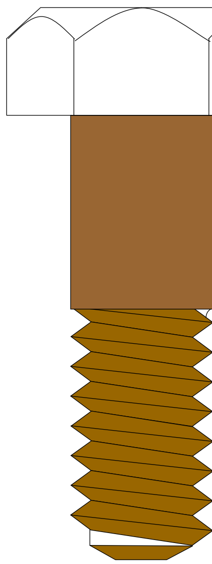
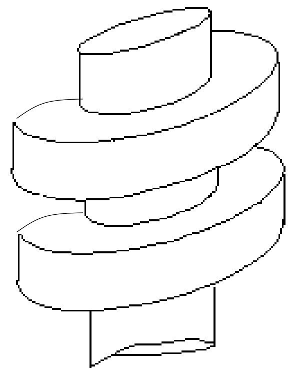
Design and Behavior of Bolted Joints



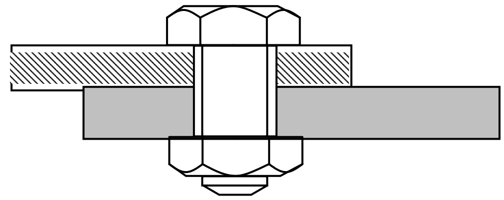


Nutek, Inc. (Since 1987)

Quality Engineering Seminar and Software

Bloomfield Hills, MI, USA. 2014

Design and Behavior of Bolted Joints



Course Description:

Modern buildings, vehicles, machinery, and physical products of all sizes and shapes are put together by joining smaller components with another. A vast majority among these is assembled with fasteners, as they need to come apart for potential repair, replacement, or maintenance. Simply put, a fastener is a screw, nut, bolt or stud with external or internal threads. Although, there are numerous types of fasteners used in commerce, understanding the design and behavior of a threaded joint serves to comprehend the basic principles applicable to all fasteners.

Why should you study fasteners? Approximately 200 billion fasteners are utilized by the industry each year. Many such fasteners play important roles in transportation, safety and comfort of our modern life. A typical automobile, for example, uses about 4000 nuts and bolts. Because a few of them once in a while would come loose, over half of the warranty dollars for the same automobile can be related to fasteners.

In spite of its immense importance, bolted joints are not well understood. Part of the theoretical and empirical relations work fairly well in the design phase. Unfortunately, in installations, that is, in the assembly process, the behavior of a bolted joint depends on a large number of variables that are difficult or impossible to predict and control. Obtaining the desired load and configuration is subjected to a high degree of uncertainty that calls for a greater need for understanding of the operating principles involved. Thus the specialists who design and assemble things which must not fail; airplanes, nuclear reactor, engine connecting rods, engine block heads, all kinds of safety related items in an automobile, etc., must learn all there is to known about the behavior of the joints concerned.

This short session is intended for practicing design and manufacturing professionals who are involved in assembly of electro-mechanical hardware components of any size and shapes with fasteners of all kinds. Attendees are expected to participate in hands-on group exercises and solve a number of problems on theoretical principles discussed in the class (*Attendees are required to bring a scientific calculator*).

Learning Objectives (bullet form):

Upon completion of this seminar, you will learn how to:

- Calculate forces in the fasteners
- Establish what torque to specify
- How to increase functional life of a joint
- Analyze joints and its failure mechanism
- Achieve better control of bolt tension and applied torque in the assembly operations
- Utilize torque application machines more effectively
- Reduce fastener related warranty and rework costs

Instructor's Background

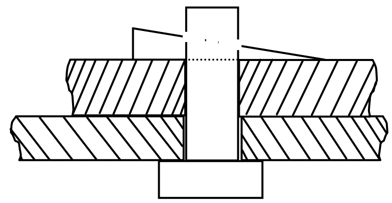
Ranjit K. Roy, Ph.D., P.E., PMP (Mechanical Engineering, president of NUTEK, INC.), is an internationally known consultant and trainer specializing in the Taguchi approach of quality improvement. Dr. Roy has achieved recognition for his down-to-earth style of teaching of the Taguchi experimental design technique to industrial practitioners. Dr. Roy began his career with The Burroughs Corporation following the completion of graduate studies in engineering at the University of Missouri-Rolla in 1972. He then worked for General Motors Corp. (1976-1987) assuming various engineering responsibilities, his last position being that of reliability manager. While at GM, he consulted on a large number of documented Taguchi case studies of significant cost savings. He is the author of the textbooks *A Primer On The Taguchi Method* - published by the Society of Manufacturing Engineers in Dearborn, Michigan, *Design of Experiments Using the Taguchi Approach: 16 Steps to Product and Process Improvement published (January 2001)* by John Wiley & Sons, New York, and of Qualitek-4 software for design and analysis of Taguchi experiments. Dr. Roy is a fellow of the American Society for Quality and an adjunct professor at Oakland University, Rochester, Michigan. Dr. Roy is listed in the **Marquis Who's Who** in the world.

ABOUT THE COURSE

Course Content

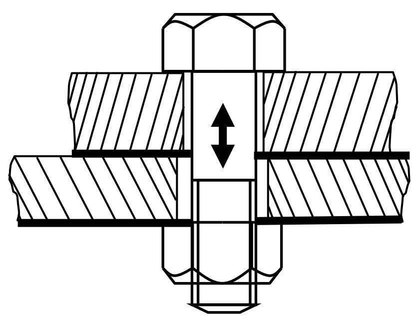
Section - A
Fundamental Principles and Supporting Theories

- Why Study Fasteners?
- Basic Principles of BOLT Operation
- English vs. Metric Units of Measurements
- Physics of Stationery Bodies
- Rigid Body in a State of Equilibrium
- Bolt Load and Free Body Diagram (FBD)
- Effect of Friction
- Friction Forces on Inclined Plane
- Principles of Conservation of Energy
- General Stiffness Principles
- Stress, Strain and Mechanical Properties
- Properties of Engineering Materials
- Equivalent Joint Stiffness
- Effect Of Joint Relaxation On Preload
- How To Minimize Relaxation
- Thermal Effect On Bolt Tension
- Joint Behavior And Geometry Under External Load



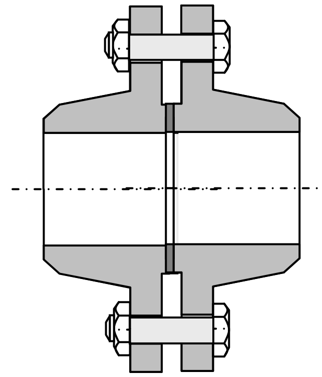
Section B
Fasteners Design Strategies and Assembly Considerations

- Torque and Tension Relationship
- The Short-Form of Torque vs. Tension Relation
- Uncertainty in Assembly Caused By Variability in Nut factor (K)
- Factors That Affect Tension Variability
- Assembly Torque and Tension Behavior
- Process Variation and Process Capabilities
- Primary Influencing Factors Affecting Preload
- Bolt Tightening Strategies
- Three Strategies Commonly Used to Control Preload
- Inspection of Installed Torque
- Bolt And Thread Geometry
- Bolt Identification
- Torque, Angle, and Tension Measuring Devices
- Torque Scatter Due To Tool
- Standardized Torque and Tension Values



- Bolted Joint Design Strategy
- Joint Assembly and Behavior
- Generalized Hooke's Law
- Mechanical Properties of Steel

Section – C



Gasketed Joints for Leak Prevention

- Mechanical Behavior Of A Gasket
- Effect Of Creep And Relaxation On Gasket Behavior
- Example of Creep Relaxation
- Factors that Affect Creep
- Gasket Strenth - The P x T Factor
- Leakage Behavior Of Gasket – m and y Factors
- Gasket Selection
- Simplistic Design & Assmebly Guidelines
- Exercises

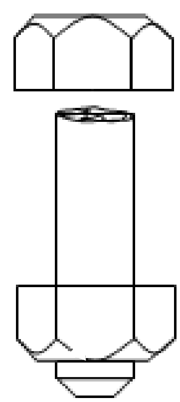
Gasketed Flanged Joint Design

- Objectives and Design Challenges
- Types of Flanged Joints
- Analysis of Flanged Joints – Simplified Model
- Design Steps
- Standards and Codes for Flanged Joints
- Example Flange Joint Design
- Exercise

Section – D

Causes and Prevention of Bolt Failure

- Corrosion
- Essential conditions for Corrosion
- Chemistry of Corrosion
- Strategies for Corrosion Reduction
- Combating Corrosion
- Common Types of Coating
- Commercial Fastener Coatings
- Causes Of Joint Failure
- Mechanical Failure Modes of Bolted Joints
- Prevention of Vibration Loosening
- Mechanical Properties Of Typical Medium/Low



- Carbon And Case Hardened Steels

Appendix

Section – A

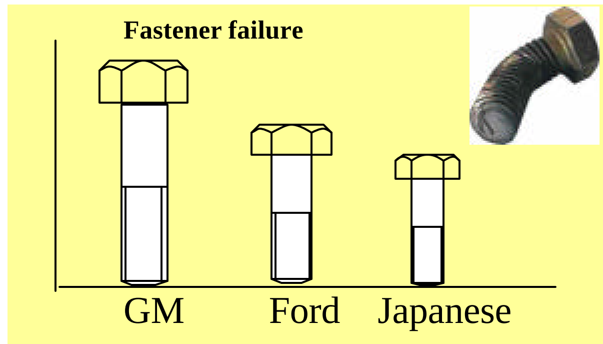
Fundamental Principles and Supporting Theories

Why Study Fasteners?

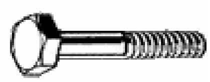
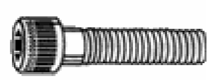
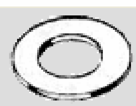
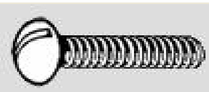
Fasteners is the most common cause of warranty failures.

Bolted joints are the primary means of fastening in automobiles. Typical vehicles like Ford Taurus uses 4000 bolts. About 1,500 are used in body alone.

What is the probability that some of these bolts are loose?



Common Threaded Fasteners (Bolts and Screws)

US Fa	
Wood screws A tapered shank screw for use exclusively in wood. Available in a variety of head styles and materials.	
Hex bolts Hex bolts, also known as hex cap screws or machine bolts.	
Carriage bolts A bolt mostly used in wood with a domed top and a square under the head. This pulls into the wood as the nut is tightened.	
Socket products Socket screws, also known as Allen head are fastened with a hex Allen wrench. Available in 3 head styles and 2 materials.	
Washers Washers provide a greater bearing surface under the fastener. This helps prevent a nut, bolt or screw from pulling through the material.	
Sheet metal screws Highly versatile, used in wood, fiberglass and metal, also called self-tapping screws. Available in steel and stainless steel.	
Machine screws Available in many styles and materials. Also referred to as a stove bolt.	
Lag bolts Often called a lag screw. Hex lag bolts are for fastening in wood. Available in a variety of materials.	
Nuts Used to attach machine thread fasteners.	
US Assortments Pre-assembled US fastener assortments. Available in metal trays plastic boxes.	

Why Bolted Joints?

Bolted joints are primary fasteners used in automobile and other mechanical parts and machinery. Principles and theories that support the behavior of bolt are also applicable to other fasteners (screws, rivets, welds, etc). Bolts are preferred fasteners for the following reasons:

- Easy to remove and re-assemble
- Strong
- Long lasting
- Reusable
- More resistant to corrosion

Safety

- There are over 4,000 fasteners used to put together an automobile (Taurus, Escort).
- Many of these fasteners are also holding the safety components of the vehicle.
  - Recalls
  - Litigations

97 Explorer – Compressor hose, 1982 Corvette – brake pads
79 Buick Century

Warranty
60% of warranty repairs are related to fasteners.
Leaks, Rattles, Loose bolt, Poor attachments, etc.

There are many ways to fasten two or more pieces of materials:

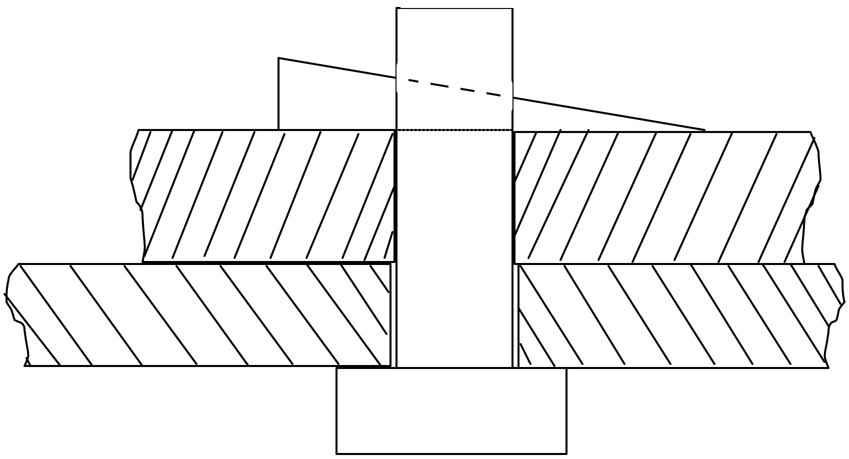
- Weld
- Rivets
- Bolts
- Screws
- Adhesive
- Interference fit

Why are BOLTS AND SCREWS used?
Ease of assembly, Adjustment, Long lasting, Strength, Serviceability, etc.

Also, understanding how BOLTS work helps us understand basic working principles of all fasteners.

Basic Principles of BOLT Operation

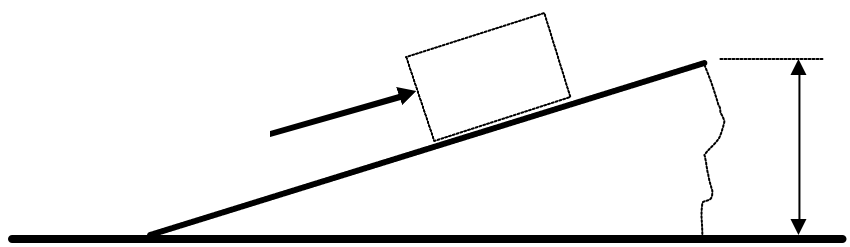
Roman Wedge Bolt



What keeps the two pieces together without sliding?
What keeps the two plates stay tight?

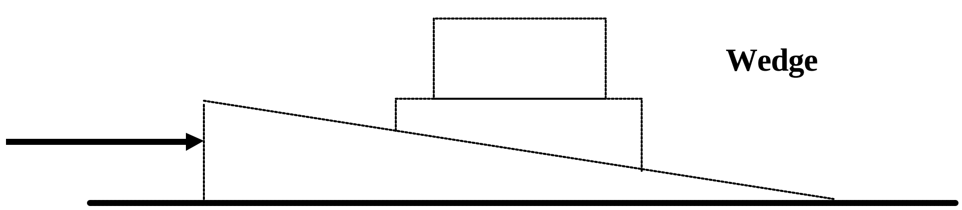
Force (load, tension, preload, compressive force. . .)

What creates the compressive load?
What keeps the wedge in place?



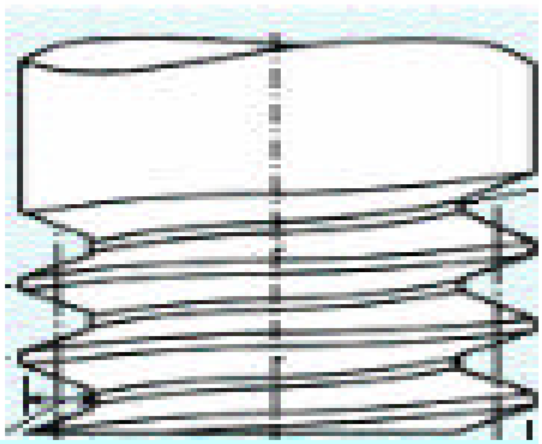
Inclined plane – Wedge

Friction – between Wedge and the contacting surfaces



Wedge

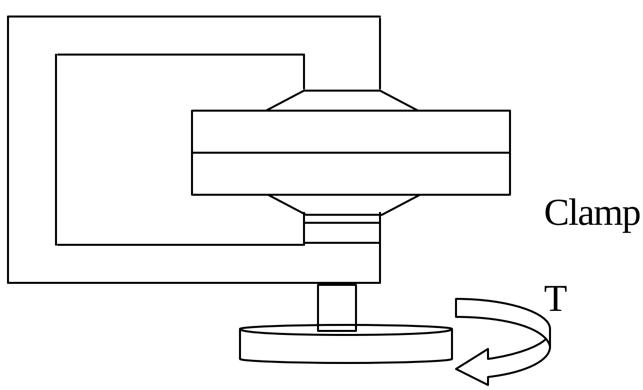
Instead of the WEDGE, modern screw/thread uses SPIRAL/INCLINED surface to produce the load in the bolt.



Spiraled Inclined Surface – SCREW

A turn of the screw moves the screw one pitch distance.

Example: Screw Jack for automobile lifts the vehicle one pitch distance when turned one full revolution.

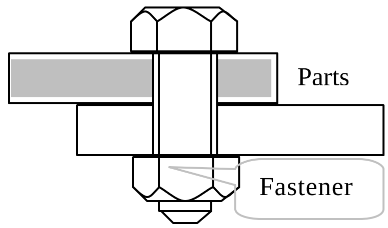


Securing assembly is the objective of any fastener. The key to securing assembly that last for the life of the product is PROPER CLAMP LOAD.

How is CLAMP LOAD obtained?

Correctly designed parts + Proper assembly method = Desired Clamp Load

A joint consists of joining parts and fasteners.

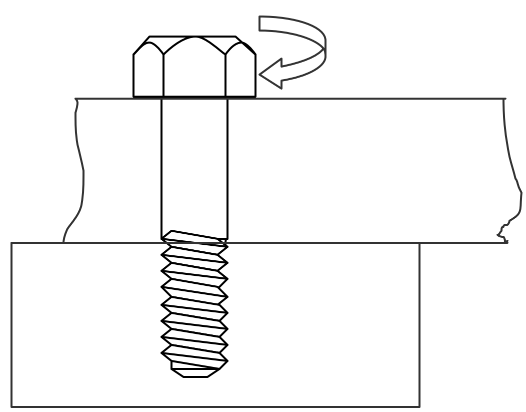


The reliability of a joint depends on three items.

Quality of Parts	Designs of Joint	Assembly Methods
• Fasteners • Joining Parts • Geometry • Friction	Specifications: • Assembly Technique • Preload • Loads/Application • Vibration loads • Relaxation • Tightening Strategies	**Tools:** Performance, Maintenance, Certification, etc. **Operators:** Training and Experience **Quality Control:** Monitoring

Reliable Assembly (Joint)

What happens when a bolt is tightened?



From the structural point of view, the bolt serves one of two functions:

(a) It acts as a PIN to keep the two or more joint members together (prevent slipping relative to each other).
(b) It can act as a heavy spring and clamp the joint members together.

In most applications, the bolt is used as a clamp.

Regardless of the way it functions, the bolt must be tightened properly in order for it to perform the intended function.

When we tighten the bolt (by turning the head in above case or by turning the nut while the head is held fixed), we stretch the bolt slightly. The stretching does not occur, of course, until the bolt has made solid contact with the part. The clamping force is produced only when the bolt is stretched. The bolt therefore, must be stretched to be effective as a clamp.

The stretch creates strain, which then produce tensile stress in the bolt. The tensile load in the bolt causes friction torque between the bolt and the joint surfaces and between the bolt and the joint threads. The torsional stress produced due to friction toque can be as much as 1/3 to ½ the tensile stress in the bolt.

When we stop tightening, a good portion of both the tensile and torsional stresses disappear. This is primarily due to relaxation (10 – 20% in first few minutes) and localized plastic deformation.

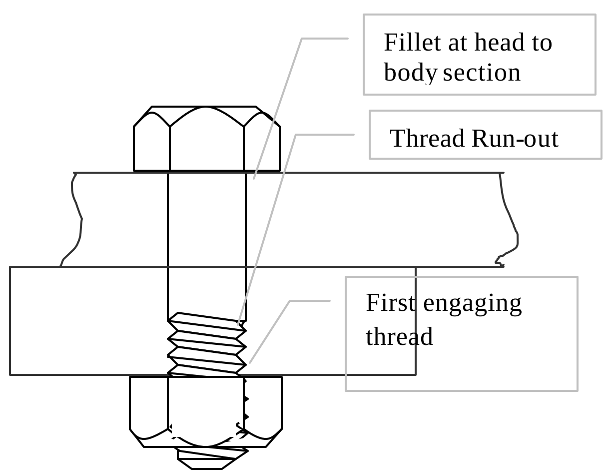
The initial clamping load or tension generated in the bolt (which is called as preload) is the objective of tightening a bolt in application. For economical design of a joint, it is necessary that CORRECT preload is obtained.

