Design and Analysis of Bolted Joints

for Aerospace Engineers

Short Course

Objectives:

- Help you understand how to design bolted joints that
  - function as required
  - are relatively inexpensive and easy to assemble
  - are trouble-free
- Share traditional methods of analysis and help you understand when they apply and why they are effective
- Help you understand NASA's new standard for threaded fastening systems, NASA-STD-5020 (released in March 2012)
- Provide a valuable reference and a trail to data sources

Audience:

- Mechanical design engineers
- Structural analysts
- Others interested in the design of joints

Course developed and taught by Tom Sarafin

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DABJ History

- First version developed in 1998 at the request of NASA Goddard
  - Titled "Design and Analysis of Fastened Joints" (DAFJ); 8-hour course, taught twice in 1998
- Revised into a 16-hour course in 1999
  - Taught 21 times in this format; course materials revised (improved) after nearly each class
- Revised into a 24-hour course in 2005 at NASA JSC request to include a section on compliance with NSTS 08307 and other NASA standards
  - This exercise included several meetings with JSC experts; all concluded that NSTS 08307 should be revised or replaced
    - Helped fuel the NASA-STD-5020 project (new standard for threaded fastening systems), which started in 2007. Tom Sarafin served as one of the core team members for this project.
  - Course renamed "Design and Analysis of Bolted Joints" (DABJ)
  - Taught 19 times in this format, with periodic revision to capture additional information
- Major revision in June 2010 to include a section on analysis criteria per the draft NASA-STD-5020, which was being developed at that time
  - Taught 10 times between June 2010 and March 2012
- Major revision in April 2012 to be consistent with the final version of NASA-STD-5020, which was released in March 2012
  - Taught 8 times as of July 2013

To date, Tom Sarafin has taught this course, in its evolving versions, a total of 60 times to more than 1200 engineers in the space and aircraft industries

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Course Topics

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Representative Fastener Issues on the Shuttle and Space Station Programs

1996 Space Shuttle mission (STS-80): A small screw with no locking feature backed out and jammed a gear in external airlock, preventing the astronauts from opening the hatch and performing the Extra-Vehicular Activity (EVA) part of the mission

In a 2006 EVA, while astronauts tried to activate the Solar Alpha Rotary Joint on the International Space Station, a bolt in the launch restraint seized and required extremely difficult removal, injuring a crew member

Multi-Purpose Logistics Module (MPLM): Hundreds of fasteners too short to fully engage threads and engage the locking feature, not detected during installation. Running torque not verified for any of the fasteners used to assemble the MPLM.

Space Shuttle Ku Band Antenna: After several missions, 2 of 4 main attachment bolts were discovered in 2006 to be too short to engage locking features or provide adequate strength; required risky repair on launch pad

During a 2006 Shuttle mission, an EVA camera came loose and was lost because the mounting screws backed out

Clearly the space industry needs to improve how threaded fasteners are used, controlled, and assessed!

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Designing a Bolted Joint

Where in course	
Sec. 1	1. Identify functional requirements and constraints for the structure being designed – Items to support, envelope, alignment
Sec. 3	2. Develop a concept – Structural configuration and form of construction – Method of attachment: welding, bonding, or fastening – Concept for the joint: configuration, types of fasteners, access for assembly
Sec. 1	3. Quantify requirements and identify design considerations for the joint – Life-cycle environments, design loads, loading cycles, and temperatures – Stiffness, allowable permanent deformation, and design criteria – Cost, lead time, ease of assembly, and schedule
Secs. 4 - 6	4. Size the joint – Select fastener pattern – Calculate fastener loads; size fasteners – Identify potential failure modes in the fittings and test-substantiated methods of assessment – Size fittings
Sec. 7	5. Select hardware and design details – Specific bolts, nuts, washers, pins – Edge distance, wrench clearance, hole size
Sec. 7	6. Specify assembly requirements – Lubrication, installation torque, other
Secs. 8 & 9	7. Perform detailed analysis

Iterate as needed

Goal: a joint that ...

- **functions as needed throughout its intended life cycle**
- **uses affordable and available hardware**
- **is easy to assemble (and often disassemble)**

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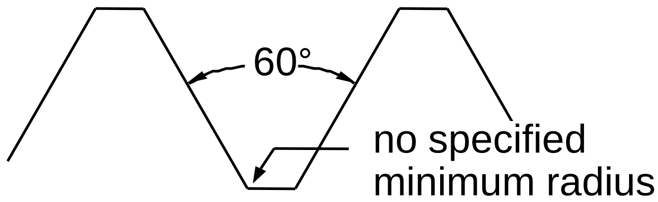
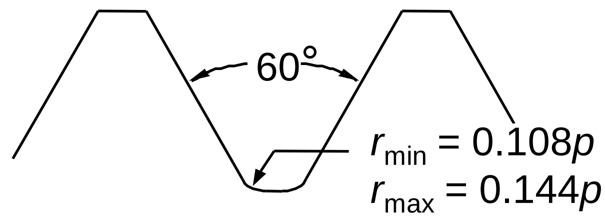
Comparison of Thread Forms

UN
(metric: M)

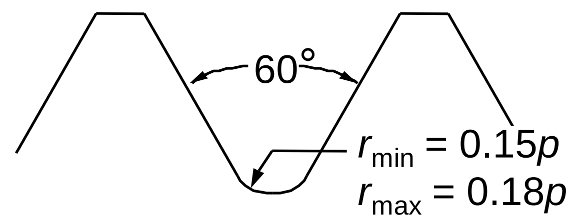


UNR
(no metric equivalent)
(external only)



For critical bolts in flight hardware, use UNJ threads, and make sure they're formed by rolling after heat treatment

UNJ
(MJ)



(full radius)

(Ref. 5, SAE AS8879 controls the dimensions of the UNJ thread form but does not specify that the threads must be rolled after heat treatment)

