-received” from manufacturing. Wipe away any visible loose particles and contaminants but do not clean with solvents.

B.4.4 Installation and Verification

a. Verify the prevailing torque locking device meets the appropriate specification, such as:

(1) NASM 25027 and NAS 3350 for self-locking nuts and nutplates.

(2) MIL-DTL-18240 for self-locking bolts.

(3) NASM 8846 for self-locking helical-coil inserts.

(4) MIL-I-45914 for self-locking key locked-in inserts.

(5) MIL-I-45932 for self-locking thin-wall locked-in inserts.

b. Finger assemble up to locking feature.

c. Assemble two full turns further.

d. Once locking feature is fully engaged, measure break away torque (torque to start motion in the on or off direction) and running torque.

e. Should have at least one full turn left before fastener starts to seat.

f. Verify measured torque (either breakaway or prevailing) is greater than the minimum breakaway torque required in the appropriate qualification specification.

g. If both measured torques are within specification, then seat and tighten fastener to tightening torque which is the sum of the preload torque required for desired preload plus the running torque.

h. Record the applied torque, preload torque, running torque, locking device part number, and lot number.

B.5 Best Practices for Adhesive Locking Features

B.5.1 Adhesives

a. Anaerobic

(1) Anaerobic adhesives (e.g., Loctite®) are single component thread-locking compounds that cure in the absence of air. They are applied to the threads of a fastener just prior to installation and then cure in place between the mating threads. The locking feature results from filling the clearance between mating threads and preventing relative slip.

(2) MIL-S-22473, Primer Coatings: Epoxy, High-Solids Sealing, Locking, and Retaining Compounds: (Single Component); MIL-S-46163, Sealing, Lubricating and Wicking Compounds: Thread-Locking, Anaerobic, Single-Component; and American Society for Testing and Materials (ASTM) D5363-03, Standard Specification for Anaerobic Single-Component Adhesives, are the specifications for anaerobic adhesives. However, MIL-S-22473, Sealing, Locking, and Retaining Compounds: (Single Component), has been cancelled and MIL-S-46163 has been inactivated for new design so the ASTM standard should be used.

(3) Detailed process steps should be followed including surface preparation, cleaning, and primer application.

(4) Typical design goals include selecting adhesive such that the break-loose torque is equal to or greater than the installation torque, and to meet prevailing torque device specification requirements when repairing worn-out, prevailing-torque locking features.

(5) Examples of anaerobic adhesives include Loctite® 222, removable grade for up to 6.35-mm (0.25-in) screws; Loctite® 242, removable grade for up to 19.05-mm

(0.75-in) bolts; and Loctite® 262, permanent grade for up to 19.05-mm (0.75-in) bolts.

(6) Caution: When using anaerobic adhesives in blind holes, entrapment of air in the hole and threads may prevent proper cure or push the adhesive out of the threads, leaving insufficient adhesive for locking.

b. Epoxy and Staking Compounds

(1) These materials are used by applying a small amount of the compound onto both the fastener (usually the head) and the surrounding structure such that when cured, the adhesive bond to both fastener and structure prevents rotation of (stakes) the fastener. This method is often used in electronic assemblies where a lost fastener is captive within the assembly and cannot adversely affect the function of other components. Staking methods are also sometimes used to secure fine-adjustment screws in position. Since adhesives are most susceptible to failure in peel, staking is considered a low-reliability locking feature. Epoxy and staking compounds are typically two-part materials requiring mixing before use.

(2) Detailed process steps should be followed, including surface preparation and cleaning, mixing components of compound, and application.

(3) Epoxy and staking compounds used in sensitive applications such as fluid wetted components, optical systems and some external spacecraft surfaces can result in system contamination and performance degradation. Use of these materials should be considered for system level contamination effects during design.

c. Other

(1) Arathane® cures with or without oxygen, remains gummy when cured, and in some instances has been found to be effective even with residual lubricant on threads (HST-TR-010334, 2008).