Too much preload ⇨ Fastener or joint fails prematurely

Too little preload ⇨ Risks other types of failure (more common)

Where is a bolt most likely to fail?

Bolt material is most likely to fail at locations where stress exceeds the material strength. Commonly, stress concentrations create stress level to rise beyond the average value. There are three points in the bolt, as called out in the joint shown, at which the bolt will usually fail.



- Head to body fillet
- Thread Run-out
- First thread at engagement (In common applications, the first three engaged threads carry most of the final clamp load)

To understand completely the behavior of a bolted joint in terms of its strengths, weaknesses and failure mechanisms, we must first review the applicable principles of:

- Forces on inclined plane
- Friction
- Springs & Stiffnesses
- Behavior of elastic materials
- Thermal expansion
- Mechanics of materials

English vs. Metric Units of Measurements

English:	lbs.	Inch	sec.	psi
Metric:	kg	m	sec.	Pa.

1 inch = 25.4 mm
= .0254 m
l lb. = .4536 kg.

1 m = 39.37 inch.
1 Kgf = 2.20 lbf.

<u>Other Useful Conversions</u>

1000 ft = 304.8 m
= 0.1894 miles

1000 lbs. = 453.6 Kg
= 14580 Troy oz

1000 ft.lbs. = 323.8 cal
= 0.37 W-hr.
= 1356 Joules
= 1.285 BTU

Force exerted by 1 lb = 1 lbf.
Mass of 1 lb = 1/g = (1/32.17)

Force = mass x accl.
1 lb. force = 1 lb.-m x 32.2ft/sec^2
1 kg. Force = 1 kg. x 9.81 m/sec^2
= 9.81 Newton

1 lb. Force = .4536 x 9.81 = 4.45N
1 N = .225 lbf.

g = 9.81 m/sec^2 = 32.17 ft/sec^2= 386.4 in/sec^2

Kinetic Energy Formula = ½ M V^2
M = 100 kg. = 220.46 lbs.
V = 20 m/sec. = 65.617 ft/sec.
K.E = 0.50x220.46x65.617^2/32.17 = 14,753 ft.lb.
= 0.50x100x20 = 20,000 Joules

====================================

VOLUME
1000 cm^3 = 1 Liter = 200 Tea Spoon = 67.62 Table Spoon
= 1.057 Quart = 2.11 Pint = 33.81 Fluid Ounce

ENERGY
1 Calorie = 1/100 of heat required to raise 1 gram of water from 1 – 100^{o}C at atmospheric pressure.
= 4.1868 Joules

Example Conversions:

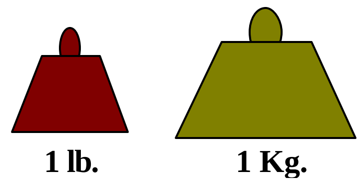
1 lbs/in = [.4536 x 9.81 N/m^2] / [.0254 m/in = 175.2 N/m

1 psi = [1 x .4536 x 9.81] / (.0254)^2 N/m^2 = 6.894 x 10^3 pa

1 inch/sec = .0254 m/sec

1 in.lbf = .0254 x (.4536 x 9.81) = .113 Nm

1 horsepower = 745.7 watt



Types of Loads

Loads are always represented by a force vector in a Free Body Diagram.

Causes of loads:

- gravitational (weight)
- mechanically applied
- inertial
- magnetic

Method of load transfers:

- Body-Contact loads

  Any force transmitted to a body from another by means of direct contact over an area.

  - Point
  - Line } always normal to the contact surface.
  - Distributed

- Friction load/force

  This force acts parallel to the contact area and is a function of normal load, the materials in contact and the relative velocity of the contacting surface.

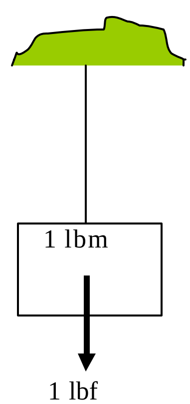
Physics of Stationery Bodies

Force = mass x accl. (Newton's law).

1 unit of force (Newton) = 1 kg. 1 m/sec^2
1 lb. = force exerted on 1 lb. mass by gravity.

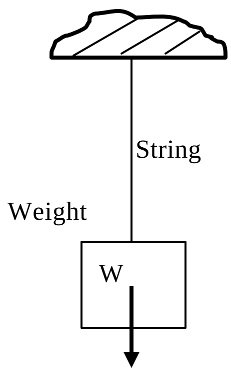
Force is designated by a
VECTOR ↓↑

(A line with an arrow).
A vector had a magnitude ar
a direction.



Free Body Diagram (FBD)

FBD is a graphical representation of a body with its supports/attachments replaced by force.



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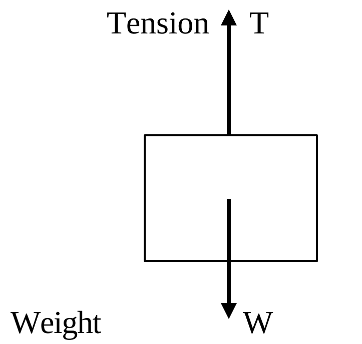

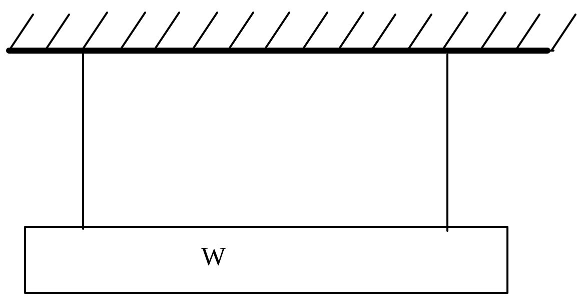

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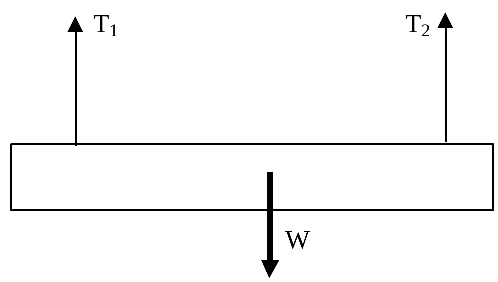

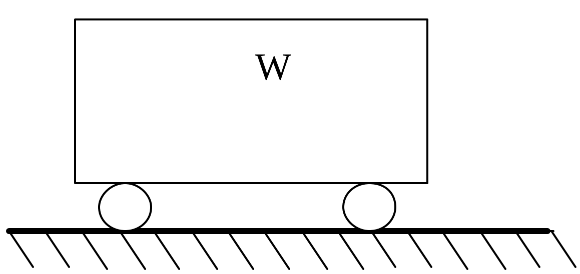

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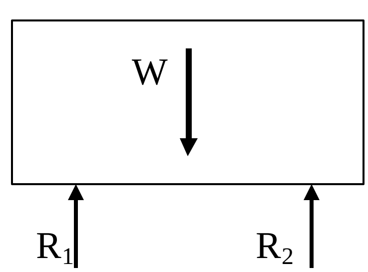

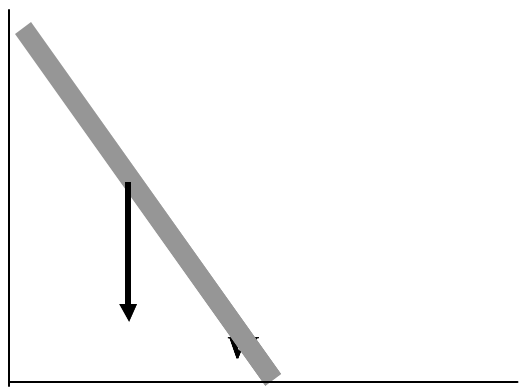

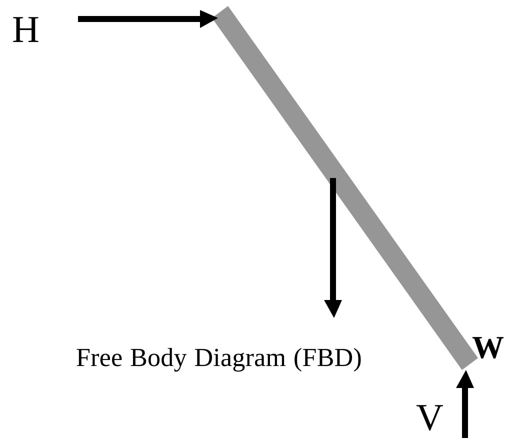

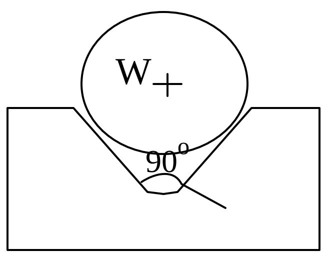

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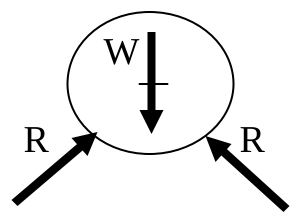

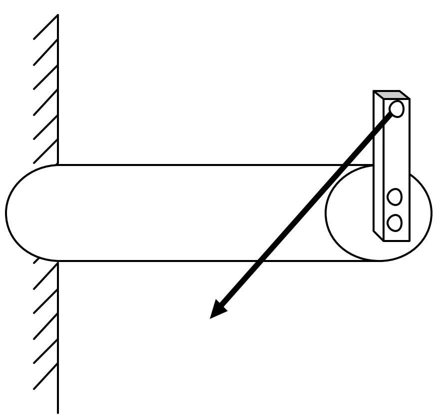

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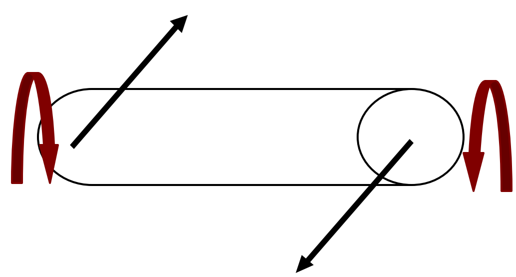
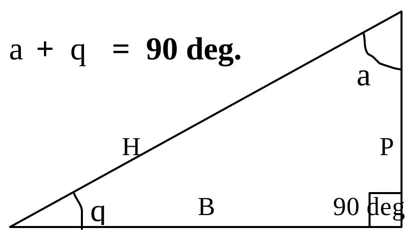


Trigonometry

Need: Break forces into components.

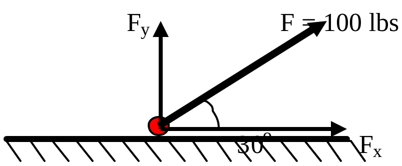


$B = H \cos\theta$

$P = H \sin\theta$

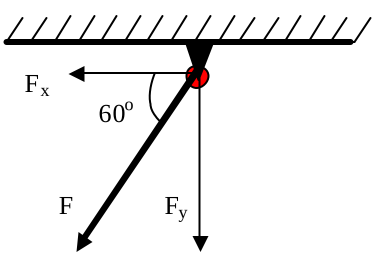
Also $H = \sqrt{P^2 + B^2}$

HOW IS THE ANGLE (q) DEFINED?



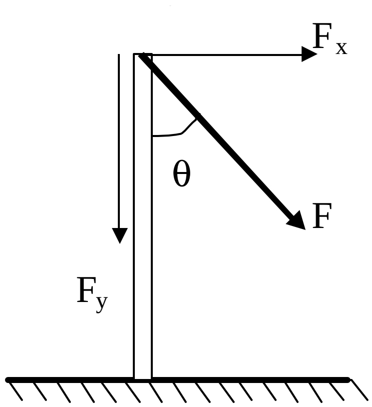
$F_x = 100 \cos 30°$

$F_y = 100 \sin 30°$



$F_x = F \cos 60°$

$F_y = F \sin 60°$

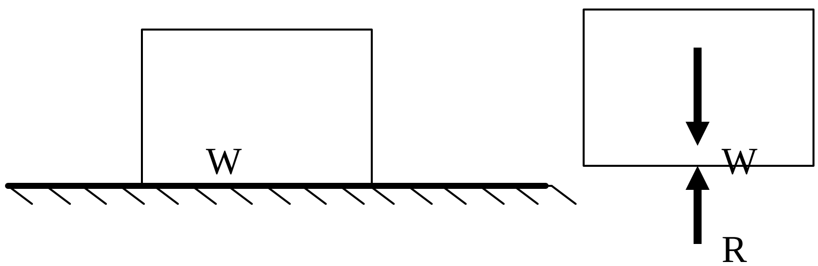


$F_y = F \cos\theta$

(Cos θ or Sin θ depends on the angle selected)

Rigid Body in a State of Equilibrium

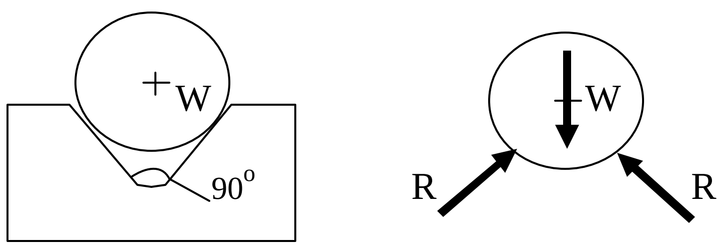
Sum of forces in a particular direction must equal zero.



$\Sigma F_V = 0 \uparrow +$

$R - W = 0$

Or $\mathbf{R = W}$

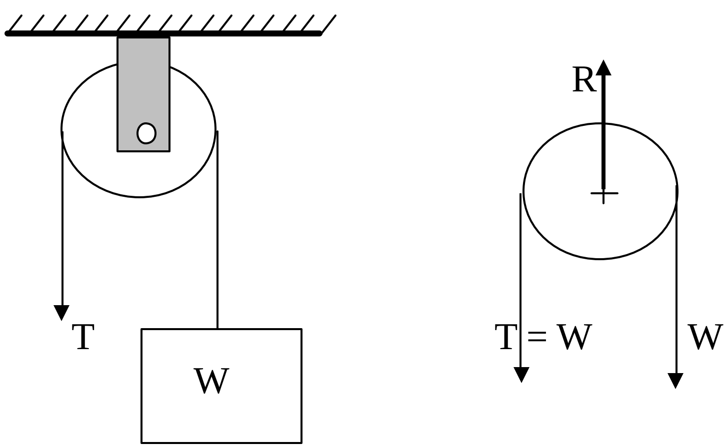


$\Sigma F_V = 0 \uparrow +$

$2R \cos 45 - W = 0$

or

$\mathbf{R = W / 2R \cos 45}$



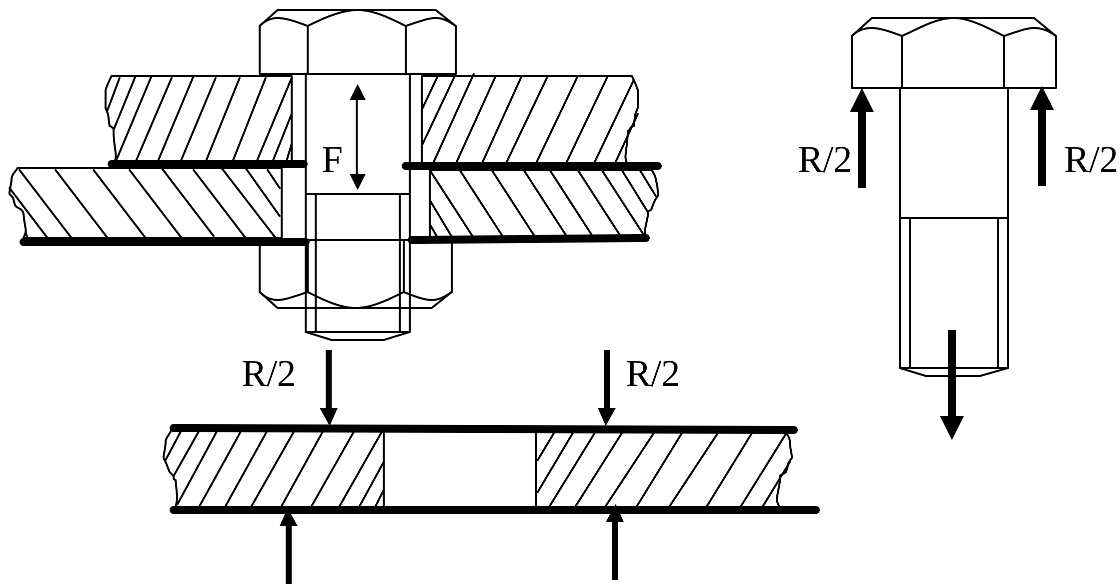
$\Sigma F_V = 0 \uparrow +$

$\mathbf{R - 2W = 0}$

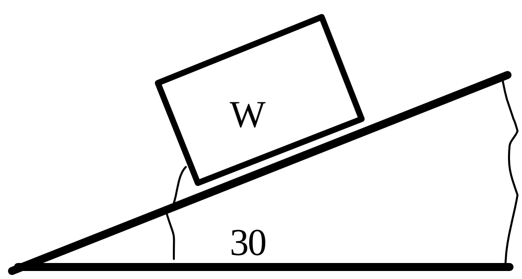
or

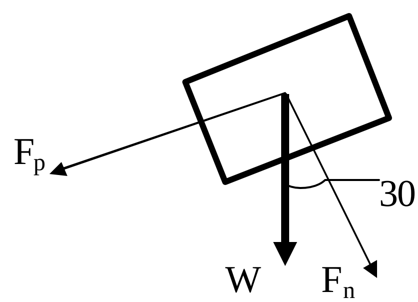
$\mathbf{R = 2W}$

Bolt Load and the Free Body Diagram (FBD)



$F_n = W \text{ Cos } 30$ and $F_p = W \text{ Sin } 30$

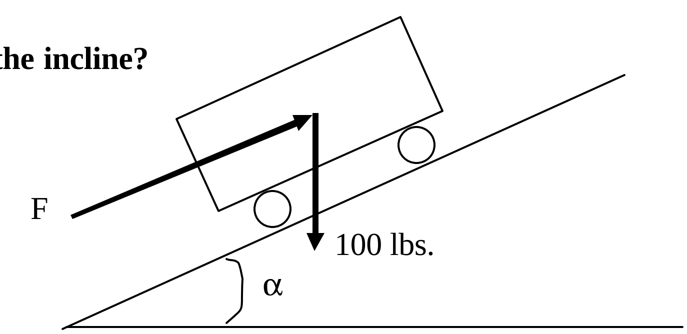




Force (F) required to push W up the incline?

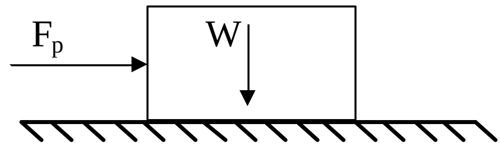
F = F Sin α = 100 Sin 30

Or F = 50 lbs.

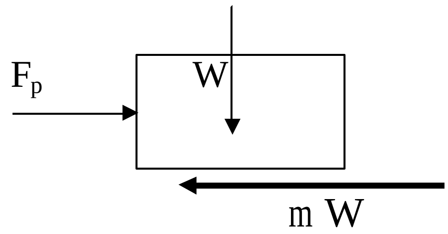


Effect of Friction

Frictional force µ normal load



F_P = force required to slide the block
W = Weight of block
µ= friction coefficient between the surfaces



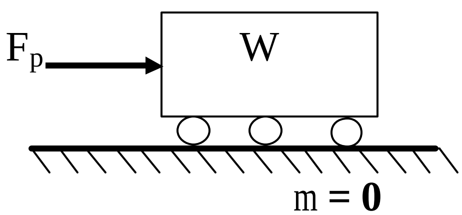
Friction force always opposes the force trying to move the body.