A large root radius provides better fatigue life

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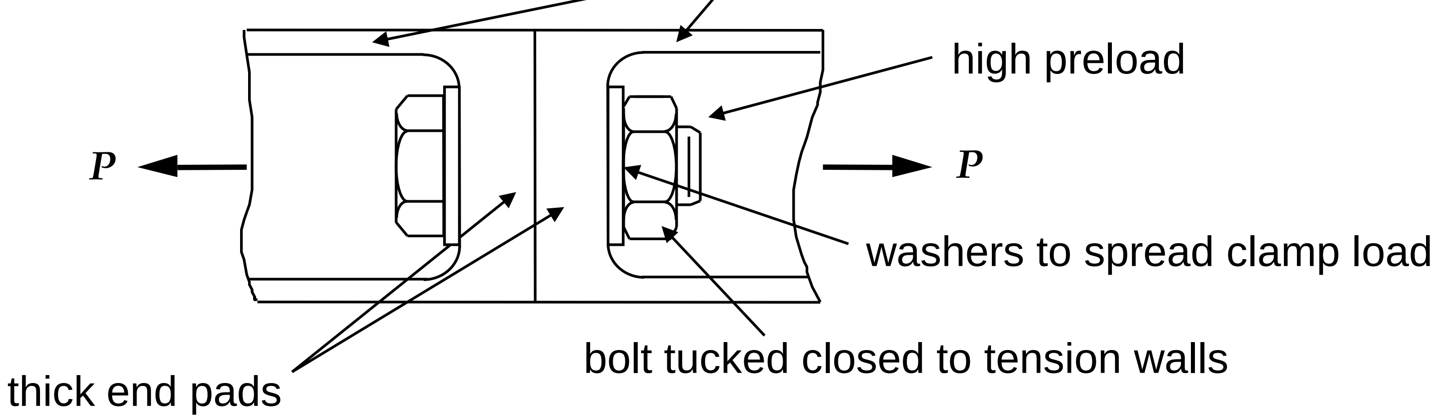
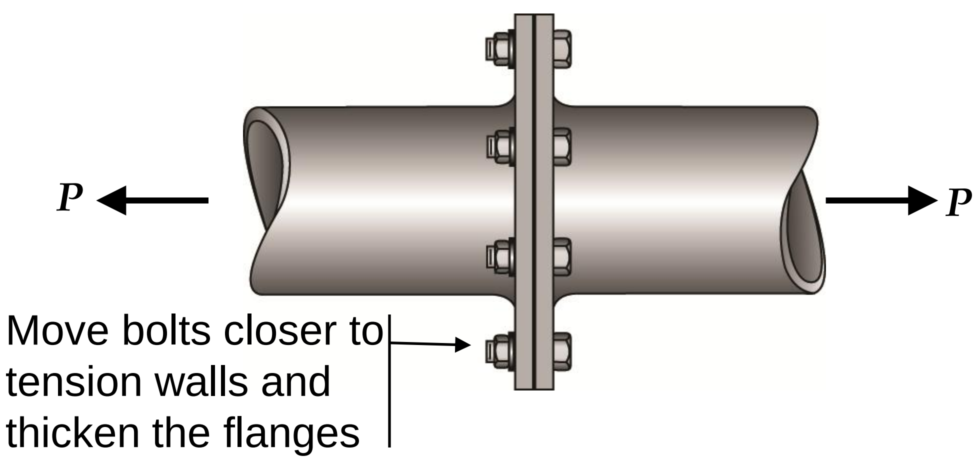
Designing Stiff Tension Joints

Joint with high and relatively linear stiffness:



Joint that is stiff in compression and flexible in tension (avoid):



- The thin flange, with bolts far from the tension walls, will bend and pry on the bolts
- Nonlinear stiffness makes it difficult to predict dynamic response or load distribution within the structural assembly

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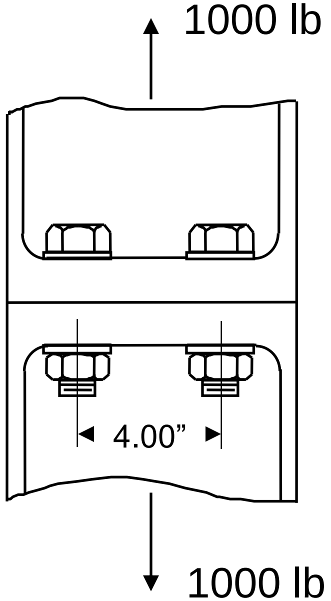
Calculating Bolt Loads is Often Based Simply on Statics

Example: Tension joint with 2 bolts

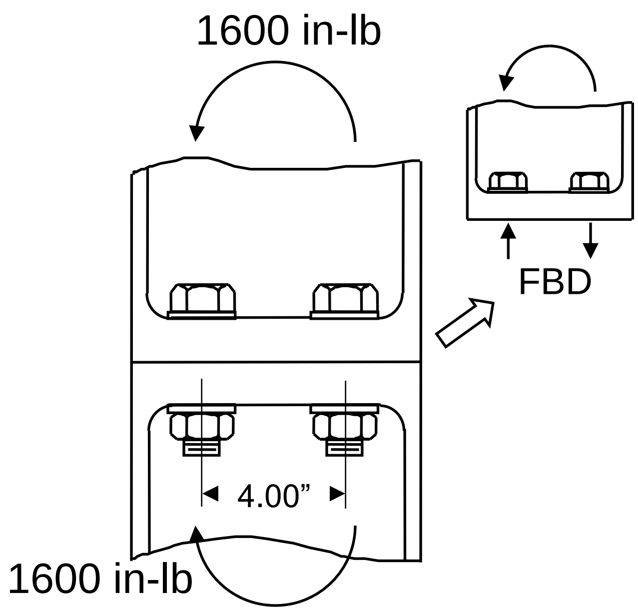
Case 1: Tensile applied load, with load vector centered between bolts



Ignoring preload …

What’s the peak bolt load?