(2) Vibra-TITE® is pre-applied and dried before installation with no bonding between mating threads and is therefore not considered an adhesive.

B.5.2 Process Validation

a. Acquire test verification specimens, as follows:

(1) Obtain at least five representative or witness fastener specimens (e.g., bolts, nuts, inserts, washers) of the same material, size, finish, part number, and lot number as used in intended application.

(2) Obtain representative clamped components of the same materials, finishes, clamping thickness, hole size, and tolerances. Parts of smaller size than clamped component in application are acceptable as long as the stiffness is comparable (e.g., cylinders).

(3) Record part and lot numbers of specimens and description of clamped components.

b. Inspect and Clean (see Appendices B.5.3 and B.5.4), as follows:

(1) Visually inspect and verify that fastener components (e.g., bolts, nuts, washers, inserts, etc.) are free of damage. Discard damaged components.

(2) Clean external threads and internal threads but not the bearing surfaces of the fastener specimens (e.g., bolt, nut, and washers) in the same manner as is used in the intended application (e.g., using cloth, wipes, or swabs dampened with appropriate solvent, such as acetone or methyl ethyl ketone (MEK)). If bearing surfaces must be cleaned, bearing surface lubrication may be necessary to prevent galling.

(3) Inspect and verify that the cleaned internal and external threads of test verification specimens are dry and free of residue, loose particles, and contamination as in intended application.

c. Prime

(1) For epoxy staking applications, apply one coat of primer to bonding surfaces.

(2) For anaerobic adhesive applications, apply one coat of primer (e.g., Loctite® Primer grade T or N) to the internal threads and external threads of the specimens as in intended application.

(3) Apply primer by spray, brush, or swab and allow primer to dry for 15 minutes minimum or up to 5 days maximum, as in intended application.

(4) Record primer part number (or grade), lot number, expiration date, and application date and time.

d. Apply Adhesive, Torque, and Cure, as follows:

(1) Prepare adhesive as specified by manufacturer (e.g., for epoxy adhesive, mix resin and hardener; for anaerobic adhesive, shake bottle).

(2) Apply adhesive to both external and internal threads, 360 degrees around and over the complete thread engagement area, as in application.

(3) Assemble representative fastener specimens and clamped components to torque specified in application.

(4) Record adhesive part number (or grade), lot number, expiration date, and application date and time.

(5) Allow to cure for cure time (e.g., 24 or 48 hours) specified in application.

e. Validation Test

(1) Measure and record break loose torque (off direction) for each specimen.

(2) Measure and record removal running torque for each specimen.

(3) Depending upon design goal, (a) verify that the measured break loose torque (off direction) is greater than or equal to the installation torque; or (b) verify the running torque meets prevailing torque device specification requirements.

(4) Verify adhesive has fully cured by inspection.

B.5.3 Inspect

a. Parts are to be inspected (see section 7.5).

b. Discard damaged components.

B.5.4 Clean

a. Clean external threads and internal threads but not the bearing surfaces of the fastener specimens (e.g., bolt, nut, and washers) in the same manner as is used in the intended application (e.g., using cloth, wipes, or swabs dampened with appropriate solvent, such as acetone or MEK). If bearing surfaces must be cleaned, bearing surface lubrication may be necessary to prevent galling.

b. Inspect and verify that the cleaned threads are dry and free of residue, loose particles, or contamination.

B.5.5 Prime (for anaerobic adhesives only)

a. Use primer on inactive metals (e.g., aluminum, stainless steel, titanium, anodized, cadmium plated, and zinc plated).

b. Apply one coat of primer to both the internal and external threads by spray, brush, or swab.

c. Allow primer to dry for 15 minutes minimum.

d. Primed parts may be stored for up to 5 days maximum in sealed plastic bags. If not used within 5 days, re-clean and re-prime parts.

e. Record primer part number (or grade), lot number, expiration date, and application date and time.