$\Sigma F_H = o \rightarrow +$

$F_P + mw = o$

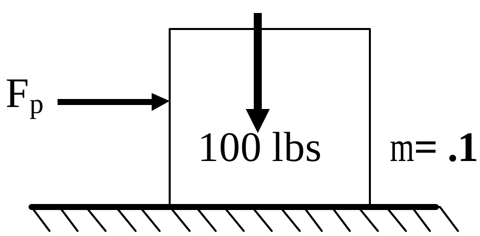
or

$$\boxed{\mathbf{F_P = \mu w}}$$

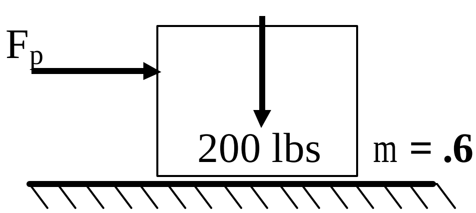


$F_P = o = mw$

Or $m = o$ no friction



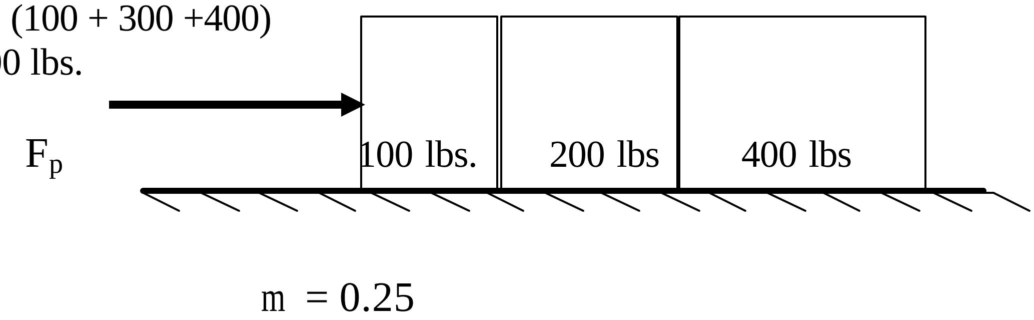
$F_P = mw = .1 \times 100 = 10$ lbs.



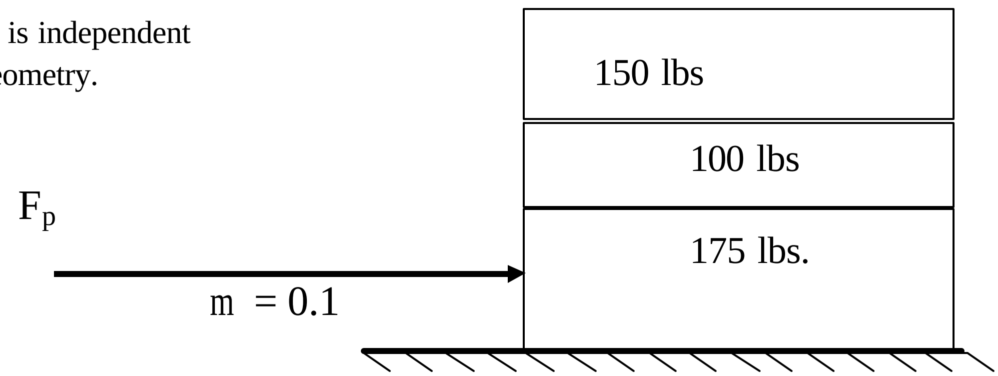
$F_P = .6 \times 200 = 120$ lbs.

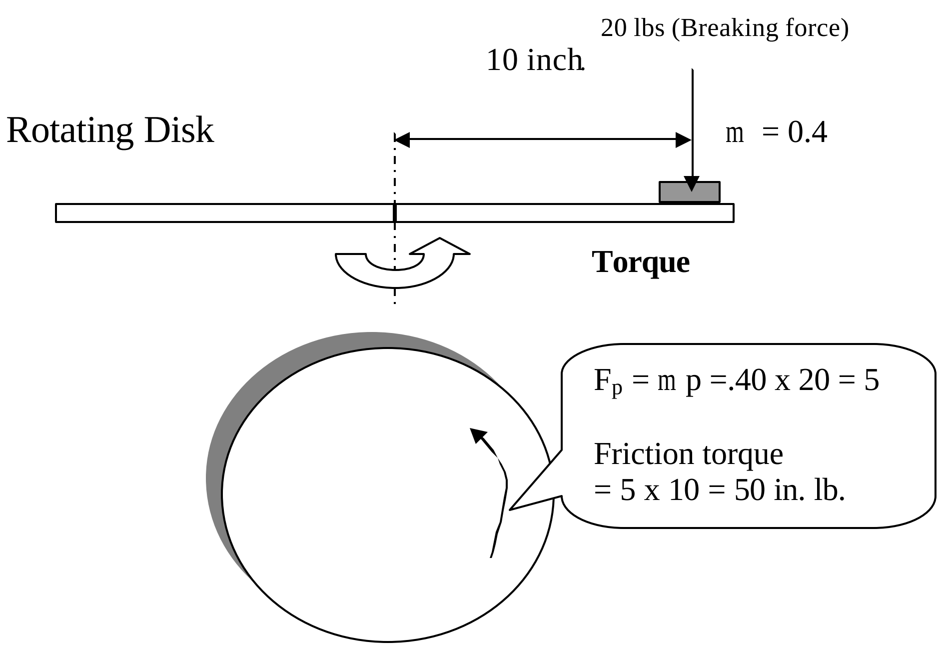
Additive Friction Effects

F_p = .25 (100 + 300 +400)
= 200 lbs.

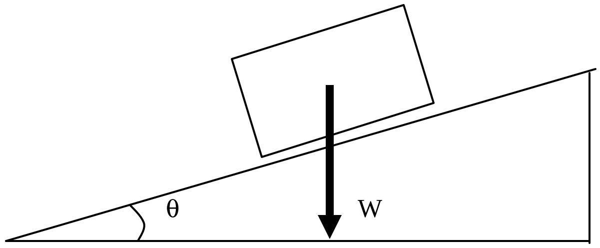


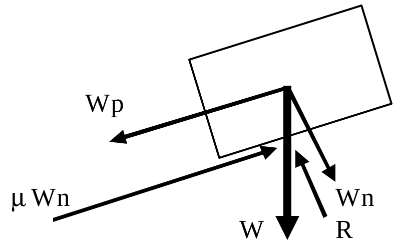
Friction force is independent
of size and geometry.





Friction Forces on Inclined Plane





$\Sigma F_P = 0 \rightarrow +$
$\mu W_N - W_P = 0$

$W_N = W \text{ Cos } \theta$
$W_p = W \text{ Sin } \theta$

or $\quad \mu W \text{ Cos } \theta = W \text{ Sin } \theta$

Or $\boxed{m = \textbf{tan}\ q}$

- This is true when the body just begins to slide.

HOW IS μ (static/dynamic) DETERMINED?

Example:

W =500, μ =0.80, and θ = 30 degrees. (In figure above)

Wp = 500 Sin 30 = 500 x ½ = 250 lbs. and

Wn = 500 Cos 30 = 500 x 0.866 = 433 lbs.

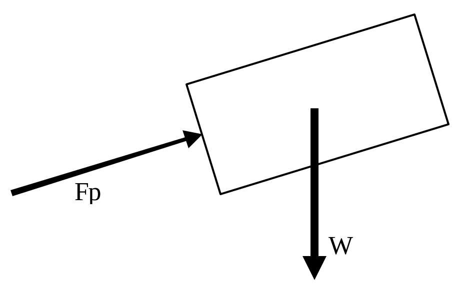
Friction force μ Wn = 0.80 x 500 Cos 30 = 0.8 x 500 x .866 = 346.4

a) Force needed to slide the body up

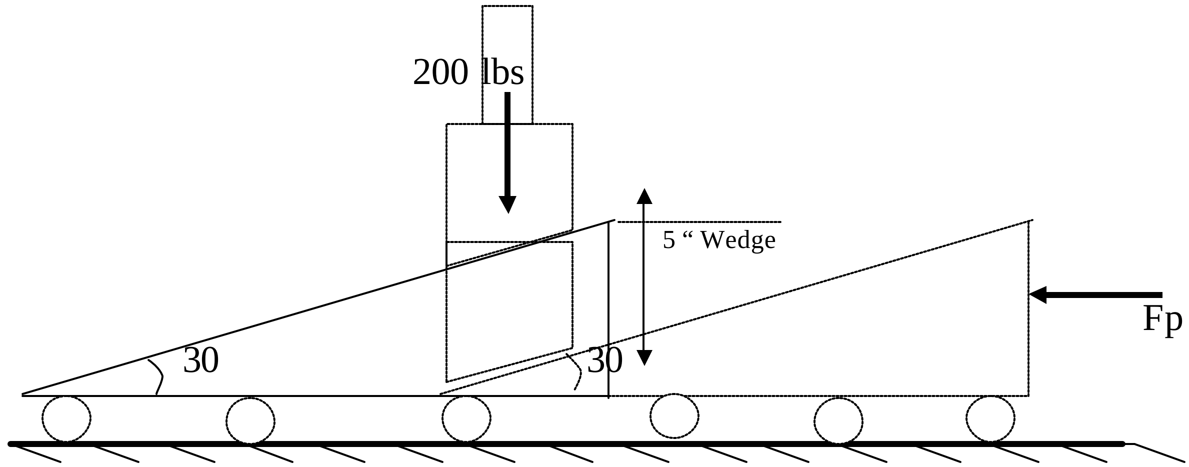
$F_P = 250 + 346.4 = 596.40$

b) Force needed to slide down

$F_P = 345.4 – 250 = 96.40$ lbs.



Principles of Conservation of Energy



Work done in lifting weight = 200 x 5 = 1,000 in lb.

Tan 30° = 5"/H or H = 5/Tan30° = 8.66 in.

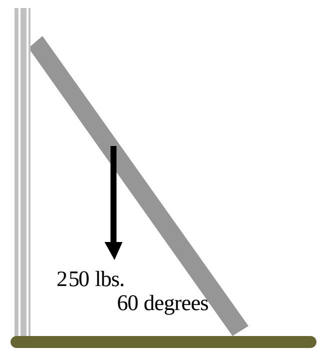
Work done by F_P in moving the block/wedge (Assume no friction at contact surface.)

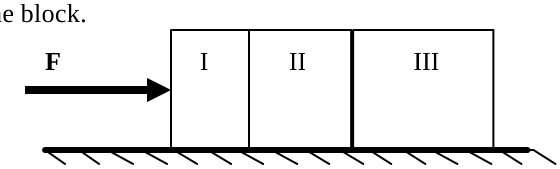
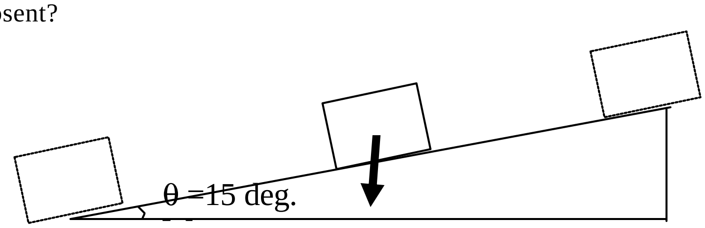
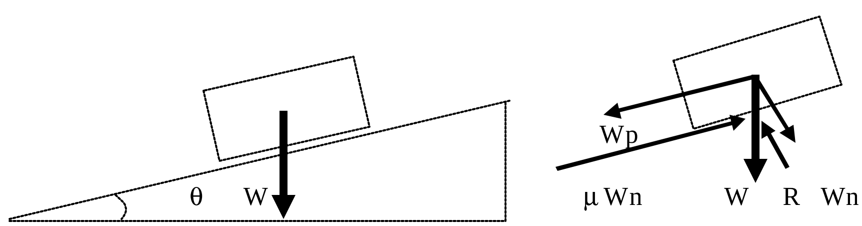
= F_P x H = F_P(8.66)

Work done by the wedge is same as the potential energy gained by the weight

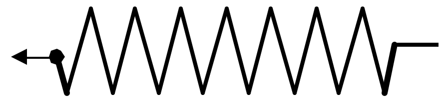
Or F_P 8.66 = 1,000 in lb.

Or F_P = 1,000/8.66 = 115.4 lbs.

Topics	Group Exercises
Units/ measure 1 in = 25.4 mm 1lb = 453.6 gm 1 kg = 9.81 N 1 lf = 4.45 N	**A-1** **a:** The diameter of the Earth is about 4,000 miles. What is the value in KM? **b:** The yield stress of a material is specified as 206.82 mpa (Mega Pascal, N/m^2). What is the value expressed in psi. (Ans: 30,000 psi) **c:** The tension in a string is specified as 13,350 N. What is the tension value in terms of lbs force? **d:** If the acceleration due to gravity is 32.2 ft/sec^2., show that the value in metric system is approximately 9.81 m/sec^2.
Trigonometry Sin 0 = 0 Sin 30 =0.50 Sin 45 = 0.707 Sin 60 = 0.866 Sin 90 = 1.0	**A-2** **a:** A 5,500 lbs vehicle is skidded into a ditch that has 45% ramp. What is the force parallel to the ramp the toe truck has to apply (parallel to the ramp) to pull the vehicle out of the ditch? (Ignore road/dirt friction). **b:** The guy wires (60 degrees with horizontal) for a television tower are set at 25,000 lbs of total tension. Determine the axial force sustained by the tower? **c:** A 20 ft. long ladder is to carry 250 lbs. (W) load at its center, is placed against a wall at 60 degrees angle with ground. If the vertical wall can only support 65 lbs. of horizontal load, is it safe to rest the ladder against it? 

Topics	Group Exercises
Friction $F_P = \mu w$	**A-3** **a.** Three blocks of different weights and friction coefficients, rest on a rough surface. Determine the force, F, necessary to push the block. I Steel 300 lbs. Friction $\mu = 0.25$ II Cast Iron 400 lbs. Friction $\mu = 0.40$ III Wood 250 lbs. Friction $\mu = 0.20$ 
	b. What is the work done in pushing a 5,000 lbs. vehicle up the hill (15 deg.) 200 ft. if the friction is assumed to be absent?  If the friction between vehicle and the road, $\mu = 0.10$, what is magnitude of the force necessary to pull the vehicle up the hill?
At equilibrium $W_N = W \cos\theta$ $\mu W_N - W_P = 0$ $W_p = W \sin\theta$ $\mu = \tan\theta$	**c.** The friction coefficient (μ) is dependent on the contacting materials and can be easily determined in laboratory environment. In the following set up, the block just begins to slide down the incline when the ramp angle is increased to 26 deg. Determine the coefficient of friction between the block and the ramp materials. 

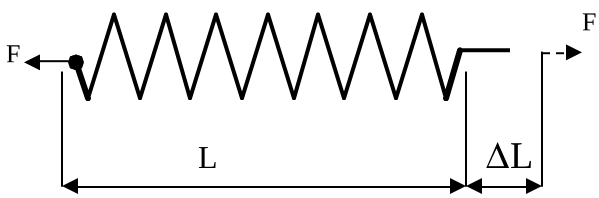
General Stiffness Principles



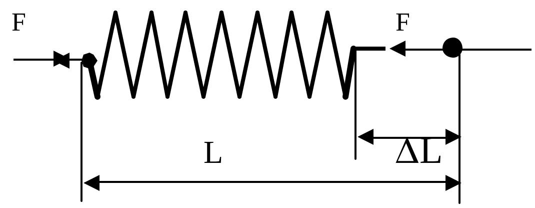
Spring of Stiffness K, Extension ΔL, Force F.
(Length L)

F = K(ΔL), Force required to extend/compress by ΔL

Spring under extension



Spring under compression



F = KD**L** Or ΔL = F/K

Also **K = F/**D**L**

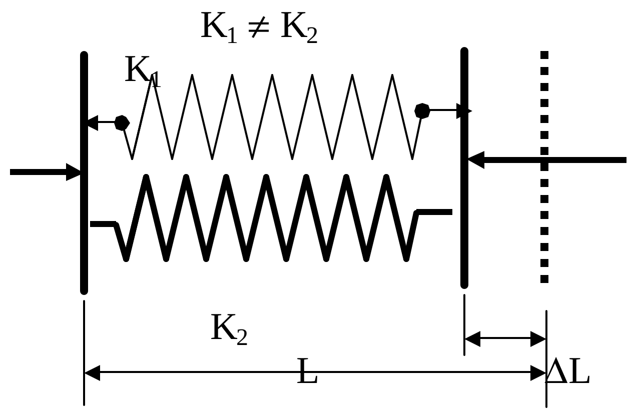
Any of the above three expressions apply to spring properties (elastic extension or compression). Given any two, the third can be found.

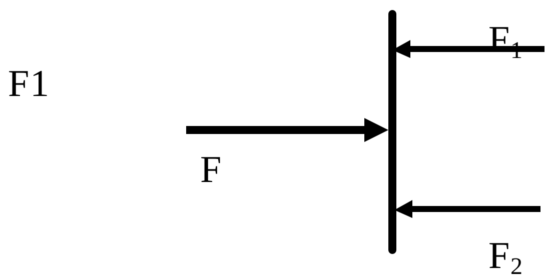
- Find stiffness for 100 lbs. Force and 5” extension.
  Ans.: K = F/ΔL = 100lb./5 in. = 20 lb./in.

- Find elongation when 30 lb. is applied to a spring of stiffness 10 lb./in.
  Ans.: ΔL = F/K = 30 lb./10lb./in. = 3 in.

Equivalent Stiffness

Springs in parallel





$F = F_{1(\text{spring 1})} + F_{2(\text{spring 2})}$

$= K_1\Delta L + K_2\Delta L$

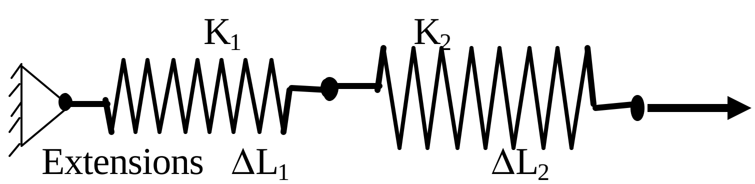
or $F = (K_1 + K_2)\, \Delta L$

or

(or system) $\boxed{F = K_{equiv.}\, \Delta L}$

$\boxed{K_{equiv.} = K_1 + K_2}$

Springs in Series



$\Delta L = \Delta L_1 + \Delta L_2$

final extension under load F

Observation: Both spring carries $\Delta L_1 = F/K_1$

The same load F $\Delta L_2 = F/K_2$

Since total deflection is the result of individual spring deflections:

$\Delta L = \Delta L_1 + \Delta L_2 = F/K_1 + F/K_2$

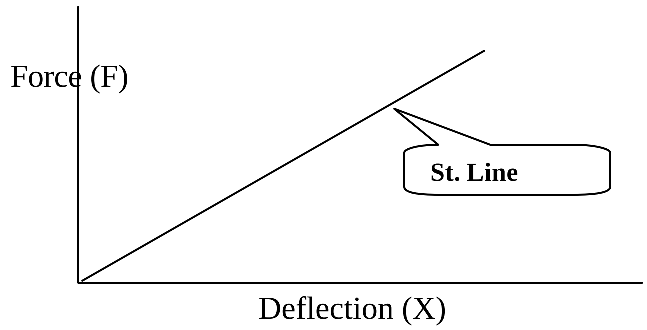
$= F(1/K_1 + 1/K_2)$

or $\Delta L = F/K_{equiv.}$ where $\boxed{1/K_{equiv.} = 1/K_1 + 1/K_2}$

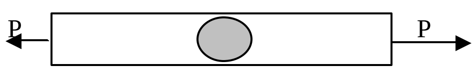
Stress, Strain and Mechanical Properties of Materials

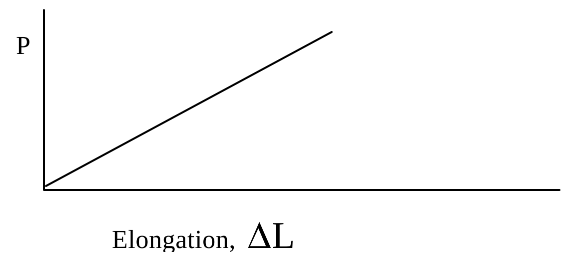
Spring

Force (F)
Deflection (X)



Rod





A rod or any other rigid body under load also behaves the same as springs.
(up to a point)

For materials like steel, Aluminum, Plastic, wood, etc. plot of stress And strain yields some values that Hold true regardless of geometry.