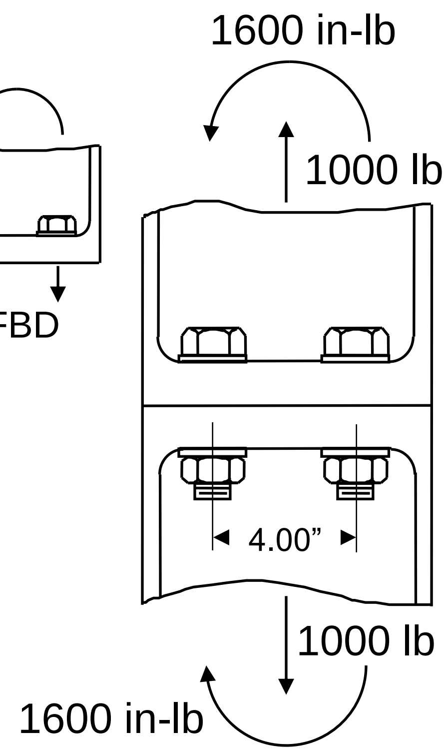
Case 2: Applied moment



With loads carried only by the bolts,

What’s the peak bolt load?

Case 3: Combined loads



What’s the peak bolt load?

Super-position applies

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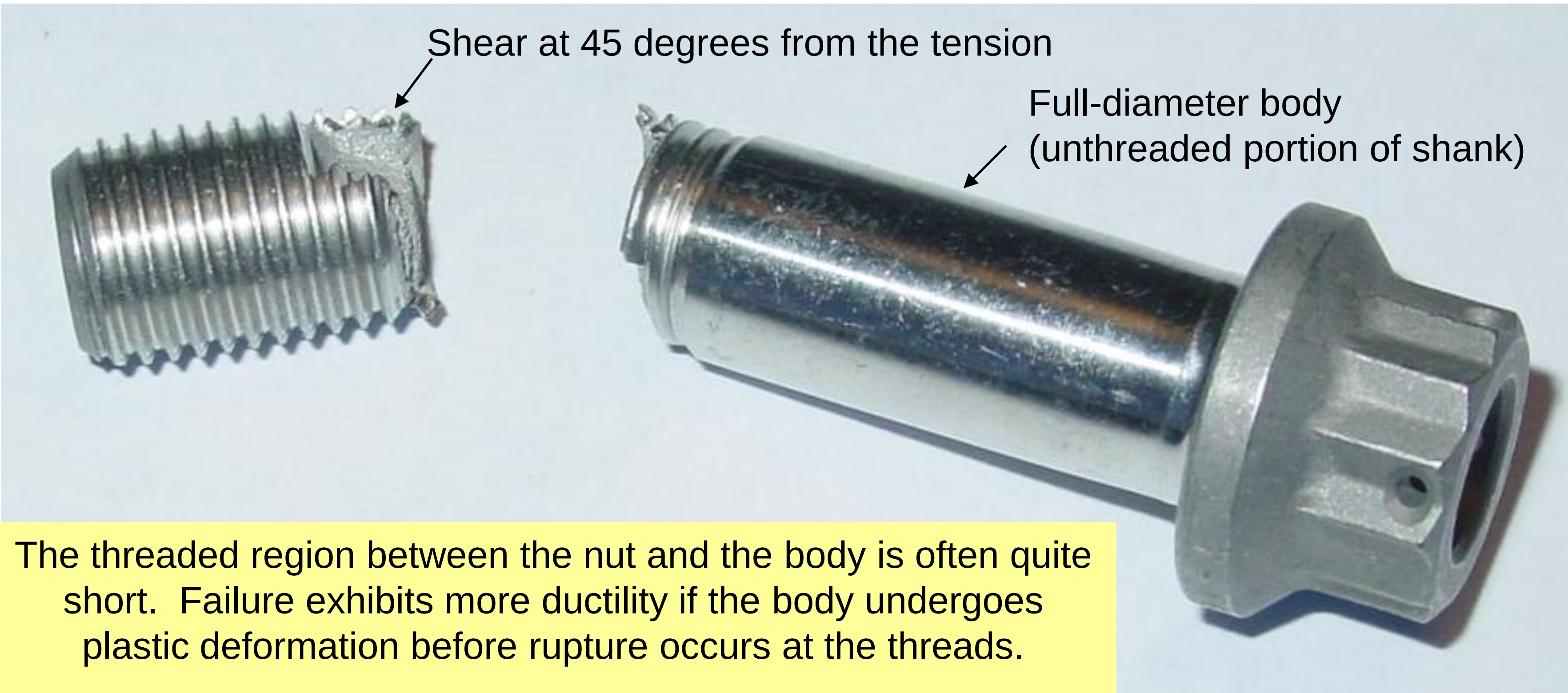
A Bolt that Failed in a Tension Test



The threaded region between the nut and the body is often quite short. Failure exhibits more ductility if the body undergoes plastic deformation before rupture occurs at the threads.

Thus, the most robust bolt is one made of a material that has a large difference between its proportional limit and its ultimate strength. This is not the case with ultra-high-strength fasteners and 160-ksi titanium fasteners.

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Bearing Ultimate Failure for Ductile Materials

The material can undergo very large plastic deformation, elongating the hole.

Rupture is typically shear tearout.

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Design Shear Joints to Be Bearing Critical

- ***Shear-critical joint***: one that would fail first by fastener shear
  - Can have little plastic deformation before failure
  - The highest-loaded fasteners can fail before the others take their share of load
  - Result: the joint can "unzip", as each fastener next to a failed one becomes overloaded and then also fails
- ***Bearing-critical joint***: one that would fail in bearing before the fasteners fail in shear
  - Bearing failure is ductile, so the fasteners will share load before the joint fails
  - Can't get a true bearing-critical joint with composites made of brittle materials

With metal plates, design the joint to be bearing-critical by increasing fastener diameter relative to plate thickness.