B.5.6 Installation

a. Verify adhesive is within shelf life.

b. Prepare adhesive as specified by manufacturer (e.g., for epoxy adhesive, mix resin and hardener; for anaerobic adhesive, shake bottle).

c. Apply adhesive to internal and external threads, 360 degrees around and over the complete thread engagement area for blind holes. Apply adhesive to external threads, 360 degrees around and over the complete thread engagement area for through-hole assemblies. Ensure adhesive does not contact bearing surfaces.

d. Assemble within 5 minutes to specified torque.

e. Wipe excess adhesive from parts.

f. Record adhesive part number (or grade), lot number, expiration date, and application date and time.

g. Allow to cure for cure time (e.g., 24 or 48 hours) specified by manufacturer or engineering.

B.5.7 Verification

a. Verify by using processes outlined in the process validation section (see Appendix B.5.2) using a coupon method.

b. Five samples/coupons per configuration are required and need to be verified for each locking compound batch and technician.

c. Inspect adhesive to ensure full cure.

B.5.8 General

a. Cleaning and adhesives alter nut factor as follows:

(1) Cleaning process removes residual lubricant from components (e.g., bolts, nuts, inserts, washers, etc). Either be extremely careful not to remove residual lubricant from bearing surfaces or use lubricant between bearing surfaces of similar material (e.g., stainless steel bolt head and washer) to prevent galling.

(2) Use the same cleaning, priming, adhesive application, and bearing surface treatment in the intended application as in the torque-tension tests used to obtain nut factor if preload is determined by torque measurement.

b. Inadequate cleaning and priming of threads can lead to poor locking performance of adhesives.

c. With adequate cleaning and priming of threads, adhesives can prevent loosening and maintain preload better than prevailing torque locking devices and mechanical locking devices in severe vibration and shock environments.

d. Although adhesives are an appealing locking feature when an original prevailing torque locking feature has worn out, they are not a simple “quick fix.” The detailed process and verification steps of Appendix B.5 should be followed.

B.6 Best Practices for Free Spinning

a. The mechanism of locking for free spinning locking devices is dependent on preload. Because of this dependence on preload, free spinning locking devices cannot be used as a secondary locking feature if preload is used for primary locking.

b. Free spinning locking devices with serrated bearing surfaces perform well compared to other locking devices, however, their function is dependent on preload (see above).

c. Free spinning locking devices with serrated bearing surfaces can damage joint bearing surfaces which can lead to crack formation and fatigue failure.

d. Free spinning split lock washers provide minimal, if any, locking. Air Force-Navy Aeronautical Design Standard AND 10476, Washer – Limitations on Usage of Lock, Standard limits the use of lock washers to situations when other locking features cannot be used and when loosening would not endanger the safety of structure or personnel.

e. Split lock washers should not be used for secondary locking features.

APPENDIX C

JUSTIFICATION FOR THE LOW-RISK FATIGUE CLASSIFICATION

C.1 Purpose of Appendix C

The purpose of this appendix is to provide justification for the low-risk fatigue classification.

The low-risk classification in section 4.2 is based on fatigue analysis using data acquired in tensile fatigue testing of 160-ksi, A-286, NAS1351N fasteners and a hypothetical loading spectrum intended to be worst case for single-mission spaceflight hardware. The NAS1351N fastener is a socket-head cap screw with a UNR thread form, which has a specified root radius and is thus more fatigue resistant than the UN and M thread forms but is not as fatigue resistant as the UNJ and MJ thread forms, which have a larger specified root radius. Thus, any conclusions drawn from fatigue analysis using this test data are conservative for A-286 fasteners with rolled UNJ or MJ threads.

The fatigue tests were conducted on two sizes of fasteners, 0.190-in diameter (4.83 mm) and 0.375-in diameter (9.53 mm). The smaller fasteners were from three lots, each from a different supplier, whereas the 0.375-in (9.53 mm) diameter fasteners were from a single lot. Three fasteners were tested from each lot for each of eleven combinations of maximum load and stress ratio, R, which is defined as the minimum applied load within a loading cycle divided by the maximum applied load within that cycle. Each fastener was cycled within its given load parameters until rupture or until one million cycles were reached, whichever occurred first.