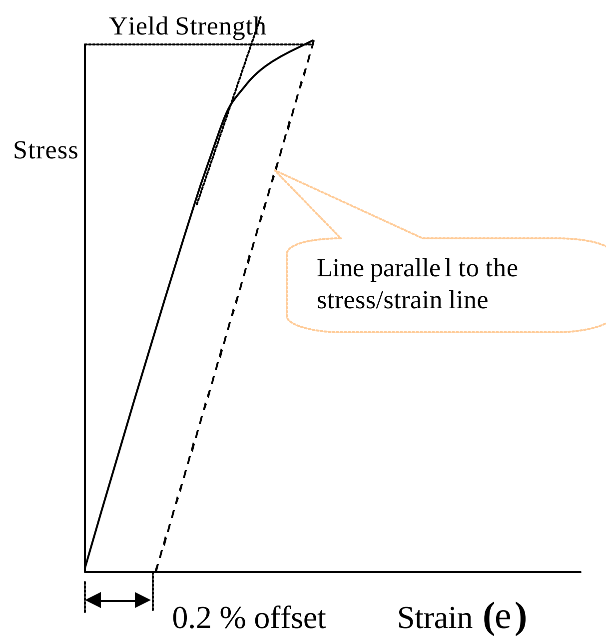
Stress = Load/area (P/A) (σ)
Strain = Deformation/Length ($\Delta L/L$)

Many materials exhibit ELASTIC (like spring) behavior shows:

Stress/Strain = Constant* = E

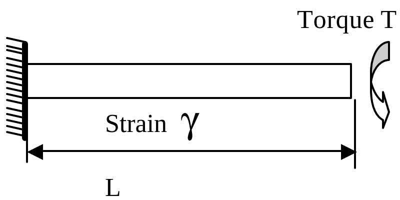
E = Young's Modulus.

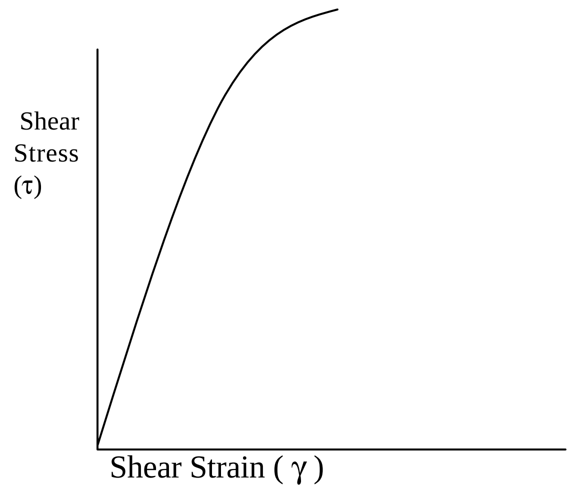
HOW IS IT DETERMINED?



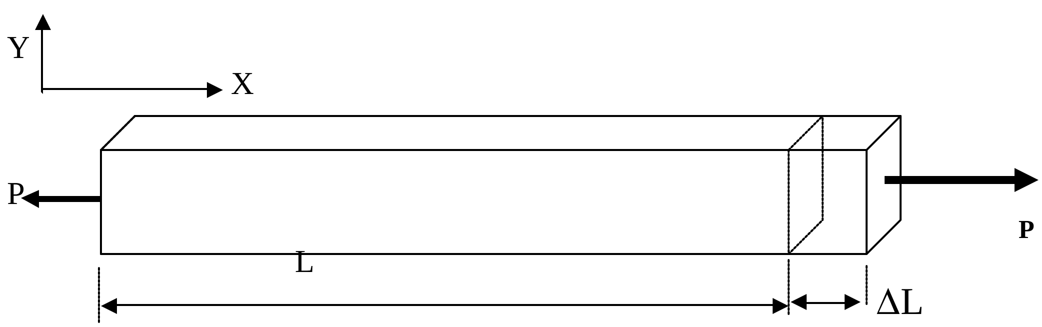
*We encounter many constants in engineering and science. Some are universal, others hold true for a group or type of subjects. And there are some that are only fixed for a particular condition. E is such a constant for a material. ($E = 30 \times 10^6$ psi regardless of size, shape or <u>hardness</u>)

Similarly, if a rod is subjected to torsion, its torsional strength and property can be determined.





Shear Stress/(↑)/Shear Strain (g) = G, Rigidity Modus



Strain along X causes proportional strain in the other two perpendicular directions.

$\varepsilon_y = -\mu\varepsilon_x$ and $\varepsilon_z = -\mu\varepsilon_x$

μ_P = Poisson's ratio

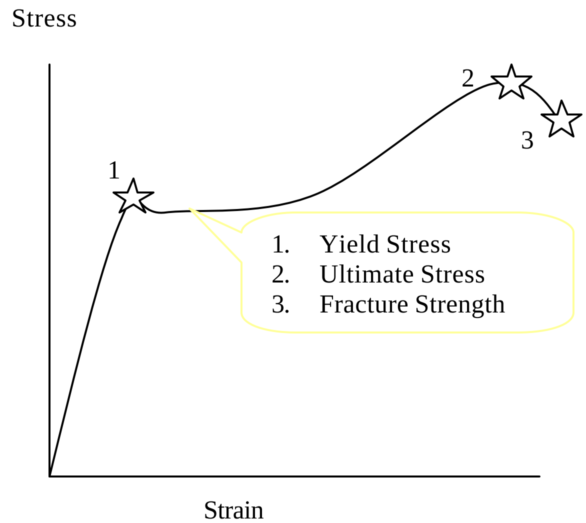
Properties of Engineering Materials

In typical fasteners, say bolts, the torsional load is negligible and the transverse deformation is minimal.

- Yield Strength, Strength,
- Ultimate Strength, and
- Fracture Strength.

All refer to the stress values at the conditions applicable.

Yield Stress (σ_y) is generally used for design purposes.



$E = \sigma/\varepsilon$ in the elastic range
i.e. $\sigma_y > \sigma$

(Axial load only)

σ_y, σ_u, etc. will vary for the same material depending on the hardness. But E, G and μ do not.

$$G = E/2(1 + \mu)$$

Also since $\sigma = P/A$, $\varepsilon = \Delta L/L$

$$P/A = E\ \Delta L/L$$

Or $\boxed{P = (AE/L)\ \Delta L}$

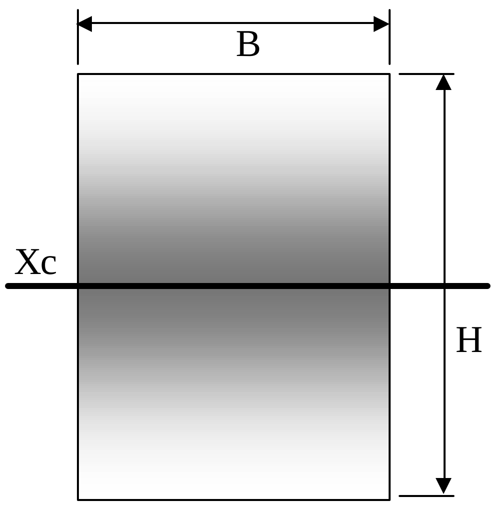
This equation is similar to $F = K x$ for spring

Where

$F = P$	Force applied
$X = \Delta L$	Elongation under axial load
$K = (AE/L)$	Stiffness of the body

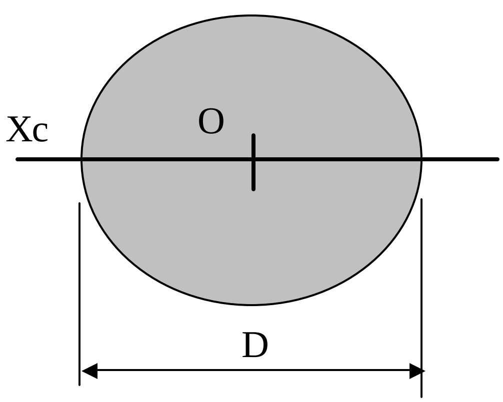
Properties of Plane Areas (Common cross-sectional areas)

B = Width of rectangular section,
H = Height of the section
Xc = Horizontal axis through the center
D = Diameter of the circular section
A = Area of section
Ixc = Area moment of inertia (about Xc axis)
Jo = Polar moment of inertia of circular section



$$A = B\,H$$

$$I_{xc} = \frac{B\,H^3}{12}$$



$$A = \frac{?\ D^2}{4}$$

$$I_{xc} = \frac{?\ D^4}{64}$$

$$J_o = \frac{?\ D^4}{32}$$

These formulas will be used to compute stresses and strengths.

Basic Relationships of Load, Stress, Strain, and Stiffness

Load & Stress

P = A s

Stress & Strain

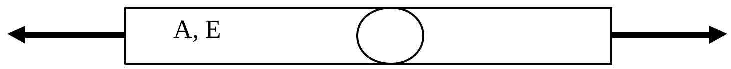
s = E e

Strain & Elongation

e = ?L/L

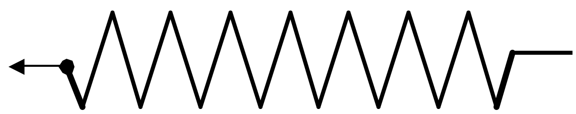
Stiffness & Load

K = P/?L



Stiffness K = (AE/L)

Stiffness of spring, K



Stiffness of a rod, K = P/ ?L = (A s)/ ?L = (A (E e))/ ?L = (AE)x (e /?L) = AE/L

Or K = AE/L

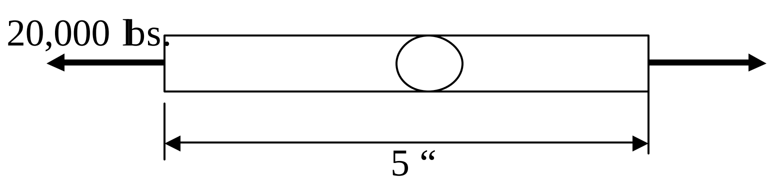
Example: Find elongation under 20,000 lb. tension

½" dia, 5" long rod. $E = 30 \times 10^6$ psi

$A = \pi/4 \times (1/2)^2 = \pi/16$

$K = \pi/16 \times 30 \times 10^6/5 = 1.18 \times 10^6$ lbs./in.

Elongation

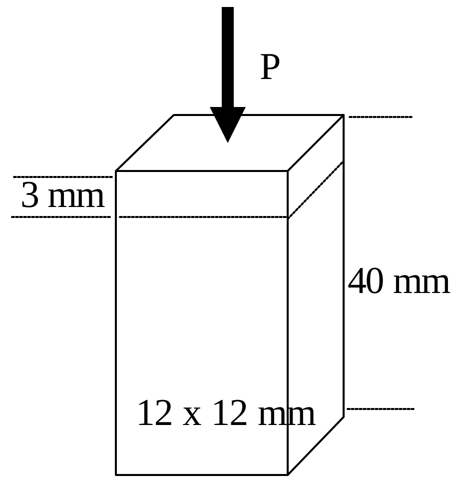


$F = K\Delta L$

Or $\Delta L = K/F = 1.18 \times 10^6/20{,}000 = .006$

Example:

Find force required to
Compress the column by
3 mm (E = 30 x 10^6 psi)



Ans.:

$E = 30 \times 10^6$ lbs./in^2

$= 30 \times 10^6 \times 4.45^N/(.0254)^2$

$= 206.9 \times 10^3$Mpa

[10^6pa = 1Mpa, 1 pa = 1N/m^2]

$K = \Delta E/L = 12 \times 12/40 \times 100 \times 206.9\ 10^3 \times 10^6$ N/m

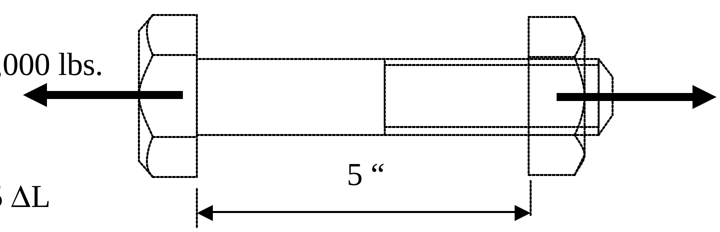
Or $K = 744.8 \times 10^3$ N/mm

But $P = (K)\ \Delta L = 744.x \times 10^3 \times 3 = 2.234 \times 10^6$N

Example:

How much would a ½" dia. Bolt extends under 45,000 tension. (E = 30 x 10^6 psi)

$P = (AE/L)\ \Delta L$



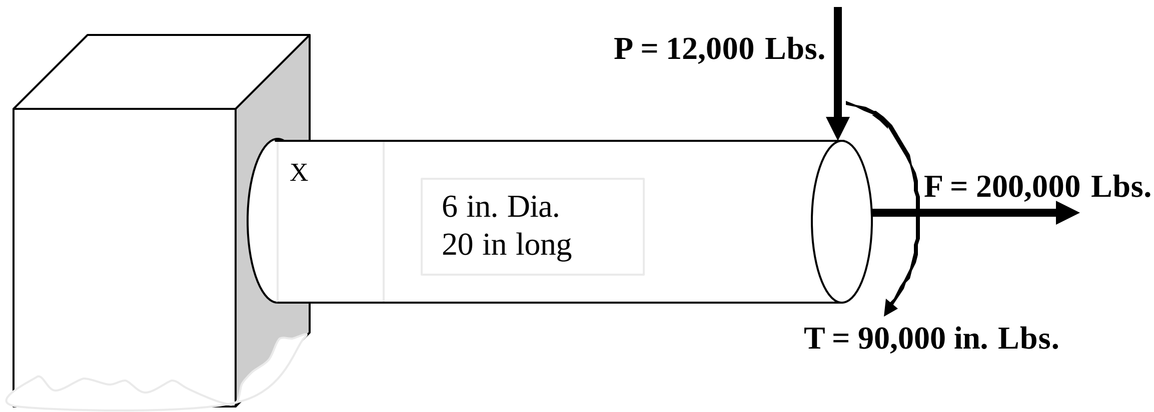
$45{,}000 = AE\ \Delta L/L$
$= \pi(.5)^2 \times 30 \times 10^6/4 \times 5\ \Delta L$

$45 \times 10^3 = 1{,}177.5 \times 10^3\ \Delta L$

Or $\Delta L = 45/1{,}177.5 = .038$ in. Ans.: .038 in.

What are Elongation, Extension, Compression, Tension, Load, and Strength?

Example: Axial, Bending, and Shear stresses in rod of circular section



Axial stress

$$S_a = F / A = 200,000 / (3.14159x36/4) = 7,073 \text{ lbs/sq.in. (uniform)}$$

Bending stress

$$S_b = M (D/2)/I = (20x12,000)x3 / (3.14159x36x36/64) = 11,317 \text{ lbs/sq.in.}$$

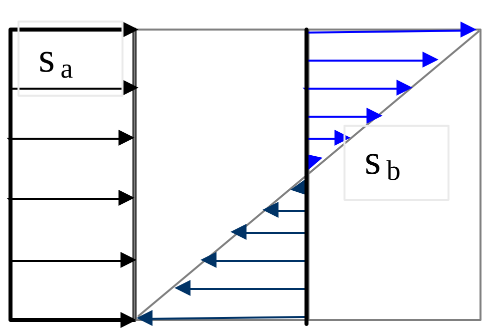
Torsional (Shear) stress

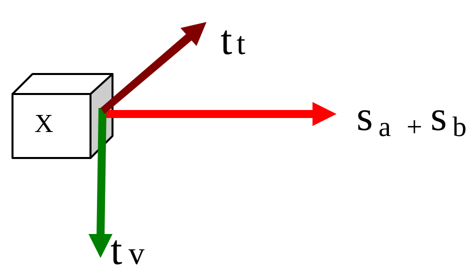
$$t_t = T (D/2) / J = (90,000 \text{ x } 3)/ (3.14159x36x36/32) = 2,122 \text{ lbs/sq.in.}$$

Vertical shear stress

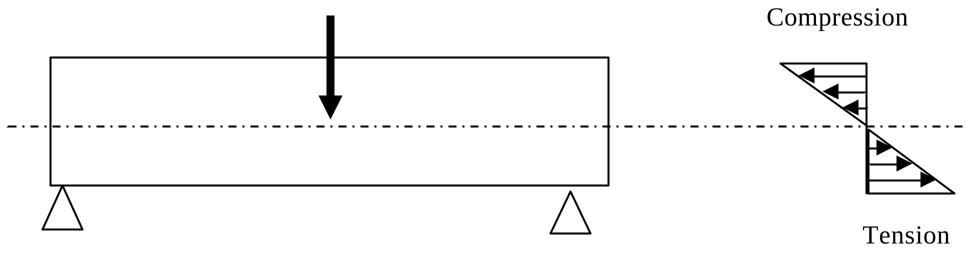
$$t_v = P /A = 12,000/ (3.14159x36x36/4) = 428 \text{ lbs/sq.in (uniform)}$$

Stresses at the upper fiber of the shaft are as shown below.





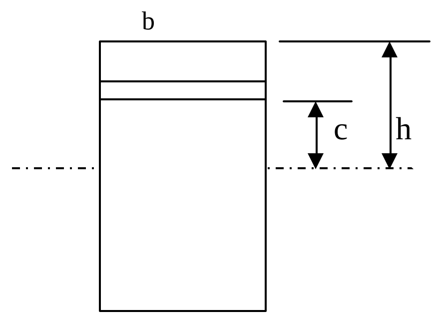
Stress Due to Bending Load



$I_x = bh^3/12$

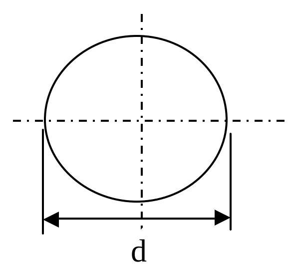
Stress at a fiber C in away from
the center of the area.

$\sigma = MC/I$, M = bending movement



(Area Moment of Inertia)

$I_x = \pi d^4 / 64$



Maximum moment at base

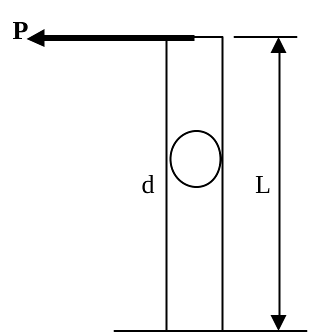
= P.L. lb.in.

Max. Stress

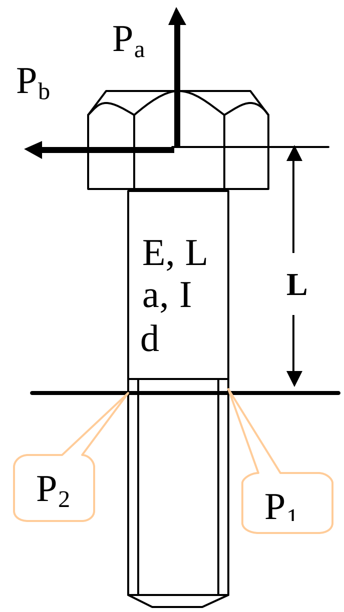
σ_{stress} = (PL)/I (d/2)

= $[PL/(\pi d^4)](d/2) \times 64$

$\sigma_{max} = 32PL/(\pi d^3)$



Combined Axial & Bending Loads



Often bolts are subjected to combined loading.

P_a = axial tension

P_b = bending load

Maximum tensile stress will occur at point P_1

σ_{max} =(P_a/a) + (P_bL / I) (d/2)

Bolt fails when the stress due to combined/single loading exceeds the ultimate strength of the bolt material.

For design purposes, steel bolts are designed to withstand 25,000 psi. Special and hardened bolts carry much higher stress levels.

Example:
A 5/8"- 18 bolt UNF threads is subjected to axial load P_a = 2,500 lbs and bending load at 3.5" unsupported bolt length. If the yield strength of the fastener material 35,000 psi, find the safe bending load, P_b, that can be safely applied to the bolt as shown.

Solution:

σ_{max} =(P_a/a) + (P_bL / I) x (d/2)

a = 0.785 x $(5/8)^2$ = 0.307
I = 3.14 x $(5/8)^4$ /64 = 0.00749

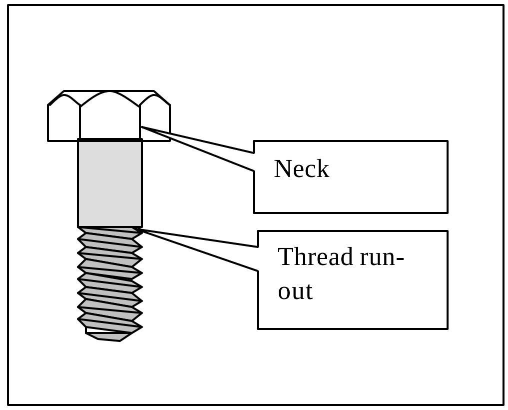
35,000 = 2,500/0.307 + (P_b x 3.5/ 0.00749) x (5/16)

or 35,000 = 8,143 + 146 P_b

or P_b = (3,500 – 8,143)/146 = 184 lbs.