Even if the joint is bearing-critical, avoid long strings of fasteners in a splice, such as the one shown on the previous page.

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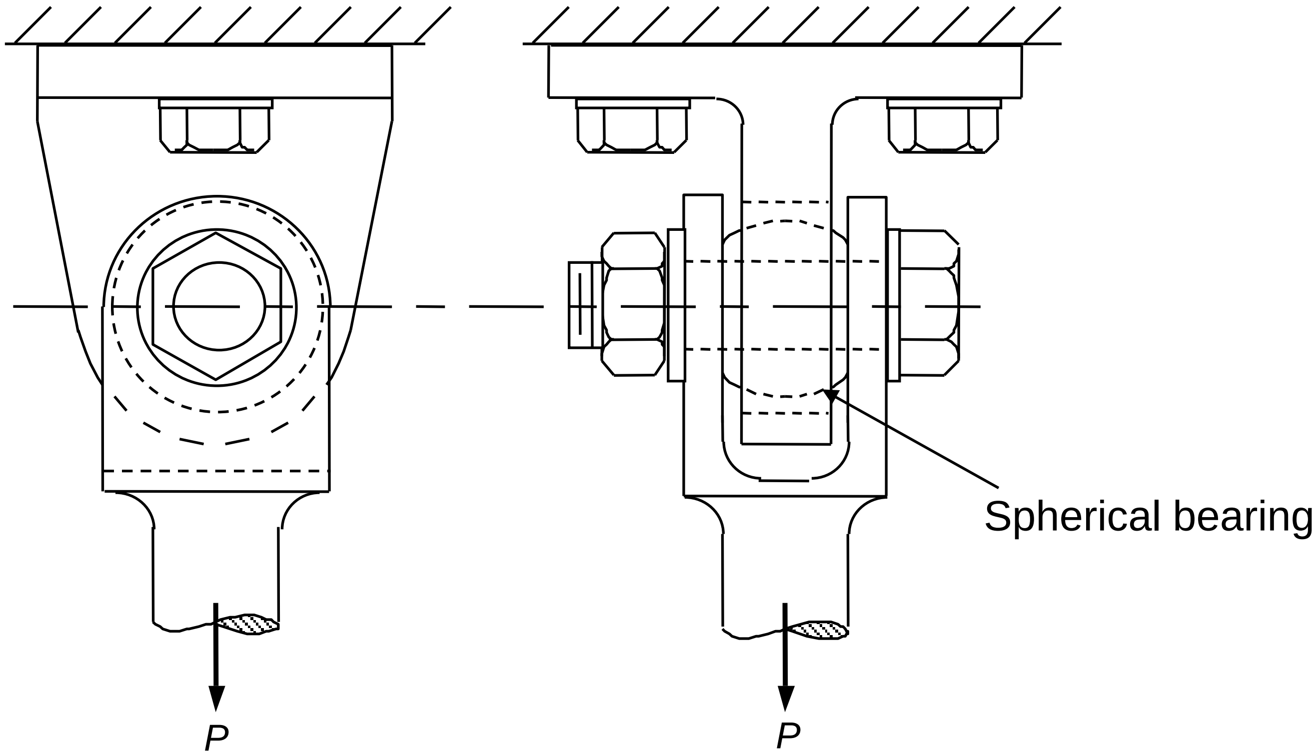
Class Exercise: Recognizing Potential Failure Modes (Problem 1)

A. Identify all potential ultimate failure modes of concern.

B. Explain how you would assess those failure modes and obtain corresponding allowables.

C. How would you improve the design?



Materials: ductile metals

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Examples of Mismatched Hardware

Some combinations in use in space programs (NASA survey results, Dec 2007):

Bolt size	Bolt spec	Bolt material	Bolt mat'l ult tensile strength, Ftu (ksi)	Bolt ult tensile strength, Ptu (lb)	Nut or insert	Nut or insert spec	Nut or insert material	Nut or insert ult strength (lb)	Minimum shear engagem area, in^2	Pullout strength in 6061-T651 (lb)	Pullout strength in 7075-T7351 (lb)
#10	NAS1351	A286	160	**3200**	Keensert	MS51830	Steel	**2500**	0.0945	**2550**	**3590**
#10	NAS1351	A286	160	**3200**	Nut	MS21043	A286	**2460**			
#10	NAS6703	A286	160	**3620**	Nut	NAS1291	A286	**2460**			
#10	NAS6703	A286	160	**3620**	Nut	NAS1805	A286	**4070**			
3/8	ST12007	MP35N	260	**22800**	Keensert	MS51831	A286	**14050**	0.3843	**10400**	**14600**
3/8	NAS1956	A286	180	**17100**	Nut	NAS1805	A286	**17100**			
3/8	NAS6706U	A286	160	**15200**	Nut	MS21043	A286	**11450**			

- Good combinations for strength
  - But some sizes are dimensionally incompatible—discussion forthcoming)
  - And, for the #10, the nut is stronger than the bolt, which may lead to reduced fatigue life
- For all other combinations listed above, the nut or insert is the weak link. Failure may be relatively brittle, and the analyst may not think to check anything other than the bolt.

Note: Strength mismatch may be okay for shear joints with reduced preload

Match your hardware strength-wise!

In analysis, consider the entire fastening system!