S-N curves were used to assess fastener fatigue life for a joint and a loading spectrum that together are intended to represent a worst-case scenario for single-mission flight hardware. The joint assessed is the one shown in figure 7, for which, prior to separation, the bolt's tensile load increases by a high percentage of applied relative to most joints. The loading spectrum used in the fatigue analysis is shown in table 4, Hypothetical Applied-Loading Spectrum Used to Justify the Low-Risk Classification for Fastener Fatigue. Each level of applied load is assumed to be fully reversed (tensile and compressive).

Table 4—Hypothetical Applied-Loading Spectrum Used to Justify the Low-Risk Classification for Fastener Fatigue

Applied load as a percentage of limit load	Vibration Test	Flight	Total	Design (4x)
10%	4752	259	5011	20045
20%	13038	651	13689	54756
30%	18176	760	18935	75741
40%	19464	624	20088	80351
50%	17507	390	17897	71588
60%	13689	199	13889	55555
70%	9466	81	9547	38189
80%	5845	27	5872	23486
90%	3241	7	3249	12995
100%	2821	2	2822	11290
Total cycles	108000	3000	110999	443996
Basis (assumptions):	The bolt is loaded highest by random-vibration response of a 200-Hz mode of vibration. The flight structural assembly is protoflight tested for 1 minute per axis, 3 dB above maximum predicted environment, at three levels of assembly, for a total of 3 minutes per axis. The 200-Hz mode is excited equally by all three axes of excitation (conservative assumption). Random vibration is at or near maximum predicted environment for 15 seconds during launch. The loading spectra for test and launch are based on a Rayleigh distribution. Limit load for protoflight-level random vibration is 3σ (three times the RMS value) and is set equal to $0.5P_{tu\text{-}allow}$. All loading cycles above 3σ from the Rayleigh distribution are included in the cycles at 100% limit load. The number of cycles at each load increment includes a service life factor of 4.			

To assess the bolt in the subject joint for fatigue life, the applied-loading spectrum is transformed into two bolt-loading spectra . One spectrum is based on the assumption of a high preload, equal to $0.75P_{tu\text{-}allow}$, and the other spectrum is based on the assumption of a low preload, equal to $0.5P_{tu\text{-}allow}$, which is the lowest the preload could be in order for the separation criteria to be met with a limit load equal to $0.5P_{tu\text{-}allow}$. For each level of applied load, the total bolt load is calculated using Eq. (6-7). For fully reversed applied load, as assumed in this study, Eq. (6-7) is conservatively assumed to apply for compressive loading as well as for tensile loading, such that the mean load is equal to the preload.

Miner's rule was used in this study as the failure criterion:

$$D = \sum_{i=1}^{j} \frac{n_i}{N_i} \leq 1 \tag{C-1}$$

where D is the cumulative damage, ni is the number of loading cycles at a given stress level, Ni is the number of cycles to failure at that stress level, and j is the number of different stress levels.

Table 5, Results of Fastener Fatigue Analysis for the Hypothetical Joint and Loading Spectra, shows the results of the fatigue analyses where the stress ratios were calculated using nonlinear finite element analysis. With both loading spectra, the Miner's rule criterion passes.

Table 5— Results of Fastener Fatigue Analysis for the Hypothetical Joint and Loading Spectra

Max load is normalized to the allowable ultimate tensile load, $P_{tu\text{-}allow}$.				
Case 1: High preload				
Design cycles, ni (4x)	Max load	Stress ratio, R	Allowable cycles, Ni	ni/Ni
20052	0.754	0.99	1000000	0.020
54760	0.756	0.98	1000000	0.055
75744	0.759	0.98	1000000	0.076
80356	0.762	0.97	1000000	0.080
71592	0.765	0.96	1000000	0.072
55560	0.771	0.95	1000000	0.056
38196	0.775	0.94	1000000	0.038
23492	0.782	0.92	1000000	0.023
13000	0.791	0.90	1000000	0.013
11292	0.800	0.88	125000	0.090
444044			D =	0.523
				PASS