Other Modes of Bolt Failures

Tensile Failure

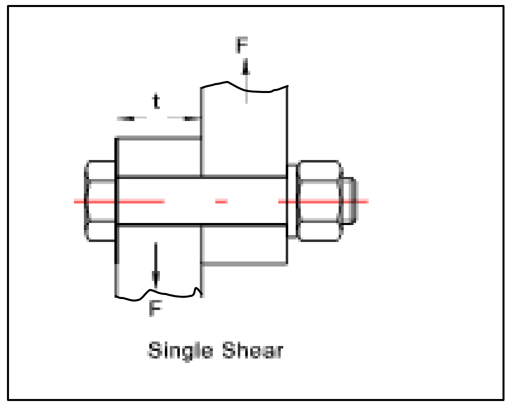


Maximum Strength (Load capacity)

$$= S_u \times A_S$$

A_S = Thread stress area under tensile load.

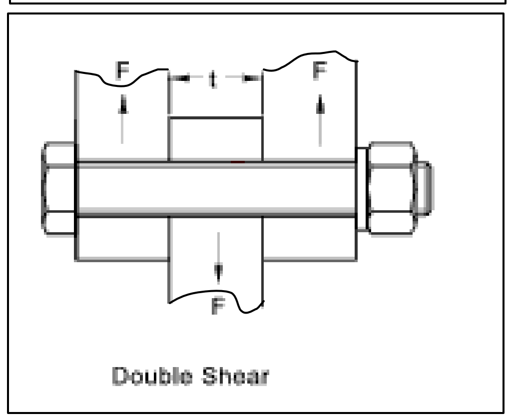
Single Shear Failure



Maximum Strength

$$= t_{max} \times (0.785 \times D^2)$$

Double Shear Failure

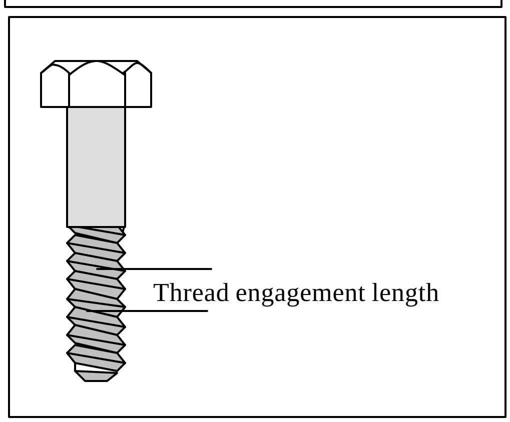


Maximum Strength

$$= t_{max} \times 2x\ (0.785 \times D^2)$$

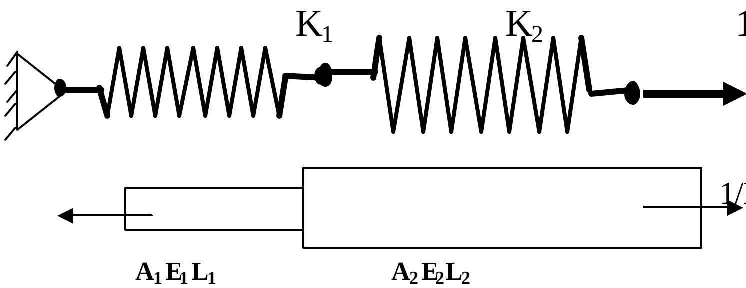
(Adjust if threaded area is involved)

Thread Stripping



Equivalent Joint Stiffness

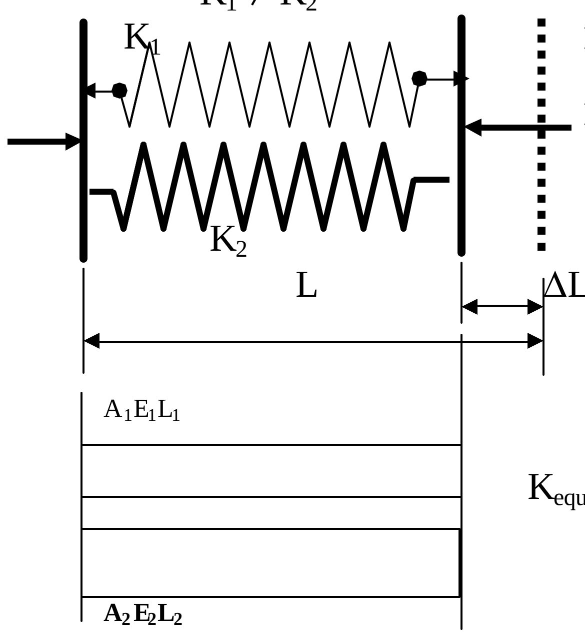
Stiffnesses in Series



$1/K_{equiv} = 1/K_1 + 1/K_2$

$1/K_{equiv} = 1/(A_1E_1/L_1) + (A_2E_2/L_2)$

Stiffnesses in Parallel



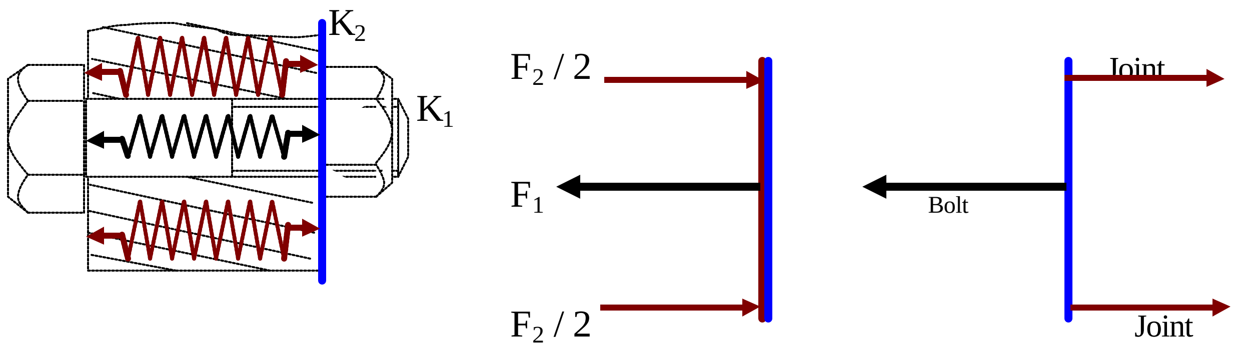
Equivalent Stiffness

$K_{equiv} = K_1 + K_2$

$K_{equiv} = (A_1E_1/L_1) + (A_2E_2/L_2)$

Characteristics of springs in parallel or series configurations:

When forces are equal in two connected springs, they are in SERIES.
When displacements in both springs are equal, they are in PARALLEL.

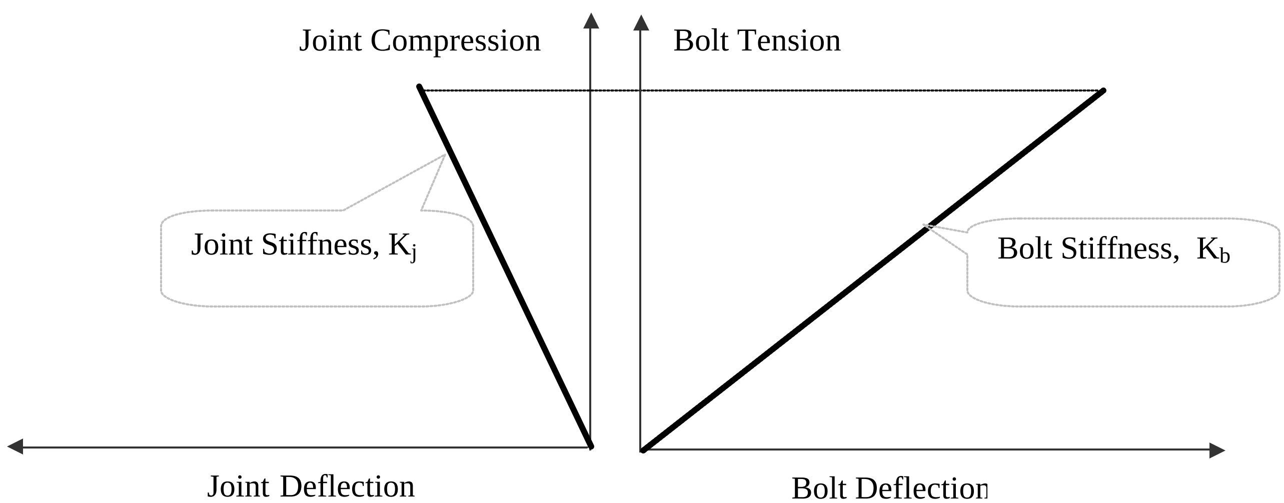


A preloaded bolt will apply the equal and opposite force to the joint. Under normal circumstance, bolt will be in tension and joint in compression. Treated as stiff springs exerting the forces, these two springs are considered in series.

How does bolt work?

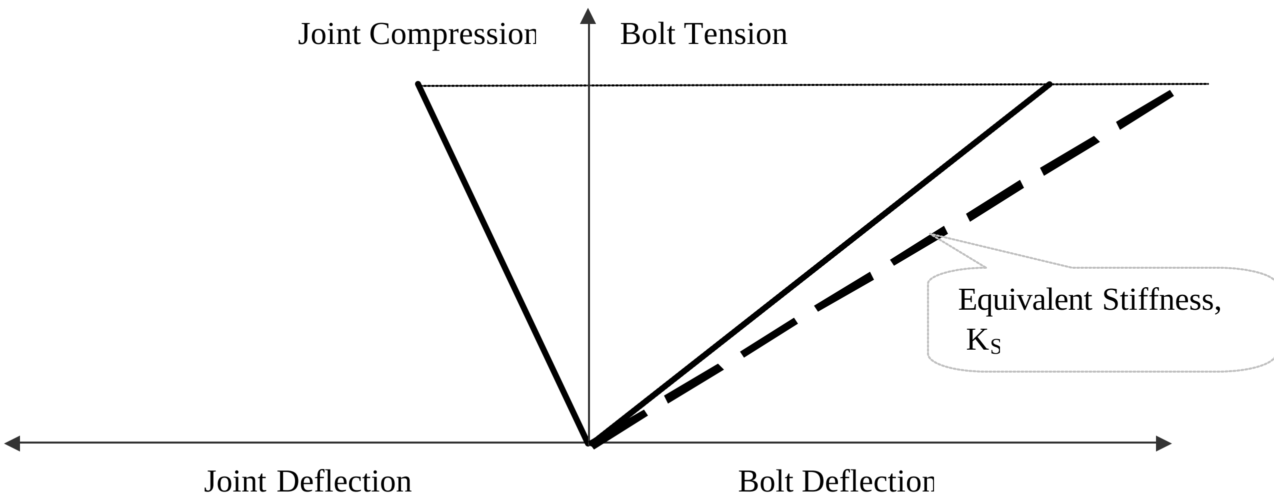
Clamping force developed by tightening the bolt. Each turn of the bolt stretches it by the thread pitch distance. The stretch produces tension or clamping force in the bolt.

How does a joint behave?



Joint stiffness is usually much greater than the bolt stiffness. The joint with the bolt tightened, behave with an equivalent stiffness of the joint system, K_S, which would always be less than either of the individual stiffnesses.

$$\boxed{1 / K_S \quad = \quad 1 / K_j \quad + \quad 1 / K_b}$$



What should be the nature of the combined stiffness relative to the same for bolt & joint?

Equivalent Bolt Stiffness and Total Bolt Stretch Formula

Generally joint is much stiffer than the bolt.

K_j = 10,000 lbs/ .001 in = 10×10^6 lbs/in

K_b = 10,000 lbs / .005 in = 2×10^6 lbs/in

$$1 / K_S = 1 / K_j + 1 / K_b$$

Or

$$1 / K_S = (1 / 10 + 1 / 2) \times 10^6 \text{ lbs/in}$$

$$K_S = (10 / 6) \times 10^6 \text{ lbs/in} = 1.67 \times 10^6 \text{ lbs/in}$$

Once the stiffness is known, it can be used to estimate the load for a certain deflection. Or, when the load is known, the deflection can be calculated.

$F = K \Delta L$ or $\Delta L = F/K$ or $K = F / \Delta L$

Observation: The equivalent stiffness of two springs is always lower than the smaller of the two components.

$1 / K_S \quad = \quad 1 / K_j \quad + \quad 1 / K_b$ or $\quad K_S = (K_j \times K_b)/ (K_j + K_b)$

Example Equivalent Stiffnesses:

K_1 = 2 lb/in, K_2 = 2 lb/in. $K_{Eqv.} = (K_1 \text{ x } K_2)/ (K_1 + K_2)$ = 4/4 = 1 lb/in.

K_1 = 4 lb/in, K_2 = 20 lb/in. $K_{Eqv.} = (K_1 \text{ x } K_2)/ (K_1 + K_2)$ = 80/24 = 3.33 lb/in.

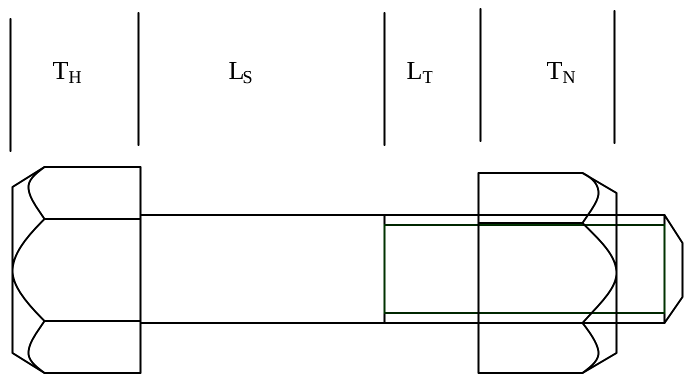
K_1 = 24 lb/in, K_2 = 300 lb/in. $K_{Eqv.} = (K_1 \text{ x } K_2)/ (K_1 + K_2)$ = 7200/324 = 22.2 lb/in.

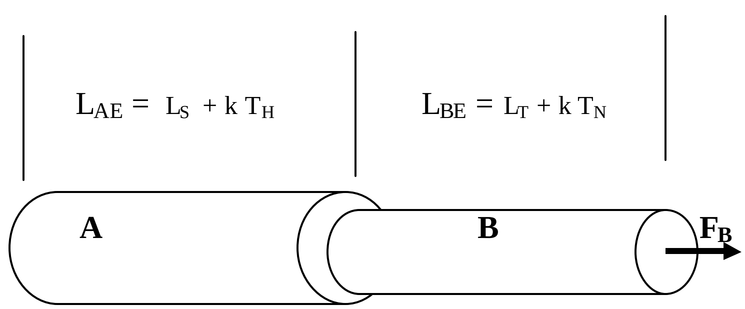
K_1 = 20 lb/in, K_2 = 1000 lb/in. $K_{Eqv.} = (K_1 \text{ x } K_2)/ (K_1 + K_2)$ = 20,000/1020 = 19.6 lb/in.

Notice that the equivalent stiffness range between 50 – 100% for the difference in values from 0 – very large.

BOLT GEOMETRY

Notations Used:
T_H = Length of bolt head
T_N = Length of nut
L_S = Length solid bolt head
L_T = Length threaded bolt head
k = Constant (0.5 for head, 0.6 for nut))
L_{AE} = Equivalent length of A
L_{AE} = Equivalent length of B
L_E = Equivalent bolt length
A_A = Stress area of A
A_B = Stress area of B
E = Modulus of elasticity
? L_T = Total length changes of all portion
K_A, K_B = Stiffnesses of portion A & B





$$\mathbf{L_E = L_{AE} + L_{BE}}$$

$$\mathbf{= L_S + k\,T_H + L_T + k\,T_N}$$

$$?\mathbf{L_T} = \mathbf{F_B \times (}1/K_A + 1/K_B) = \frac{\mathbf{F_B L_{EA}}}{\mathbf{EA_A}} + \frac{\mathbf{F_B L_{EB}}}{\mathbf{EB_A}}$$

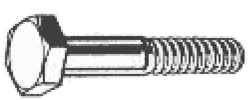
US Nut Size Table

Size	Diam.*		Height			
	Hex Nut	Machine Screw Nut	Hex Nut	Jam Nut	Nylock Nut	Machine Screw Nut
1/4	7/16	7/16	7/32	5/32	5/16	3/16
3/8	9/16	5/8	21/64	7/32	29/64	1/4
3/4	1-1/8	-	41/64	27/64	7/8	-
7/8	1-5/16	-	3/4	31/64	63/64	-
1	1-1/2	-	55/64	35/64	1-3/64	-

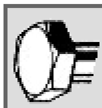
* This is the diameter across the flats. It is also the size of wrench to use.

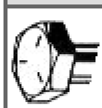
Height of hex nut and bolt head range between 60% - 90% of bolt diameter. For specific bolt, the exact dimension should be selected from table.

Bolt Materials and Grades


Source: http://www.boltdepot.com

A standard bolt has a hex head and a smooth shoulder area beyond the standard amount of threading. Shorter lengths are fully threaded.


Steel grade 2
Zinc plated low carbon steel. Best for general hardware use where high strength is not required.


Steel grade 5
Made from medium carbon steel, tempered and zinc plated. Best for automotive use and other areas where higher strength is desired.

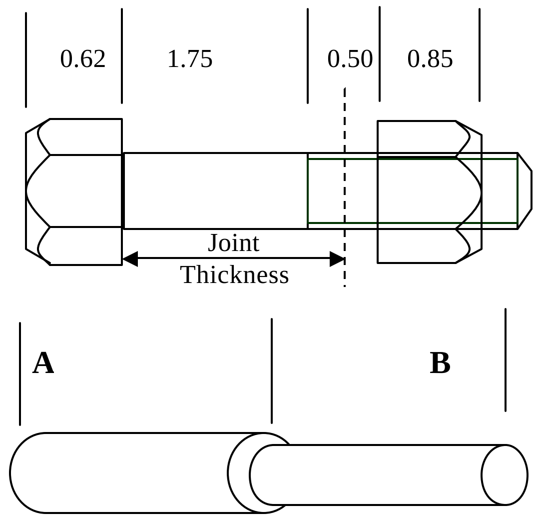

Steel grade 8
Made from medium carbon alloy steel, tempered and yellow zinc plated. Best suited for applications where high strength and hardness is required.

Stainless steel 18-8
Stainless steel 18-8 is an alloy of steel with high corrosion resistance. Stainless has become the material of choice for exterior and most marine applications.


Grade 5 chrome
A grade 5 fastener with a bright mirror-like finish providing sharp looks for a variety of applications.

Bolt grades and markings will be discussed later in this course.

Example:
Determine the equivalent length of a 1" dia. bolt with the dimension shown. Calculate equivalent lengths using 50% involvement of both bolt head and nut.



Equivalent length, part A
= 1.75 + 0.50 x 0.62 = 2.06 in.

Equivalent length, part B
= 0.50 + 0.50 x 0.85 = 0.925 in.

Alternate form (when joint thickness is known):

Equiv length of part B
= Joint Thickness - (Bolt Length – Threaded Length) + (T_N) x 0.6

Example:
For a bolt SAE J 05 and Nut SAE J 104 and ANSI B.1-1974 (thread), the dimensional specifications are as follows. Estimate the change in bolt length when subjected to maximum design load.

Nominal Bolt Diameter (D) = 3/8 in.(0.375 in.)
Height of nut ($\mathbf{T_N}$) = 0.3285 in.
Height of head ($\mathbf{T_H}$) = 0.2354 in.
Thread length ($\mathbf{L_T}$) = 1.00 in.
Tensile stress area of threads ($\mathbf{A_B}$) = 0.0775 in^2.

Bolt Length = 1.5 in.
Joint Thickness = 1.0
E = 30×10^6 psi, Yield Stress = 130 ksi

Solution

Portion A (Figure above)

A_A = ? $D^2/4$ = ? x 0.375^2 = 0.1104 in^2

L_{AE} = (1.5 – 1.0) + (T_H)/2 = 0.6173

$1/K_A$ = $(L_{AE})/(E \times A_A)$
= 0.6173 / (30 x 10^6 x 0.1104)

= 0.1864x 10^{-6}

or **K_A = 5.365 x 10^6 in/lb**

Portion B

Area (A_B) = 0.0775 in^2.

L_{BE} = Joint Thickness
- (Bolt Length – Threaded Length)
+ (T_N) x 0.6
= 1.0 - (1.5 – 1.00) + 0.3285 x 0.6
= 0.697

$1/K_B$ = $(L_{BE})/(E \times A_B)$
= 0.697 / (30 x 10^6 x 0.0775)

= 0.2998x 10^{-6}

K_B = 3.336 x 10^6 in/lb

Equivalent Stiffness of the bolt **(K_E)** can be calculated using:

$1/K_E = 1/K_A + 1/K_B$

$K_E = (K_A \times K_B)/(K_A + K_B)$

= (5.365 x 3.336)/(5.365 + 3.336) x 10^6

= **2.056 x 10^6 lbs/in.**

Assuming 60% of yield stress, 130 ksi, and bolt is subjected to stress of 78,000 psi.