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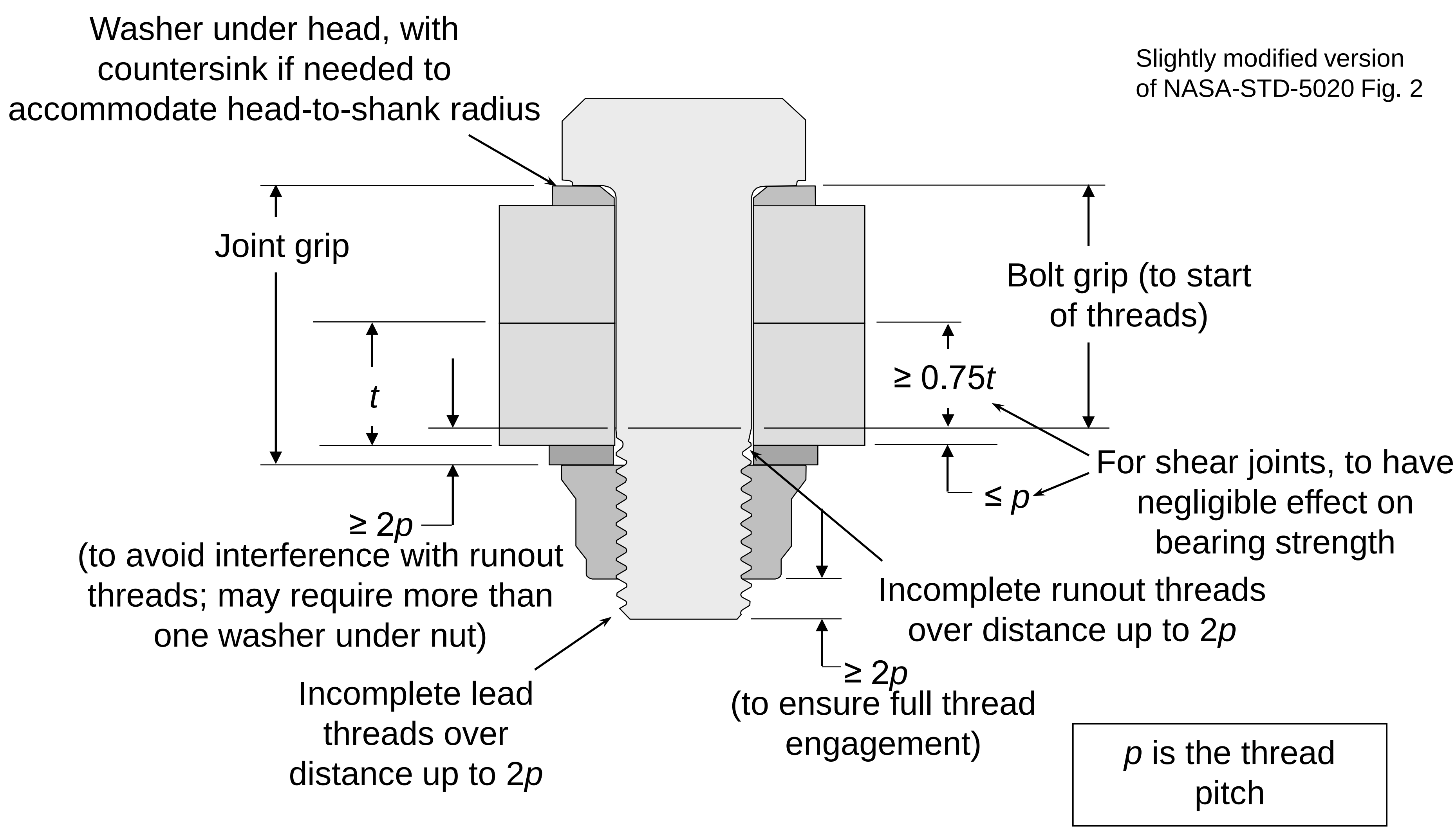
Dimensional Considerations and Washers



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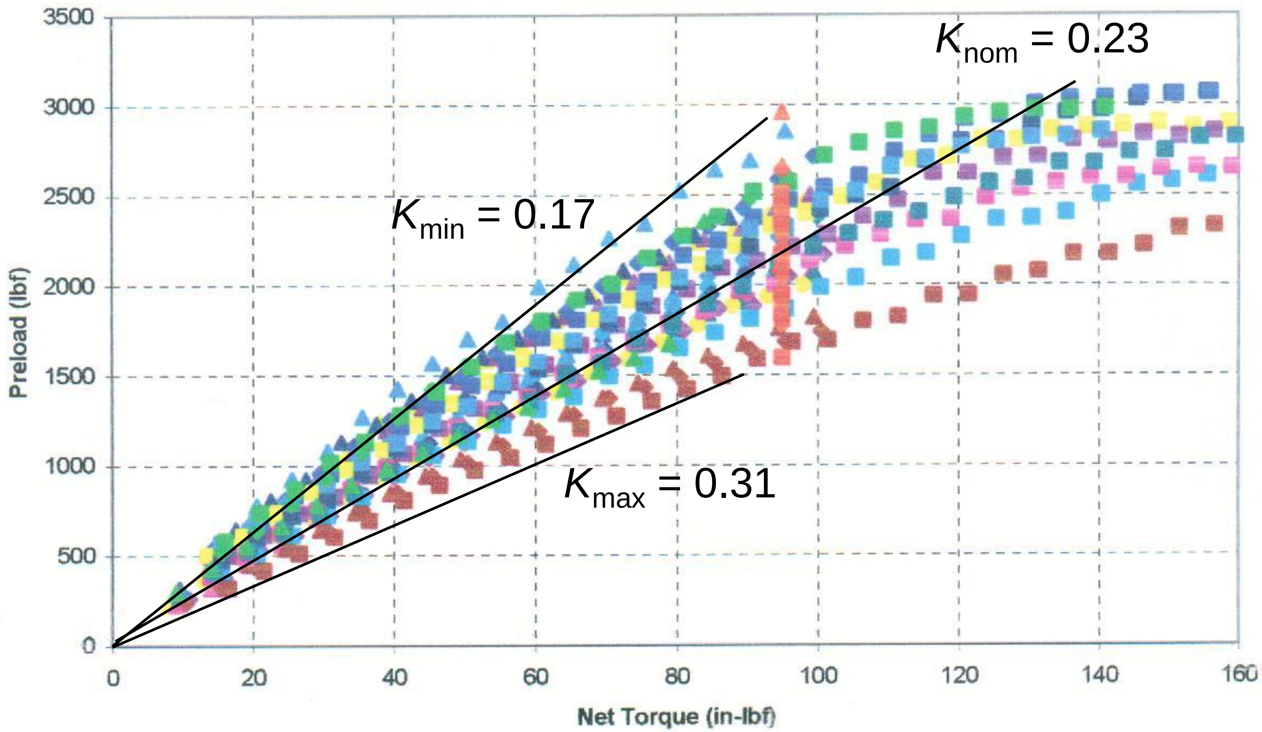
Example of Variation in the Torque-Tension Relationship