Case 2: Low preload				
Design cycles, ni (4x)	Max load	Stress ratio, R	Allowable cycles, Ni	ni/Ni
20052	0.504	0.99	1000000	0.020
54760	0.506	0.98	1000000	0.055
75744	0.509	0.96	1000000	0.076
80356	0.514	0.95	1000000	0.080
71592	0.520	0.92	1000000	0.072
55560	0.527	0.90	800000	0.069
38196	0.539	0.86	750000	0.051
23492	0.552	0.81	700000	0.034
13000	0.568	0.76	400000	0.033
11292	0.585	0.71	200000	0.056
444044			D =	0.545
				PASS

C.2 Conclusions

This analysis shows that, for a fastener meeting all of the stipulations for low-risk classification, as noted in section 4.2, fatigue under tensile loading is not a concern for the above hypothetical loading spectrum. The test data used in this study are from testing of NAS1351N fasteners, which have the UNR thread form. Although the study justifies a limit tensile load as high as $0.5P_{tu\text{-}allow}$, the stipulations in section 4.2 allow a limit tensile load of only $0.4P_{tu\text{-}allow}$ for fasteners having the UNR thread form. This apparent discrepancy is intentional as a result of uncertainty, given the limited fatigue data used in the analysis and the assumptions made in the analysis. The UNJ and MJ thread forms have larger root radii than UNR and thus have better fatigue life. Given this fact along with the lack of fatigue data for A-286 fasteners with UNJ or MJ threads, the stipulations in section 4.2 allow a limit tensile load of $0.5P_{tu\text{-}allow}$ for fasteners having UNJ or MJ threads based on the results of the above analysis.

The hardware developer should determine whether their particular loading spectrum is enveloped by the spectrum used in this analysis.

APPENDIX D

REFERENCES

D.1 Purpose of Appendix D

The purpose of this appendix is to provide guidance through the reference documents listed below.

D.2 Reference Documents

Document Number	Document Title
	HeliCoil® Technical Bulletin 68-2 (Emhart Teknologies, Shelton, Connecticut)
	Hahn, Gerald J.; Shapiro, Samuel S. (1967). *Statistical Models in Engineering*. New York: John Wiley & Sons, Inc.
	Light, D.J. (May 1983). "Vibration loosening of threaded fastenings," *Chartered Mechanical Engineer.* Vol. 30, No. 5, pp. 56-60.
	Nassar, S.A.; Abboud, A.G. (December 2009). "An Improved Stiffness Model for Bolted Joints." *ASME Journal of Mechanical Design.* Vol. 131, Issue 12, pp. 121001-1 to 121001-11.
	Nassar, S. A.; Ganganala, M. "Effect of Separating Load Eccentricity on the Clamp Load Loss in a Bolted Joint Using a Strain Hardening Model." *ASME Journal of Pressure Vessel Technology*. (Reviewed and accepted manuscript #PVT-09-1060.)
	Odeh, Robert E.; Owen D. B. (1980). *Tables for Normal Tolerance Limits, Sampling Plans and Screening*. New York: Marcel Dekker, Inc.
	Spiegel, Murray R. (1961). *Schaum's Outline of Theory and Problems of Statistics*. New York: McGraw-Hill Book Company.