F_B = 78,000 x 0.0775 = 6,045 lb

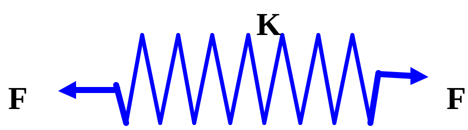
Work done to stretch the bolt
= (1/2) **F_B x ?L_T**

Potential energy stored
= (1/2) x **F_B x ?L_T**

Bolt length change (stretch, ?L_T) can be calculated as:

?L_T = F_B x ($1/K_A + 1/K_B$) = 6,045 / (2.056 x 10^{-6}) = **0.00294 in.**

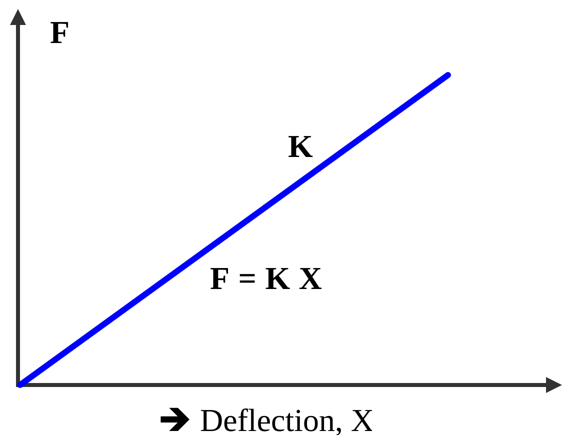
Bolt and Joint behavior in Absence of External Load

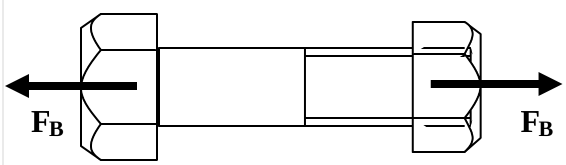


Coil Spring

During elastic range, spring stiffness will be

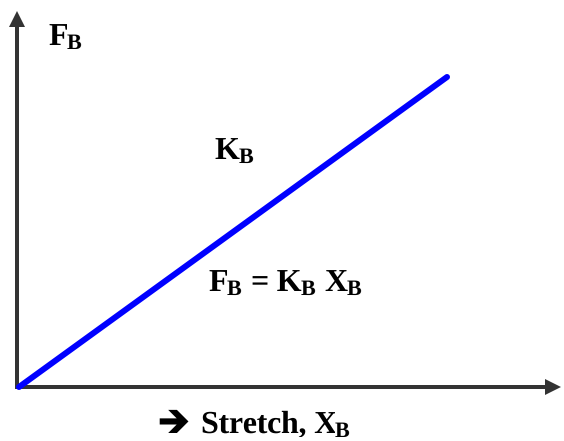
$$K = F/X \quad \text{(constant)}$$

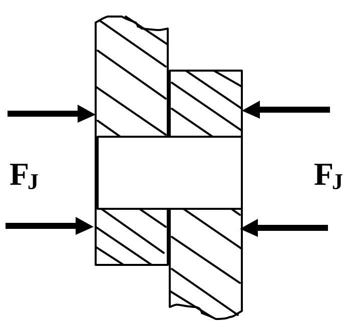




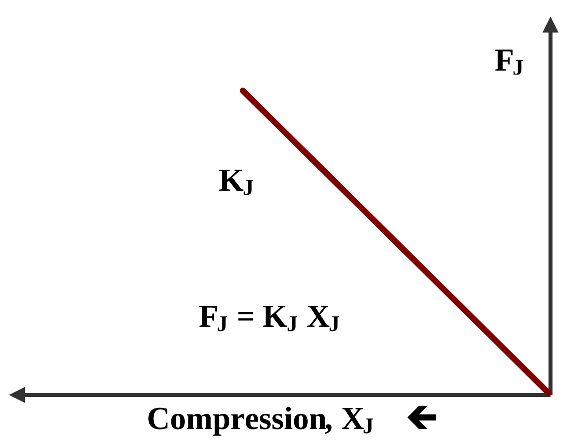
In elastic range of loading, both bolt and joint members behave like a spring.

K_B ➔ K_{eqv} (for more correct evaluation)



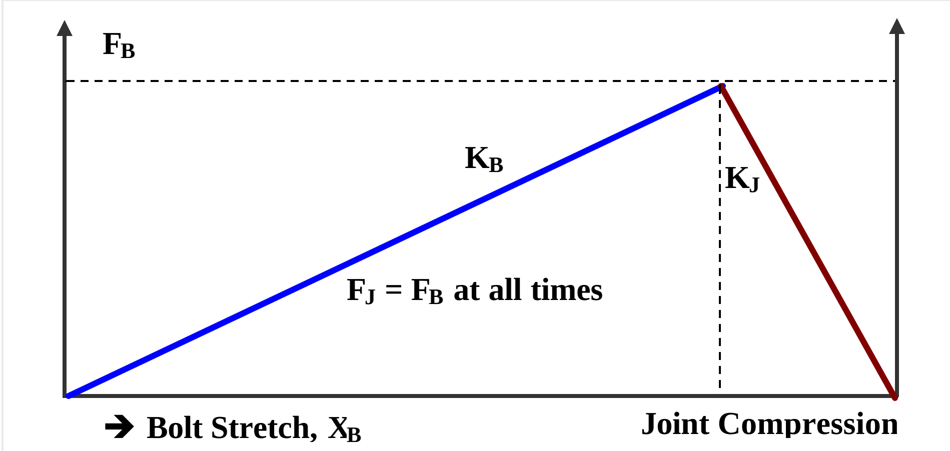


Joint is subjected to equal and opposite to that sustained by the bolt. In absence of any external load, $F_J = F_B$.



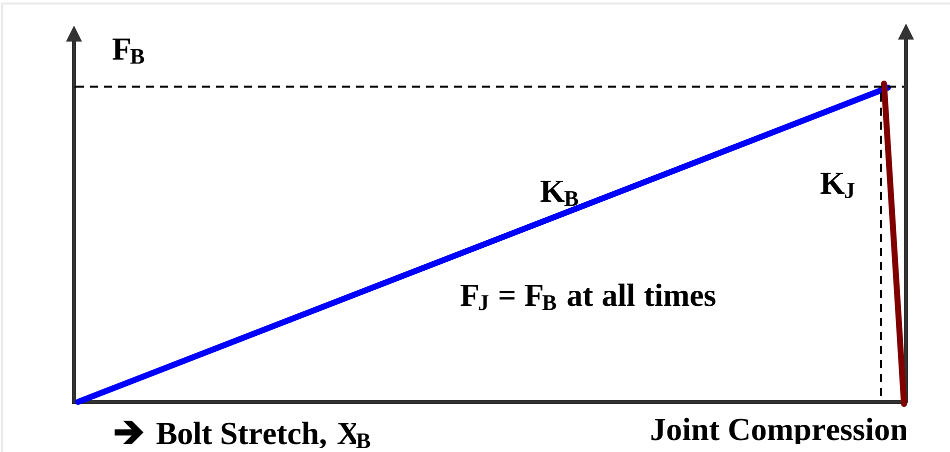
Because the load in bolt and joint is always equal, the force-displacement plots for both can be shown using the same horizontal axes.

Deflection of joint and bolt depends on their stiffnesses.



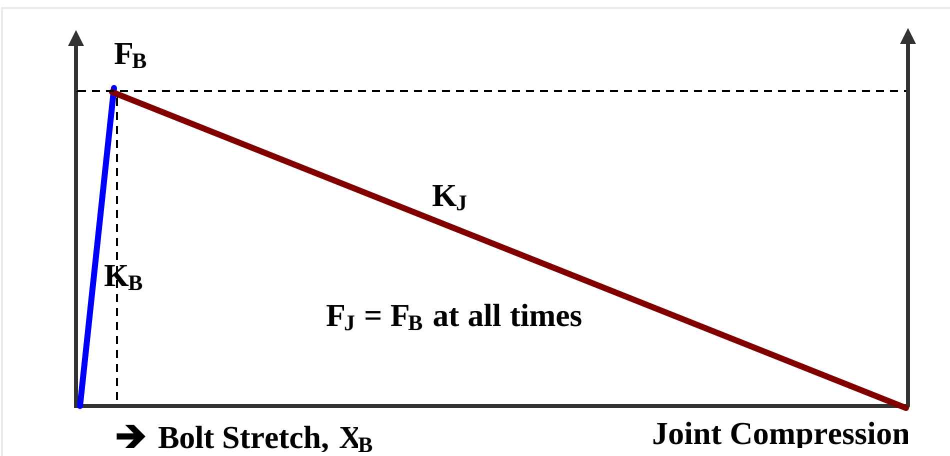
Very stiff joint and softer bolt

Joint stiffness>>
Bolt stiffness

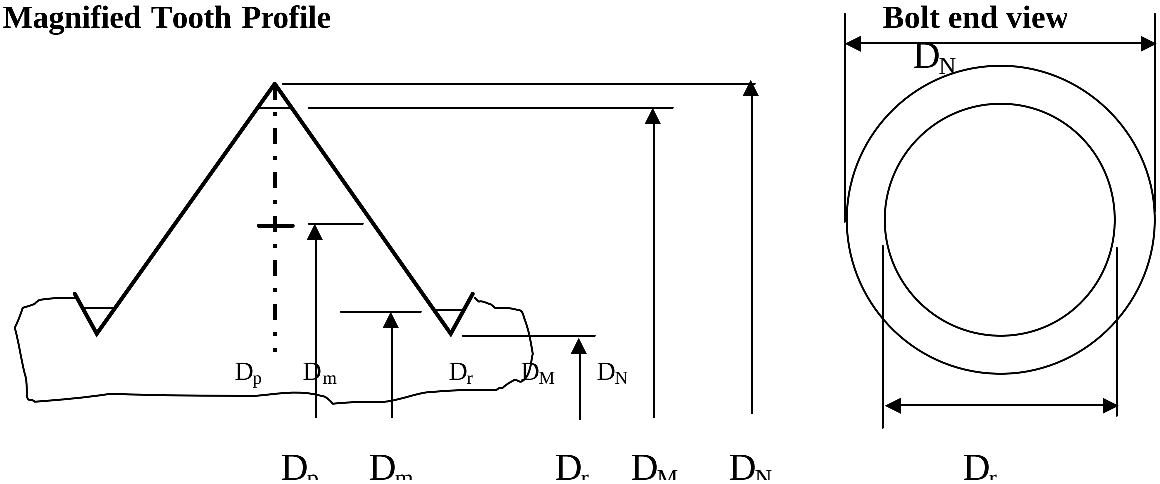


Very stiff bolt and very soft joint members.

Bolt stiffness>>
Joint stiffness



Thread Geometry



Specific d
(British A

D_N = Nominal diameter

D_M = Major diameter

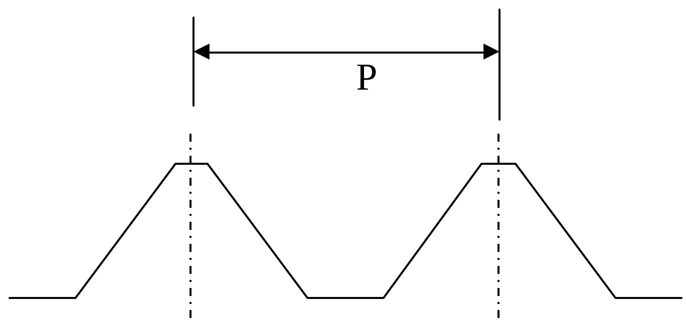
D_p = Pitch diameter

D_m = Minor diameter

D_r = Root diameter

p = pitch, distance between
two threads

1/p = number of threads per inch



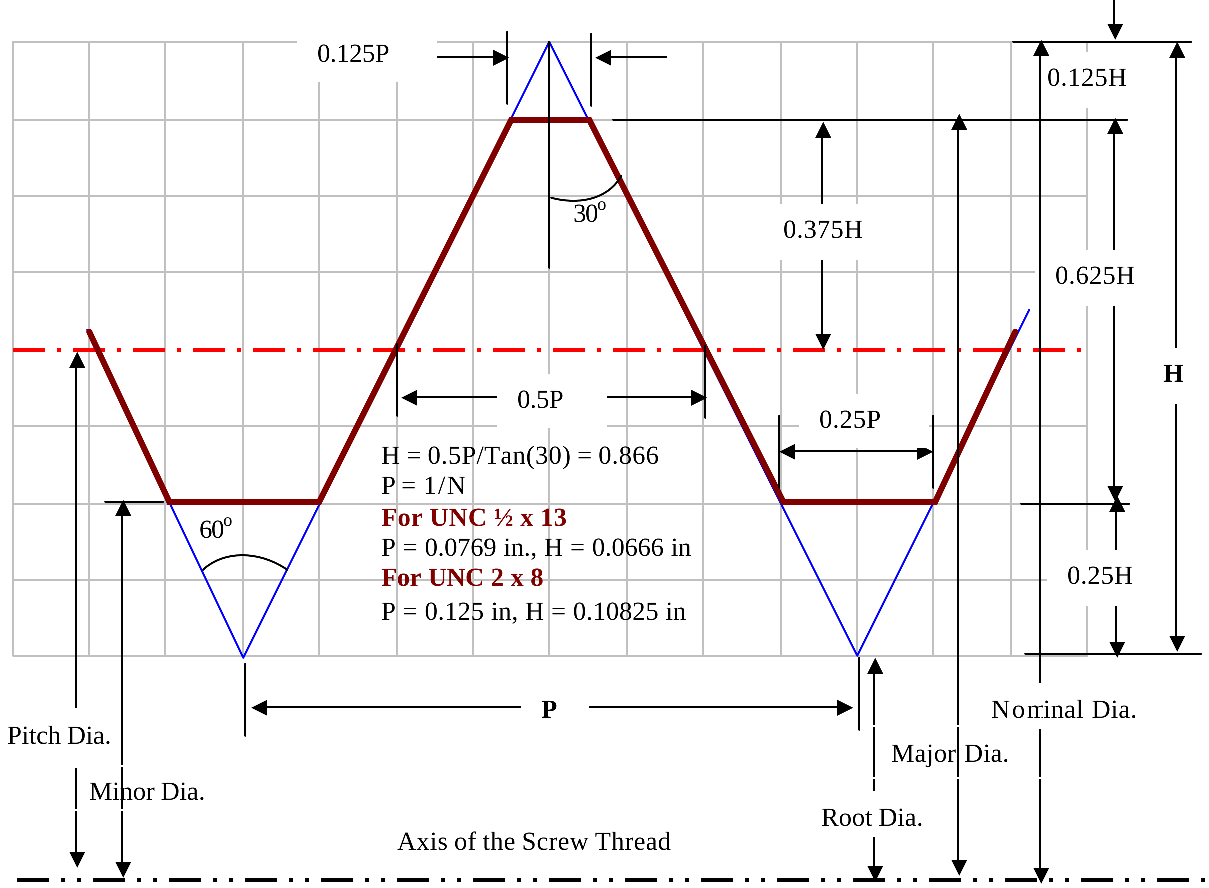
Threads

Standards for Thread Forms

ASME B1.1 & Federal Standard FED-STD-H 28/2B is for inch series thread forms. It describes basic Unified Thread Form, identified by code letters UN/UNR.

ANSI/ASME B1.18M-1982 is the standard for metric threads (shown below).

Basic profiles of UN, UNR, and Metric Thread Forms



For an UNC 2 x 8 fastener

Nominal Diameter (D_N) = 2.00 in, Pitch = 0.125 in., and H = 0.866P = 0.866/N = 0.108 in.

Major Diameter
$D_M = (D_N) - 2 \times 0.125 \times (0.866/N) = D_N - 0.2165/N$
Pitch Diameter
$D_P = (D_N) - 2 \times 0.5 \times (0.866/N) = D_N - 0.866/N$
Minor Diameter
$D_m = (D_N) - 2 \times 0.75 \times (0.866/N) = D_N - 01.30/N$
Root Diameter
$D_R = (D_N) - 2 \times (0.866/N) = D_N - 1.732/N$

Stress Area, $A_S = 0.785\ (\text{Diameter})^2$

Based on Pitch Dia., $A_S = 0.785\ (D_N - 0.866/N)^2$
Based on Minor Dia., $A_S = 0.785\ (D_N - 1.30/N)^2$

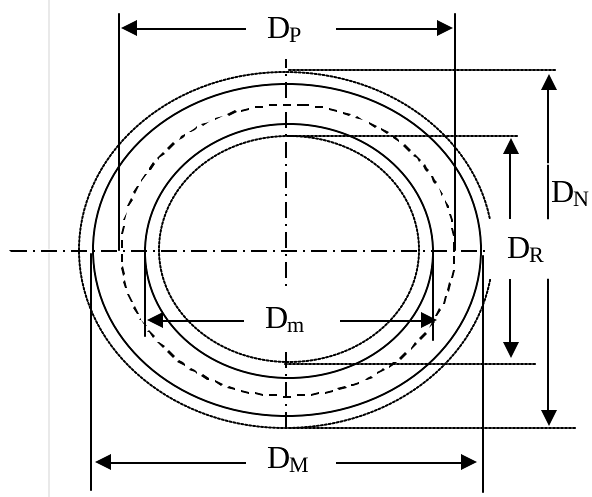


Fig.: Sectional view of a bolt showing all five diameters as defined above.

Stress Area for Threaded Bolt (Ref.2 Page 735)

Experiments have demonstrated that a general purpose bolt will break in tensile stress through an equivalent solid shaft of diameter between the pitch diameter and the minor diameter. The cross-sectional area of this solid shaft is called the stress area, A_S.

Bolt strength = (Ultimate tensile stress) x A_S
For UN or UNR inch threads, the standard (ASME B1.1) value of stress area is given below (N = Number of threads per inch, P = 1/N, D_N = Nominal diameter)

$$A_S = 0.785\,(D_N - 0.9743/N)^2 \qquad \ldots.. \qquad \text{(Use for inch threads)}$$

For M-form (same as UNR, 60^o included angle) metric threads, the standard (ASME B1.13M) value of stress area is

$$A_S = 0.785\,(D_N - 0.9382P)^2 \qquad \ldots.. \qquad \text{(Use for metric threads)}$$

(Stress area differs for other thread profiles and are established similarly with experiments)

US Threads Per Inch Table

Bolt Size	Threads Per Inch (TPI)	
	Coarse Thread UNC	Fine Thread UNF
#2	56	-
#10	24	32
#12	24	-
1/4	20	28
1/2	13	20
3/4	10	16
1	8	14
1-1/4	7	12
1-1/2	6	12

Shear Failure of Threads

Shear strength, $F_{Sh} = t_U \times A_{Sh}$

Where

t_U = Ultimate shera stress (taken as 50% of ultimate tensile stress, s_U)
A_{Sh} = Surface area through which shear occurs (tubular in shape for bolt/nut)

Depending on the relative strength of bolt and nut, the tread failure will occur either in nut or bolt threads, or in both simultaneously. The shear stress area of failure is different for each of the failure types.

(1) Failure when nut and bolt are of equal strength

Failure occurs simultaneously in both parts at the pitch diameter.

A_S = Stress area for tensile failure (defined earlier)
L_E = Effective length of shear area. This is the length of the treaded are required to develop full strength.

Full strength of bolt = $A_S \times s_U$

This must equal the resistance from shear area of length L_E.

At pitch diameter, perimeter = $p\ D_P$

Area of shear, $A_{Sh} = p\ D_P\ (1/2 \times L_E)$

Note the shear width of thread at pitch diameter is half the pitch distance.