Ten screws tested, 3 cycles each, the last cycle to failure (Source: Ref. 30)



Note: These fasteners were torqued from the head. Doing so usually results in higher *K* (less preload per given torque) than when a nut is torqued.

#10 NAS1351, UNRF-3A, A-286 (silver plated) screw in an MS21209 steel helical coil insert, with an NAS620C10L washer. No lubricant. Each screw was installed in a fresh insert for the first cycle. Torque reported is the amount above running torque from the locking feature.

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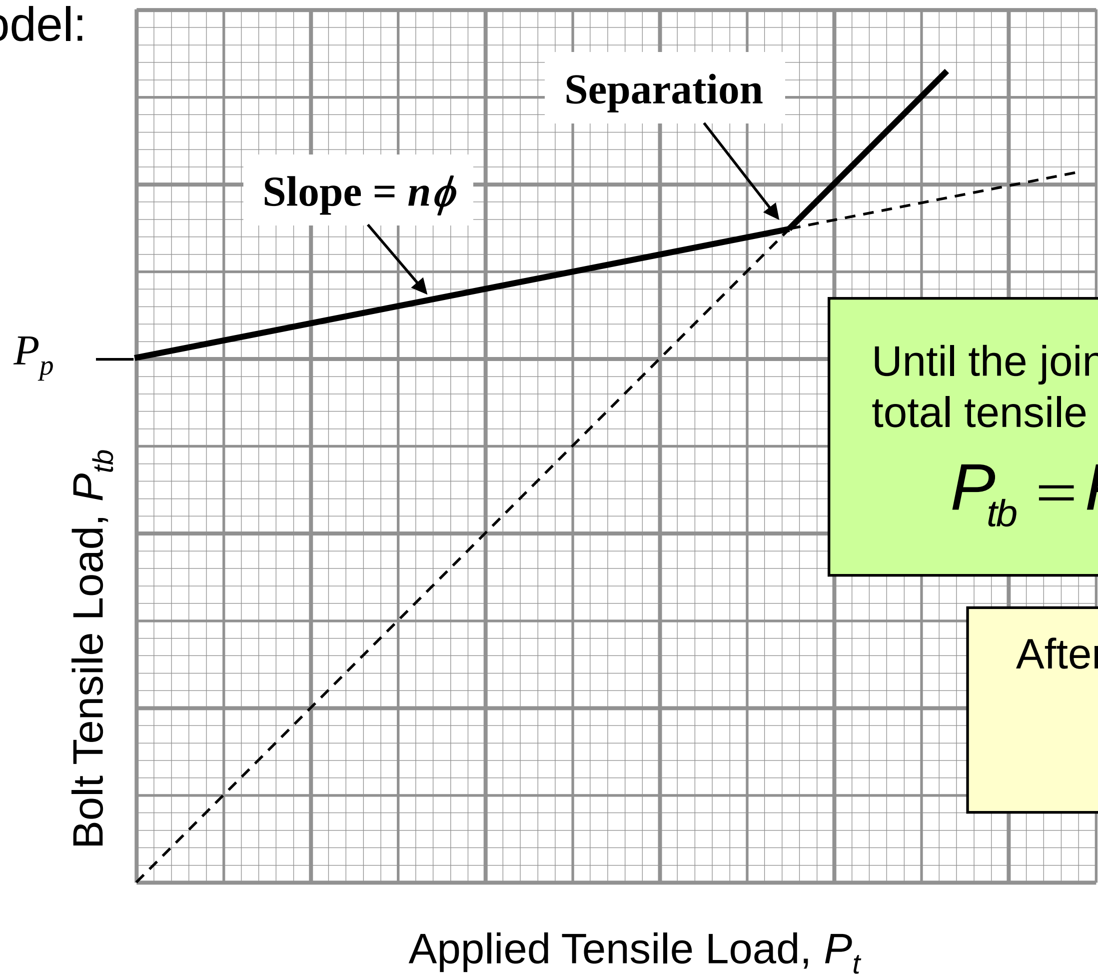
Revisiting the Plot of Applied Load vs. Bolt Load

The linear model:



Until the joint separates, the total tensile load in the bolt is

$$P_{tb} = P_p + n\phi P_t$$

After separation,

$$P_{tb} = P_t$$

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Summary of Tables 8-1 through 8-3

Given:

- Steel bolts, #4 through ½" in diameter, and aluminum joint members
- One steel washer under bolt head and, for bolt-nut assemblies, one steel washer under nut
- 0.5 loading-plane factor (commonly assumed for most joints without spacers)
- For bolt-nut assemblies: total clamped thickness between 1.5 and 4.0 times bolt diameter
- For bolt-insert assemblies: clamped part with through hole has thickness between 1.0 and 2.0 times bolt diameter
- Edge distance for clamped parts is great enough to fully capture the assumed 30° frustums

The traditionally accepted method presented herein for computing how applied tensile load distributes in a preloaded joint results in the following conclusion:

For joints fitting the above description, up until gapping the bolt load will increase by no more than 20% of the applied tensile load