Document Number	Document Title
	Stronge, William James; Yu, Tongxi. (1993). *Dynamic Models for Structural Plasticity.* London, Great Britain: Springer-Verlag.
AND 10476	Washer – Limitations on Usage of Lock
ASME B18.12-2001	Glossary of Terms for Mechanical Fasteners
ASTM D5363	Standard Specification for Anaerobic Single-Component Adhesives (AN)
FAA AC 20-71	Dual Locking Devices on Fasteners
FAA AC 23.607-1	Self-Locking Nuts on Bolts Subject to Rotation
FED-STD-H28	Federal Standard Screw-Thread Standards for Federal Services
HST-TR-010334	Cooper, James. (August 4, 2008). *Arathane Thread Locking Vibration Test Report*. Greenbelt, MD: NASA Goddard Space Flight Center.
JSC-28918	EVA Design Requirements and Considerations
MIL-DTL-18240	Fastener Element, Self-Locking, Threaded Fastener, 250° F Maximum
MIL-HDBK-60	Threaded Fasteners-Tightening to Proper Tension
MIL-I-45914	Insert, Screw Thread – Locked In, Key Locked, General Specification For
MIL-I-45932	Insert, Screw Thread, Thin Wall, Locked In, General Specification For
MIL-PRF-23377	Primer Coatings: Epoxy, High-Solids
MIL-S-22473E	Primer Coatings: Epoxy, High-Solids Sealing, Locking, and Retaining Compounds: (Single Component)
MIL-S-46163A	Sealing, Lubricating and Wicking Compounds: Thread-Locking, Anaerobic, Single-Component

Document Number	Document Title
NAS 618	Fastener – Recommended Shank, Hole, and Head-to-shank Fillet Radius Limits For
NAS 1189	Screw, Self-Locking, Flat 100 Degree Head, Full Thread
NAS 1291	Nut, Self-Locking, Hexagon – Low Height, Light Weight
NAS 1351	Screw, Cap, Socket Head, Undrilled and Drilled, Plain and Self-Locking, Alloy Steel, Corrosion-Resistant Steel and Heat-Resistant Steel, UNRF-3A
NAS 1773	Nut, Self-Locking, Plate, Two Lug, Low Height, C´Bored, Floating, 160 ksi, 450 Degrees F, 800 Degrees F
NAS 1774	Nut, Self-Locking, Plate, One Lug, Low Height, C´Bored, Floating, 160 ksi, 450 Degrees F, 800 Degrees F
NAS 3350	Nuts, Self-Locking, 450 Degrees F and 800 Degrees F, High Quality
NAS 6703–NAS 6720	Bolt, Tension, Hex Head, Close Tolerance, A286 CRES, Long Thread, Reduced Major Thread Diameter, Self-Locking and Nonlocking, 160 ksi Ftu
NAS 8100-NAS 8106	Screw, Pan Head, Cruciform Recess, A-286 CRES, Full Thread, Self-Locking and Non Locking
NAS 15981	Fasteners, Externally Threaded, Self-Locking Design and Usage Limitations For
NASM 8846	Inserts, Screw-Thread, Helical Coil
NASM 1312-15	Fastener Test Methods, Method 15, Torque-Tension

Document Number	Document Title
NASM 21043	Nut, Self-Locking, 800 Degrees F, Reduced Hexagon, Reduced Height, Ring Base, Corrosion Resistant Steel
NASM 21045	Nut, Self-Locking Hexagon – Regular Height, 450 Degrees F, 125 ksi Ftu
NASM 21046	Nut, Self-Locking, Hexagon-Regular Height, 800 Degrees F, 125 ksi Ftu
NASM 25027	Nut, Self-Locking, 250 Degrees F, 450 Degrees F, and 800 Degrees F
NASM 33540	Safety Wiring, Safety Cabling, Cotter Pinning, General Practices for
NASM 33588	Nut, Self-Locking, Aircraft, Reliability and Maintainability Usage Requirements For
NSTS 08307	"Criteria for Preloaded Bolts," NASA Johnson Space Center, Revision A, July 6, 1998
SAE AIR 1754A	Washer, Thermal Compensating, Metric Series
SAE AS567	Safety Cable, Safety Wire, Key Washers, and Cotter Pins for Propulsion Systems, General Practices for Use of
VDI 2230	"Systematic Calculation of High Duty Bolted Joints, Joints with One Cylindrical Bolt." (February 2003). Düsseldorf, Germany: Verein Deutscher Ingenieur.