Thus, for equal shear and tensile strength,

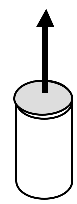
$$A_{Sh} \times t_U = A_S \times s_U$$

Or

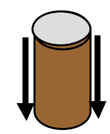
$$p\ D_P\ (1/2 \times L_E) \times (1/2 \times s_U) = A_S \times s_U$$

or

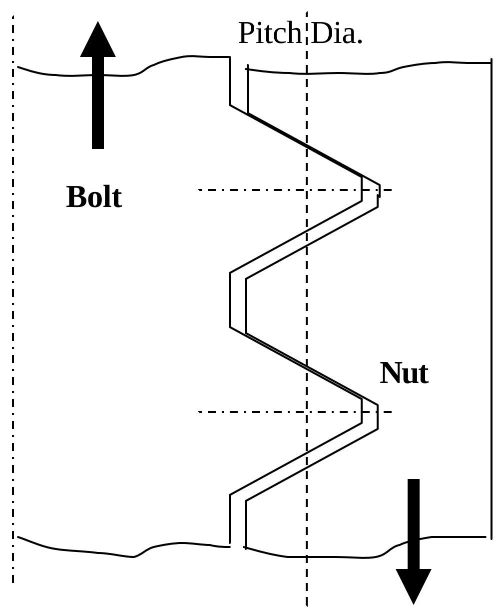
$$L_E = 4A_S/(p\ D_P) \quad (L_E = D_N \text{ commonly used})$$



Tensile Load and Area



Shear Resistance and Area



(2) Failure when nut thread is stronger than bolt

Failure occurs near the root of the bolt threads.

Exact value of the shear area are calculated by complicated formulas as per FED-STD-H28/2B standard (Ref. 2, page 136)

Reduce the area above by 12% (multiply by 0.88) to estimate correct standard value.

$$A_{Sh} = \pi\ D_P\ (5/8) \times L_E \quad \text{(simple form)}$$

Example:

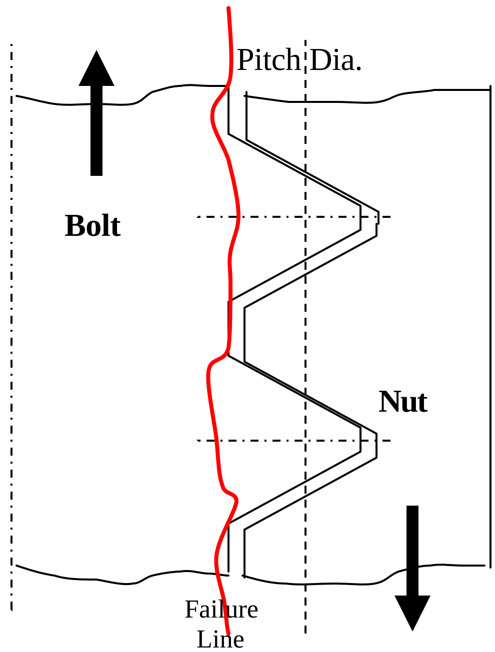
Compute A_{Sh} for a ¾-12 UN-2A thread. Assume the length of engagement (effective length) as one diameter (thickness of heavy nut).

From handbook table, pitch dia. = 0.6959

$$A_{Sh} = 3.14159 \times 0.6959 \times (5/8) \times (3/4)$$
$$= 1.025$$

With 12% correction,

$$A_{Sh} = 1.025 \times 0.88 = 0.902 \text{ in.}^2$$



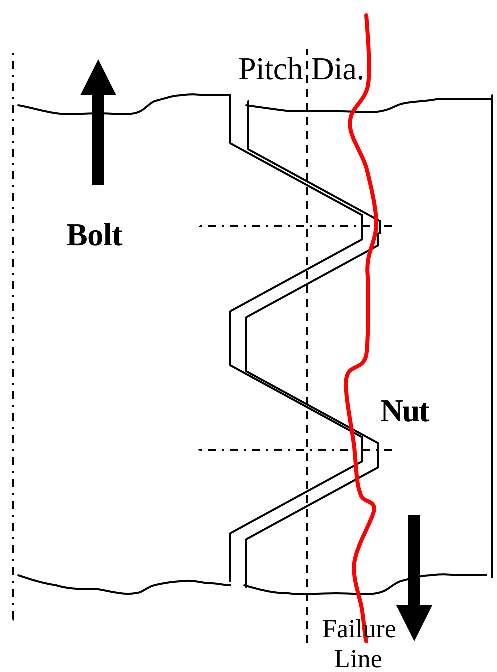
(2) Failure when nut thread is weaker than bolt

Failure occurs near the root of the bolt threads.

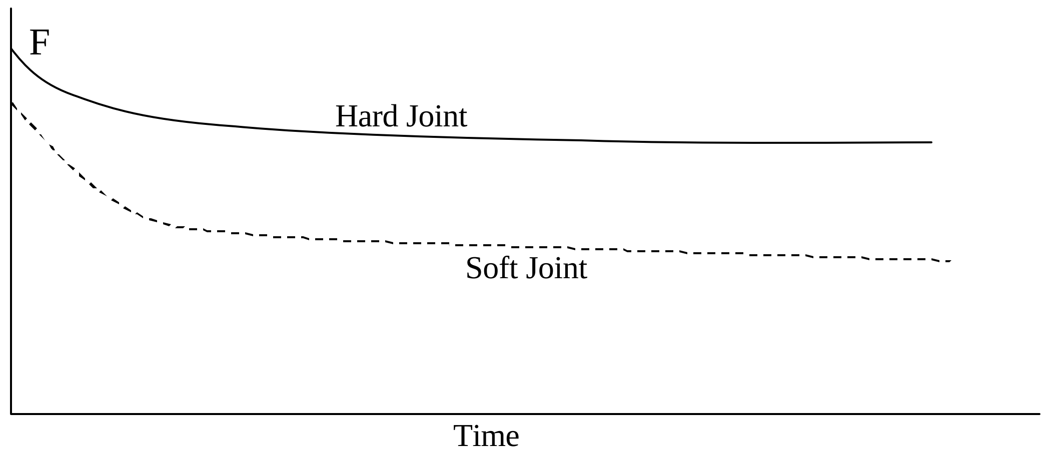
Exact value of the shear area is calculated by complicated formulas as per FED-STD-H28/2B standard (Ref. 2, page 137)

$$A_{Sh} = \pi\ D_P\ (3/4) \times L_E \quad \text{(simple form)}$$

Reduce the area above by 12% (multiply by 0.88) to estimate correct standard value.



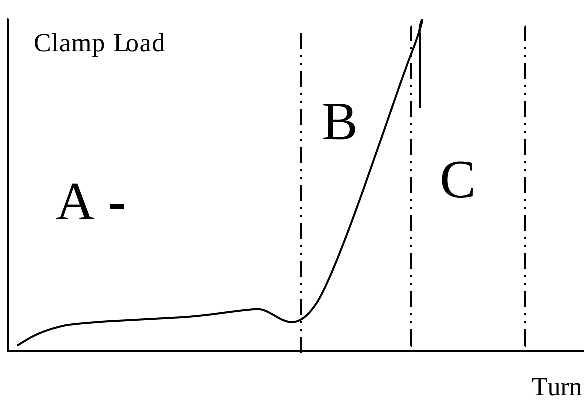
Preload Reduced by Relaxation



Effect of Joint Relaxation on Preload

Relaxation is a process in which the bolt tension (clamping force) is reduced as a result of (a) Vibration, (b) Gasket Creep, (c) Embedment, (d) Temperature expansion differentials, etc.

Embedment is localized plastic deformation caused by another part. It usually happens when a heavily loaded fastener allows one part to sink into another softer or more heavily loaded second part. Nuts or washers embed themselves in joint surfaces.

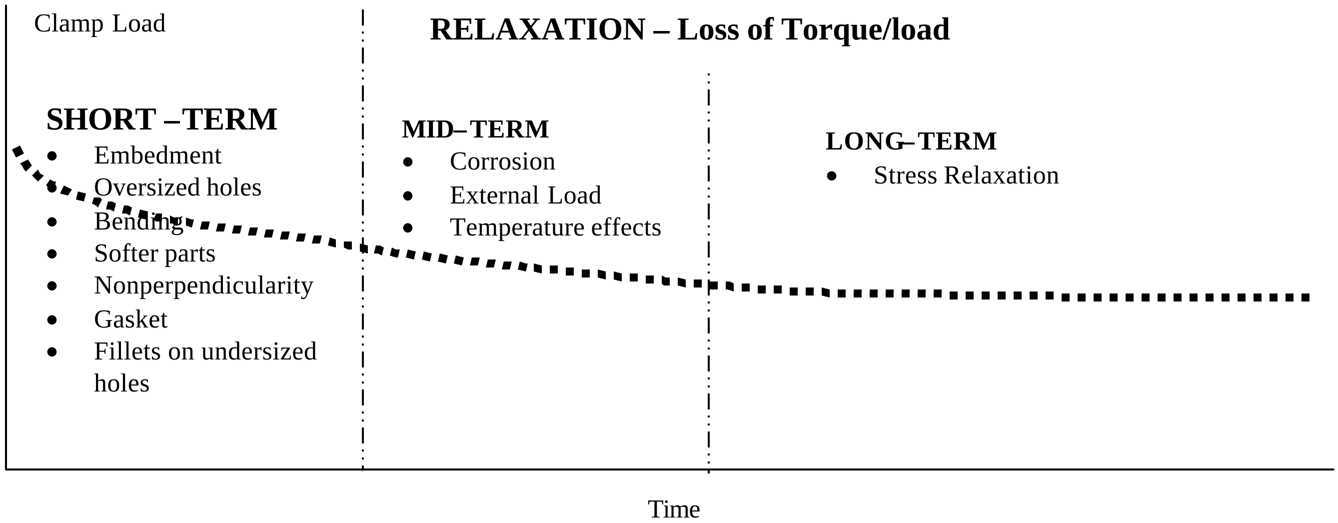


A – As bolt is turned, it compresses washers, gaskets, etc. and makes contact with joint

B – Assembly: Bolt stretces and compresses joint. Torque drops due to **relaxation.**

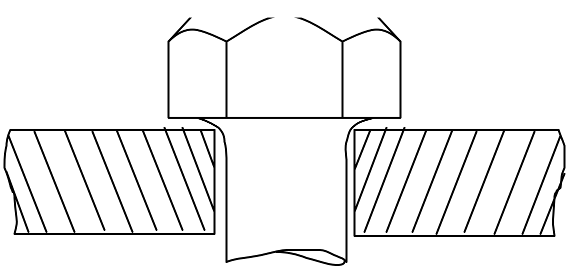
C- Service: External load is applied in service

Relaxation takes place throughout the entire life of joints. Depending on the time of occurrences, it is called short, mid, or log-term relaxation. The causes of relaxation are as shown below. (10 –20 % relaxation is common)

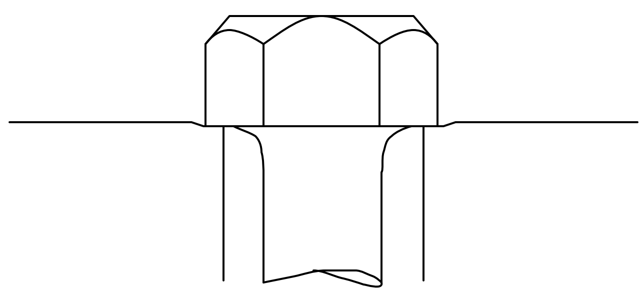


Causes of Relaxation

Oversized Fillet and Undersized Hole



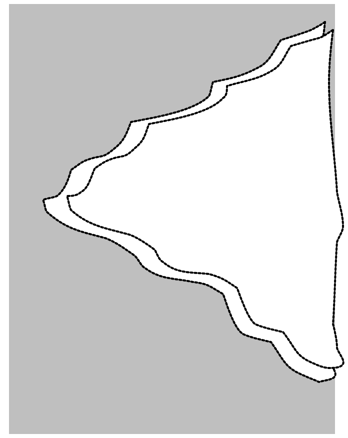
Oversized Hole, Excessive Brinnelling



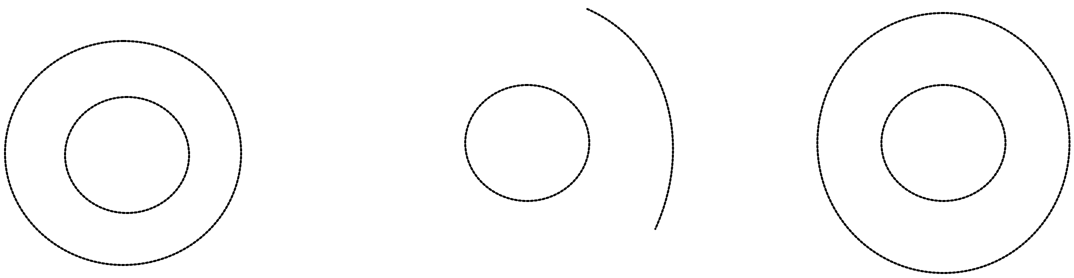
Thread geometry and contact

Engagement

Embedment



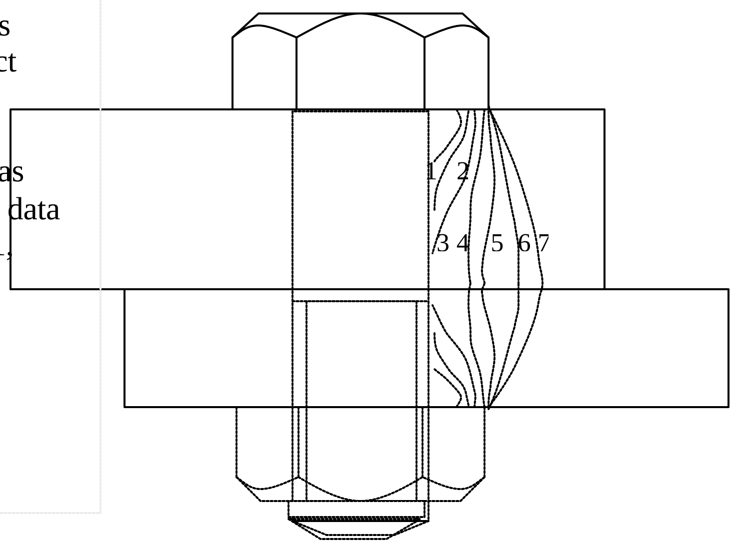
Witness Marks - Sometimes the nature of the witness marks reveals the causes for relaxation.



Stress Distribution in Joint Members
Studies conducted by Newport News Naval Shipyard demonstrated contact stress pattern in joint members.

Lines show uniform distribution areas under axial bolt load (100 kip, exact data of structure not yet published, Ref. 1, page 36).

1 – 35 psi, 2 -20 psi, 3 – 25 psi
4 – 20 psi, 5 -15 psi, 6-10psi
7 – 5 psi.



How to Minimize Relaxation

- Repeatedly tighten, loosen, and retighten fasteners to reduce embedment of the threads.

When nuts and bolts are formed they are generally rough. When they are put together and loosened, they become smooth as embedment and relaxation is reduced. Extra step to re-torque is helpful, but the cost must be justified.

- Assure thread engagement in excess of:
  - 1.25d for steel threads (d = nominal bolt diameter)
  - 2 d for Aluminum
  - 2.5 d for plastic

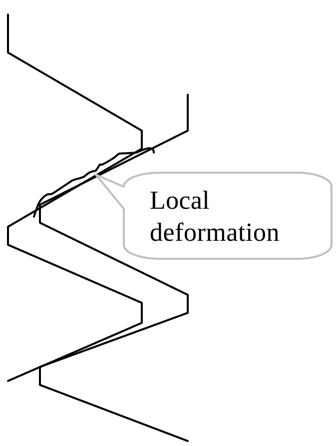
- Use longer bolts than short, stubby, or stiff ones.

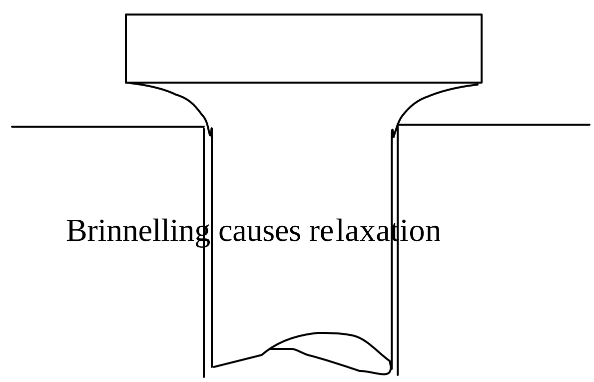
- Using automatic turners, tighten several fasteners at a time.

- Maintain better control fasteners to hole and thread to thread dimensions.

- Avoid bending by maintaining better perpendicularity of bolt axis with the joint surface.

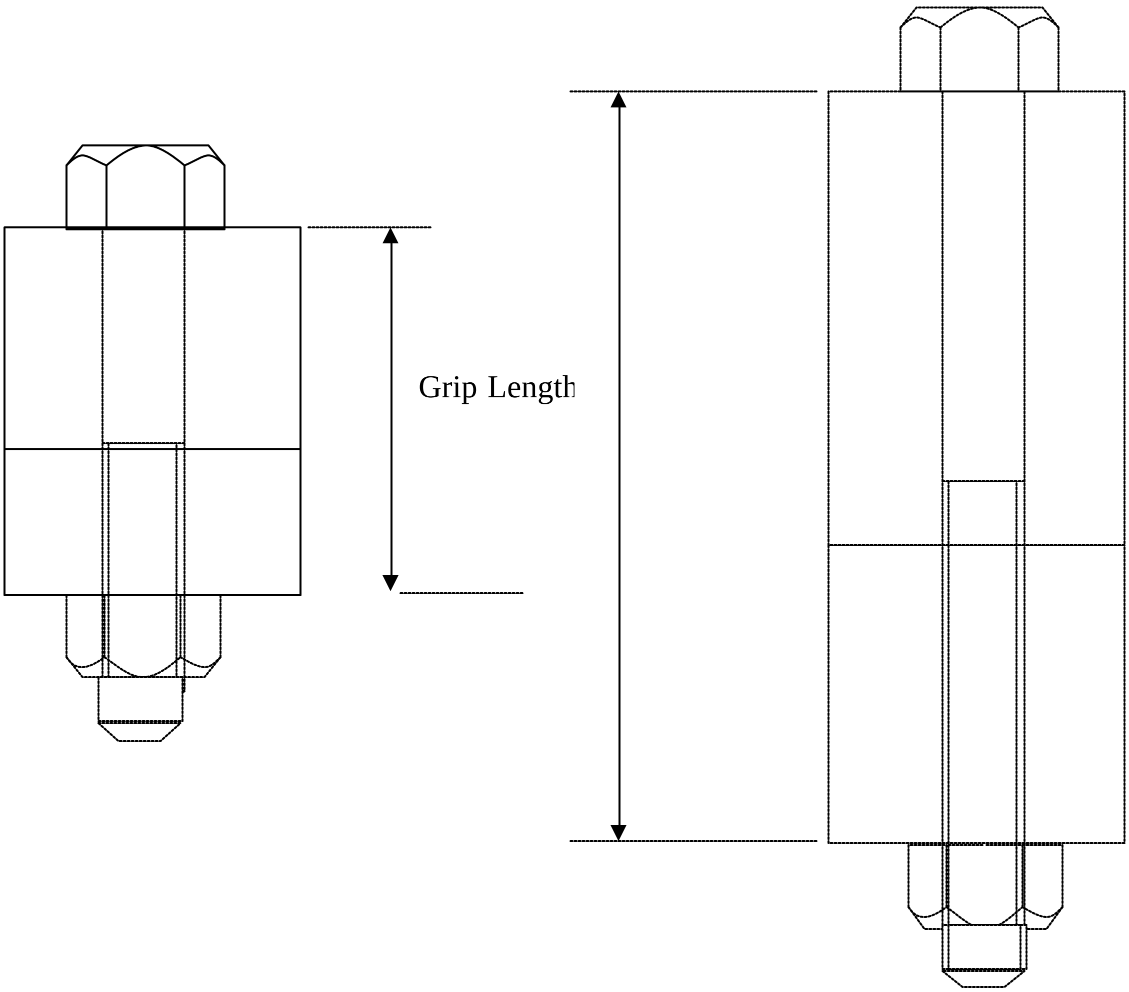
- Tighten bolts in several passes allowing time between passes for relaxation.





Bolt stretch and load

For a given diameter, shape, and load, a longer bolt will stretch more than a shorter one. This means that for a computed preload, the stretch will be a function of the Grip Length (usually thickness of the joint members). For a longer bolt, more stretch will be needed to generate the same load.