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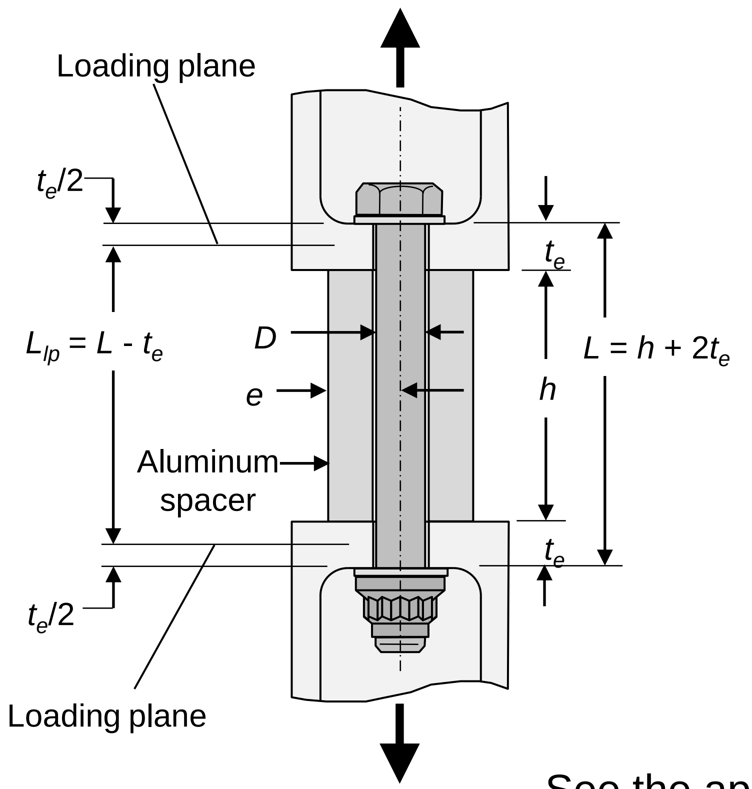
Example Problem 8-2: Extreme Case for Steel Bolts with Aluminum Joined Parts

This is the type of joint for which the loading-plane factor is highest: $n = \frac{L - t_e}{L}$



For a preloaded joint, prior to separation, determine an upper bound for the percentage of applied load by which the bolt load increases, given the following:

- Steel bolt
- Aluminum fittings (joint members)
- $e/D \geq 1.5$ for all clamped parts other than washers
- Load-introduction factor, $n \leq 0.9$

Conclusion:

For a preloaded joint meeting the above constraints, prior to separation, the bolt load will increase by no more than 25% of the applied load.

See the appendix at the end of this section for supporting analysis.

This conclusion was confirmed with two independent, nonlinear finite element analyses.

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Identifying Whether Rupture Can Occur Before Separation

Fig. 9-9

(adapted from NASA-STD-5020 Fig. 5)

$E_c > E_b/3$ and $e/D \geq 1.5$?

No → Rupture may occur before separation. Calculate k_b, k_c, ϕ, and n. (See Sec. 8.) Calculate P'_{tu} and P'_{sep} (Eqs. 6-9 & 6-10), and compute the margin of safety with Eq. 6-6 or 6-11, as appropriate.

Yes ↓

$P_{p\text{-max}} \leq 0.75 P_{tu\text{-allow}}$ and $n \leq 0.9$?

Yes → Separation would occur before rupture. Calculate the ultimate margin of safety with Eq. 6-6.

No ↓

$P_{p\text{-max}} \leq 0.85 P_{tu\text{-allow}}$ and $\delta_p/\delta_e > 0.25$?

Yes → Separation would occur before rupture. Calculate the ultimate margin of safety with Eq. 6-6.

No → Rupture may occur before separation.

$$MS_u = \frac{P_{tu\text{-allow}}}{P_{tu}} - 1 \quad \textbf{(Eq. 6-6)}$$

P_{tu} = design ultimate tensile load (applied load only)

where E_c = elastic modulus of clamped material

E_b = elastic modulus of bolt material

e = shortest in-plane distance from center of hole to edge of any clamped part other than washers

D = bolt nominal diameter

$P_{tu\text{-allow}}$ = allowable ultimate tensile load for the threaded fastening system

$P_{p\text{-max}}$ = maximum preload

δ_p = plastic displacement at rupture

δ_e = elastic displacement at rupture

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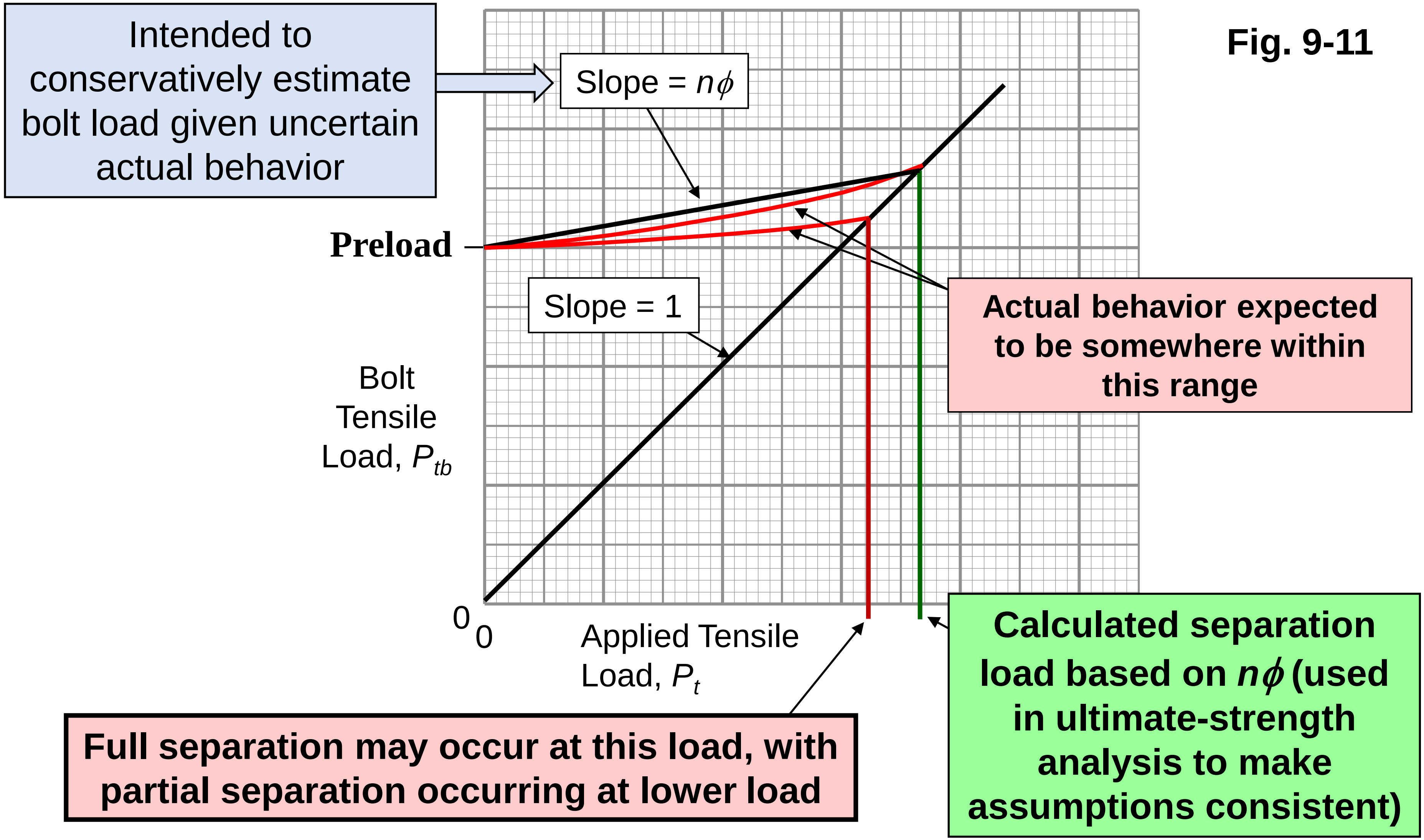
Actual Behavior Compared with the Linear Model



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Is Fastener Yielding Under Design Yield Loads Detrimental?

Fastener yielding under design yield loads may be detrimental if …

- yielding causes the joint to separate under an applied tensile load that is less than the design separation load (see upcoming pages), or
- yielding causes the joint to suffer detrimental slip under an applied shear load that is less than the design shear load, or
- some other design-specific reason exists for why fastener yielding is detrimental
  - not the case for most joints,
  - but you have to understand any unique aspects of your design!

Detrimental for separation

Detrimental for joint slip

See upcoming pages

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Does Preload Reduce the Strength of a Shear Joint?

- With fasteners made of ductile materials, preload does not interact with applied shear to reduce the strength of a shear joint.
  - Plastic deformation in the bolt causes prestress to relax prior to rupture.
- Tests at NASA/Goddard in August 2009 helped substantiate the argument that preload does not reduce the strength of a shear joint:
  - Double shear, steel plates, lubricated with dry-film moly-disulphide
  - 3/8” 180-ksi A286 through-bolts with nuts in holes sized at 0.386 +0.007/-0.002”
  - Tests performed to rupture
    - 5 specimens without preload
    - 5 specimens with extremely high preload (above yield)
  - Preloaded joints carried 1.4% less load on average; statistically insignificant

See NASA-STD-5020 Appendix A.7

- Tests at NASA/Marshall in 2010 (Ref. 42) showed similar results.
  - 3/8” 180-ksi A286 bolts
  - Tests performed with threads in shear plane and with threads not in shear plane

Conclusion: Preload does not need to be included in fastener shear analysis.

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Interaction: Threads Not In Shear Plane

For a fastener whose threads are not in the shear plane, the following interaction criterion applies for combined shear and applied tension:

$$\left(\frac{P_{su}}{P_{su-\text{allow}}}\right)^{2.5} + \left(\frac{P_{tu}}{P_{tu-\text{allow}}}\right)^{1.5} \le 1 \quad \textbf{(Eq. 9.1)}$$

(Equivalent to NASA-STD-5020 Eq. 6-15 when bending stress is omitted)

P_{su} = design ultimate shear load
$P_{su\text{-allow}}$ = allowable ultimate shear load per Eq. 6-12
P_{tu} = design ultimate tensile load
$P_{tu\text{-allow}}$ = allowable ultimate tensile load based on the tensile stress area

Note: The above criterion is more conservative than the traditional criterion, which uses exponents of 3 for shear and 2 for tension per Ref. 3 (Astronautics Structures Manual). Tests* of A-286 bolts at Marshall Space Flight Center in 2010 showed ultimate failure at load combinations well below those allowed by the traditional criterion. Equation 9.1, above, was derived to match the 2010 test results.

No need to include preload in the interaction check, based on results of the Ref. 42 test.

*Ref. 42

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Summary of NASA-STD-5020 (Analysis Portions)

- NASA-STD-5020 …
  - changes how the margin of safety is calculated for preloaded bolts (representing the extent that applied loads can increase before the criteria are not satisfied);
  - provides criteria for recognizing and properly addressing the situations in which preload can contribute to failure or fastener yielding is detrimental;
  - allows use of a joint’s conservatively estimated friction capability for all but ultimate-strength analysis;
  - requires that nominal nut factors be substantiated by test;
  - requires a separation-analysis approach that is more conservative than has previously been used on NASA programs;
  - but allows minimum preload to be less conservatively calculated based on number of fasteners in a joint for separation analysis of non-separation-critical joints and for slip analysis;
  - is well grounded in mechanics and is substantiated by test in most areas.
- Additional testing is anticipated to establish empirical methods of accounting for the effects of bolt bending.

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