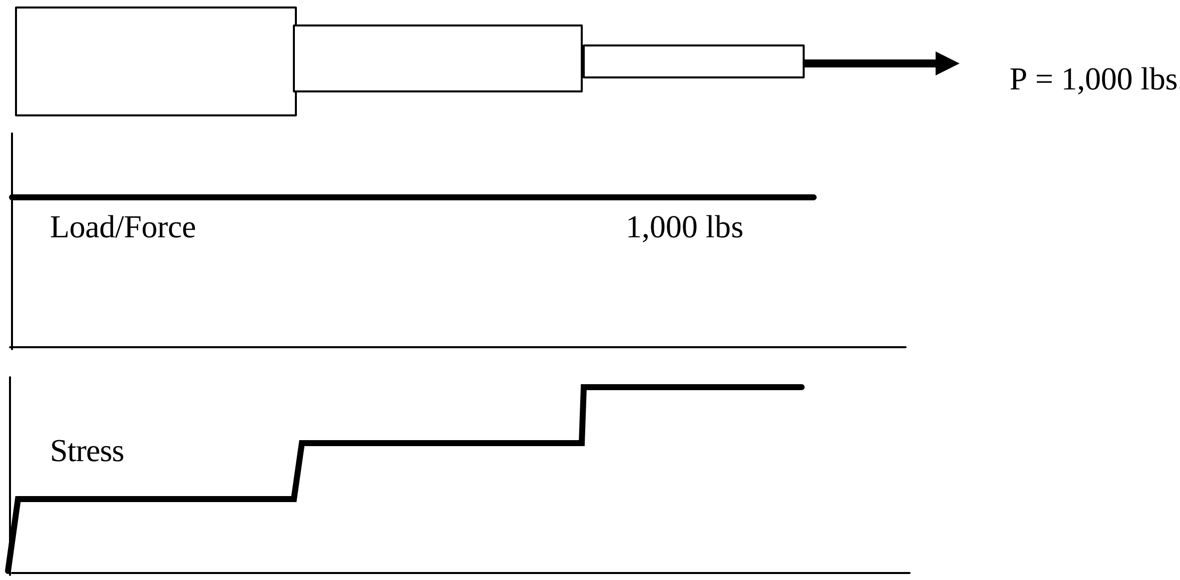
Stretch (Elongation)

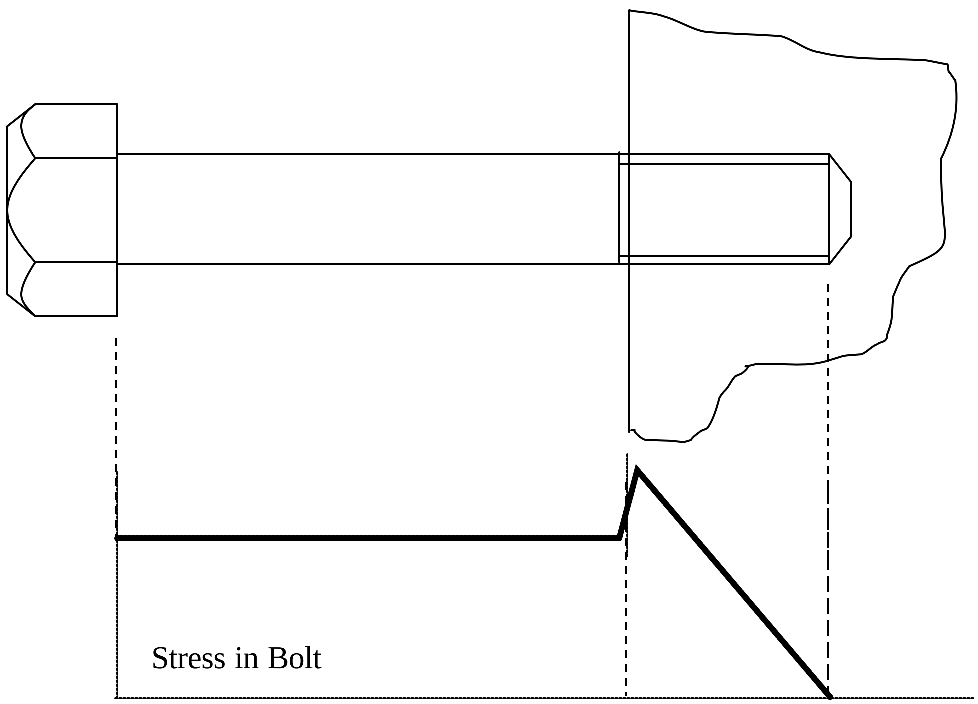
$\Delta L = F \; L/(AE)$ since $F/A = E \; \Delta L / L$

For a bolt twice as long, the stretch needed will be twice as much.

Longer bolts are advantageous in terms of loss of load due to relaxation.

Load and Stress Distributions
(Crossectional Effects)

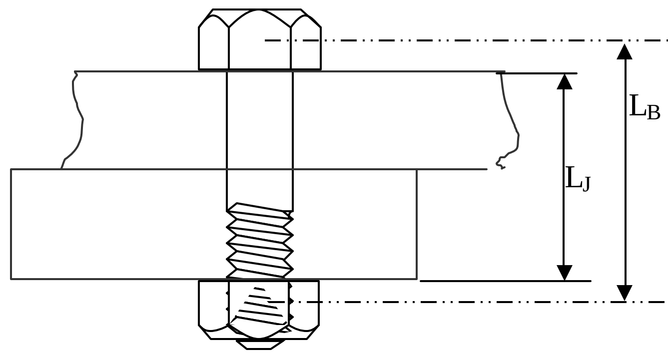




Thermal Effect on Bolt Tension

When there is change in temperature (say rise), both bolt and the joint members will change in length. If bolt and the joint members are of different materials, there will be a difference in the thermal expansion resulting in additional bolt load.

Bolt – Steel
Joint members – Aluminum



Coefficient of Thermal Expansion

$\alpha_J = 12.8 \times 10^{-6}$ for Aluminum joint
$\alpha_B = 6.5 \times 10^{-6}$ for Steel bolt

$\alpha = 8.4 \times 10^{-6}$ for Stainless Steel at room temperature

Expansion of Joint members $\Delta L_J = \alpha_J L_J T$

Expansion of Bolt $\Delta L_B = \alpha_B L_B T$

For a change of temperature T.

If $\Delta L_J > \Delta L_B$ then the force developed in the bolt due to temperature effect is

$$F_{adl.} = (\Delta L_J - \Delta L_B) A_B E_B / L_B$$

Example:
Steel Bolt and Aluminum Joint: T = 50 Degree F., L_B = 6 in. L_J = 5.5 in.
Bolt properties: A = 0.8 in^2, E = 30×10^6 psi.

$$\Delta L_B = 6.5 \times 10^{-6} \times 6 \times 50$$
$$= 1.95 \times 10^{-3}$$

$$\Delta L_J = 12.8 \times 10^{-6} \times 5.5 \times 50$$
$$= 3.52 \times 10^{-3}$$

$$F_{adl.} = (3.52 \times 10^{-3} - 1.95 \times 10^{-3}) \times 0.8 \times 30 \times 10^{6} / 6.0$$
$$= 6{,}280 \text{ lbs.}$$

What happens when preload is incorrect?

Fastener Failure – If preload is excessive faster may fail by body separation or thread stripping.

Joint Member Failure – Excessive preload may cause joint member to crush, warp, gall, or fracture.

Fatigue Failure of Bolt – Higher preload enhances the chance of bolt failure by fatigue under externally applied cyclic loading**.**

Joint Separation – Low preload may cause joint to separate and initiate leakage. (Example: fluid in a pipeline, combustion product in engine, etc.)

Weight and Cost – Insufficient preload forces a design to have a larger number of fasteners which increases cost and weight.

Loosening of nut due to Vibration – Inadequate preload causes loosening of nut.

Slippage of Joint members - Slip of joint member may cause misalignment of joint members.

FACTORS THAT AFFECT PRELOAD BY TIGHTENING THE BOLT

Tool Accuracy –

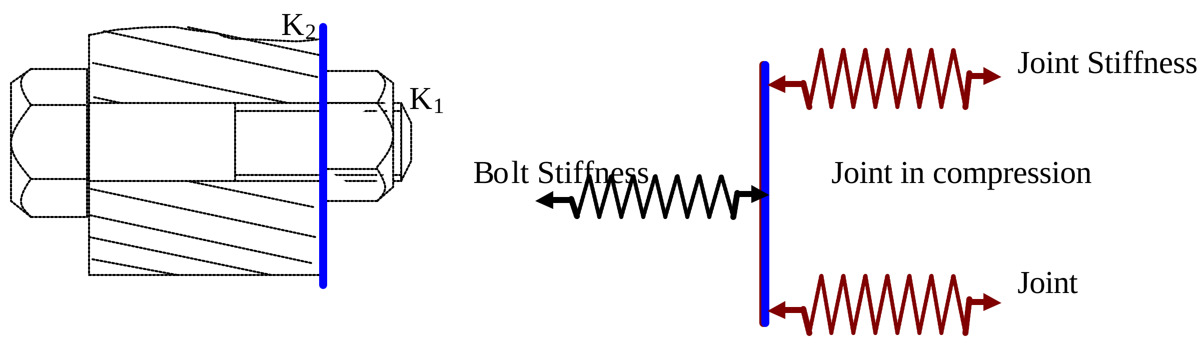
Operator Skill –

Parameter Control – The accuracy of control of specified parameters such as Torque, turn, threshold toque, etc.

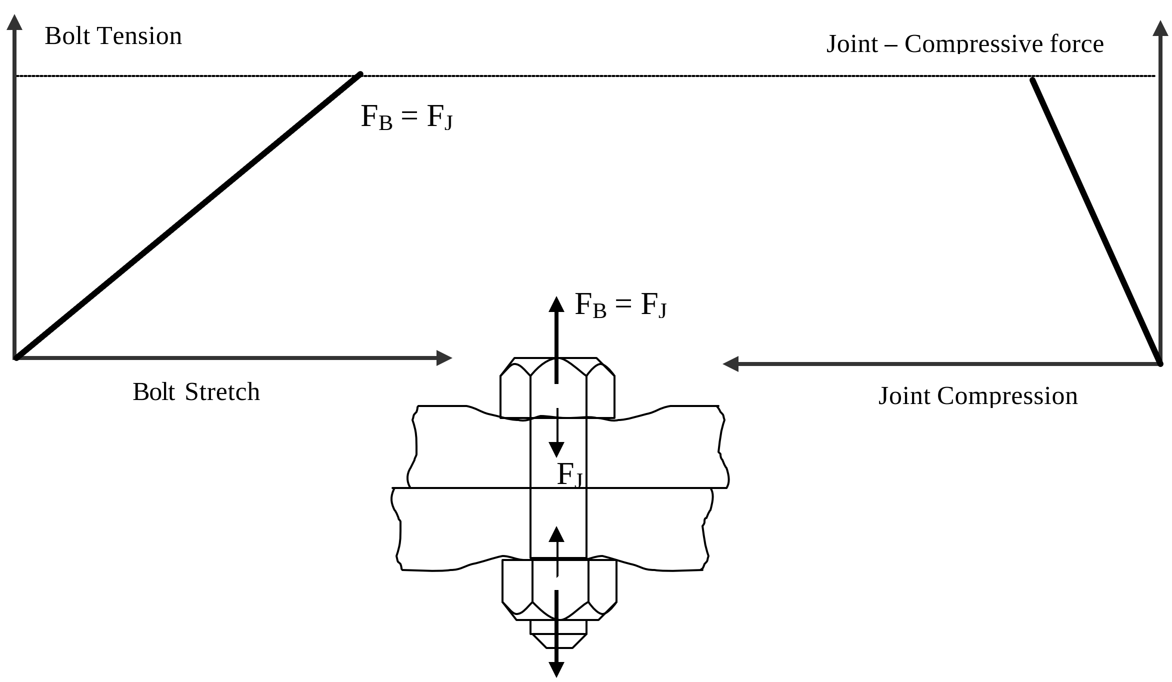
Short Term Relaxation –
Long Term Relaxation –
Thermal Effects –
External Loads –
External Loads –
Quality of Parts –
etc.

Joint Behavior and Geometry under External Load

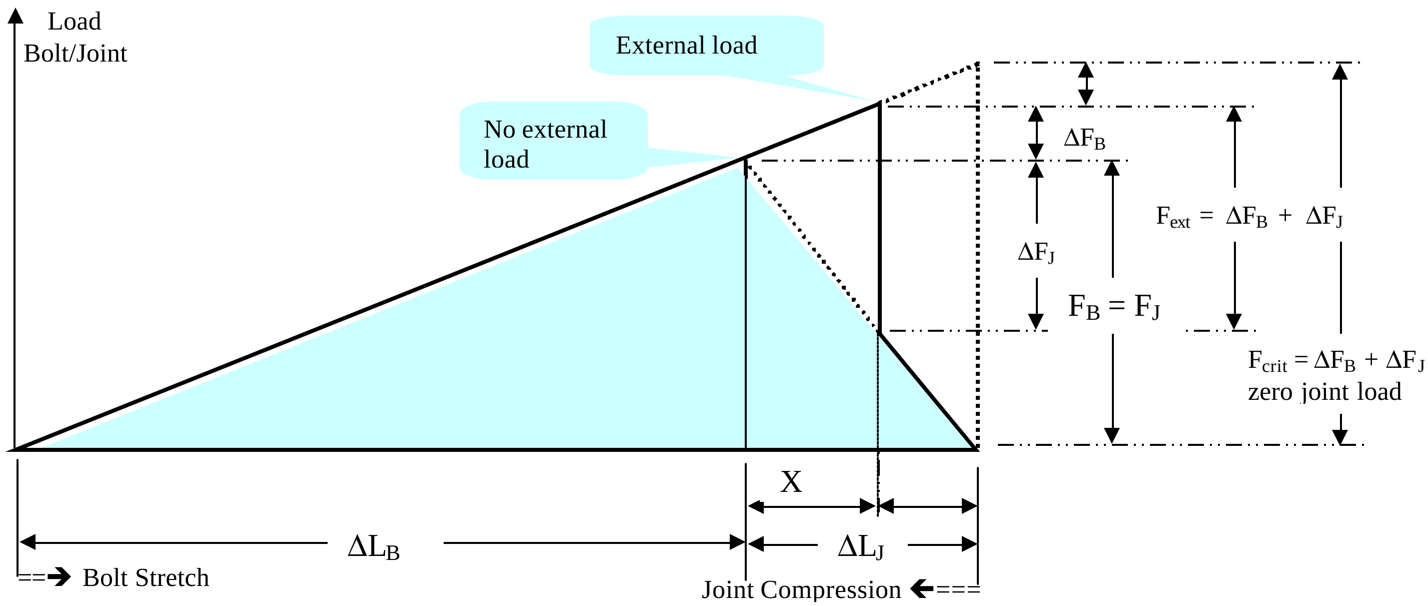
In earlier discussion we established that bolt and joint behave like two springs placed in series. Which means that the tensile force in the bolt equals the compressive force in the joint members.



When the bolt is tightened, the bolt and the joint deform by different amount depending on their stiffnesses (Joint stiffness > bolt stiffness). In absence of any external load, the bolt tension is always equal to the compressive load on the joint and behaves the following ways. (The plots below have the same vertical and horizontal scales, but different origin)



When external load of amount F_{ext} is applied to the bolt, the bolt stretches and joint deformation can be drawn in a graph adjacent to each other when stiffness values are known.



How is the joint stiffness determined?

While bolt stiffness is easily calculated from the material properties and standard bolt geometry, this cannot be easily done for the joint. The joint of interest is very specific and its geometry (stressed area) may be difficult to determine. However, the joint stiffness can be calculated from experimentally determined torque and turn data discussed later in this course.

When stiffness values of bolt and joint are known, the above diagram can be drawn to scale for any value of the preload and some key questions can be answered from the diagram or calculated using the formulas shown below.

Key questions:

What is the additional extension of bolt under external load?

At what external load will the joint separate from the bolt?

Additional stretch on bolt due to external load F_{ext} on the bolt head.

$\Delta F_B = K_B X = F_{ext.} [K_B /(K_B + K_J)]$

Since $\Delta F_B = K_B X$ and $\Delta F_J = K_J X$
$F_{ext.} = \Delta F_B + \Delta F_J = X (K_B + K_J)$

Or $X = F_{ext.} /(K_B + K_J)$

Note: ΔF_J is the reduction of compressive load in the joint which must be overcome by the external load before increasing the load on bolt.

External load ($F_{ext} = F_{crit}$) needed to create separation between bolt head and the joint surface (zero joint load).

$F_{crit} = F_B + \Delta F_B$ (at zero joint load)

Also,

ΔF_B(at zero joint load)
$= K_B \Delta L_J = K_B (F_B/K_J)$

$F_{crit.} = F_B + K_B (F_B/K_J) = F_B (1 + K_B / K_J)$

Note: $K_B \Delta L_J$ is load necessary to stretch the bolt by the amount joint compression before external load is applied.

Example:
Bolt stiffness, $K_B = 200 \times 10^3$ lbs/in.
Joint stiffness, $K_J = 600 \times 10^3$ lbs/in.

Preload in the bolt, F_B = 8,000 lbs. $F_{ext.}$ = 4,000 lbs.

Analytical Solution:
Bolt extension due to external load, $X = 4{,}000 \times 10^{-3} / (200 + 600) = 5 \times 10^{-3}$ in.

Critical load, $F_{crit.} = F_B (1 + K_B / K_J) = 8{,}000 (1 + 200/600) = 10{,}640$ lbs.

Graphical Solution:

Step1. Select a graph paper with vertical and horizontal grid lines. Select origin at O and draw axes OY and OB.

Step 2. Select a suitable scale such as Y-axis: 2,000 lbs/unit, X-axis: 5×10^{-3} in/unit.

Step 3. From given bolt load (equals joint load), calculate

Bolt stretch = $8{,}000/(200 \times 10^3) = 40 \times 10^{-3}$ in. which is 8 units along X-axis.

Draw line OA with A 8 units to the right from O.

Step 4. Identify point C 4 units above point A as the 8,000 lbs bolt load. Draw a line to join point O to C to represent bolt stiffness.

Step 5. Calculate

Joint compression = $8{,}000/(600 \times 10^{3}) = 13.33 \times 10^{-3}$ in.

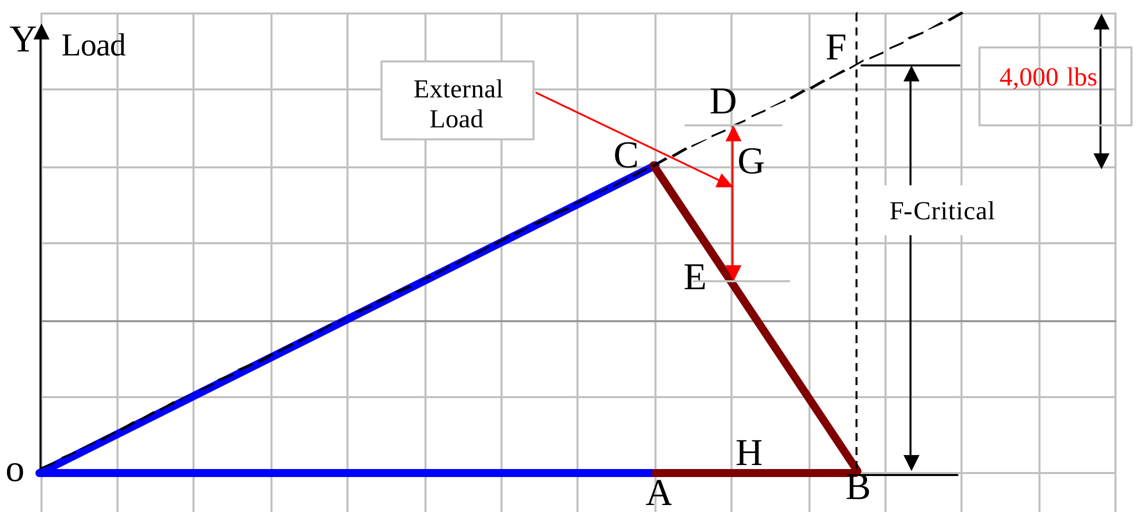
which is 2.67 units to the right of A along X-axis. Mark this point as B and join B and A. Pont B represent the origin of joint load-compression plot.

Step 6. Extend line OC to point F and beyond.

Step 7. Prepare a vertical line of 2 units in length that represents 4,000 lbs of external load as shown at top right in the diagram below.

Step 9. Slide this line toward point C while keeping it vertical at all times until it touches line OF at D and line BC at E. Measure distance **CG (1 unit = 5 x 10^{-3} in) as the additional bolt stretch due to external load**, DE.

Step 10. Draw a vertical line from point B. Let this line intersect bolt stiffness line OF at F. The vertical distance **BF (5.3 units = 10,600 lbs) now represents the critical external load**.



AC = Bolt Load, OA = Bolt Stretch, AB= Joint Comp., DE = External load, BF = Critical load

Scale: Y-axis: 2,000 lbs/unit, X-axis: 5 x10^{-3} in/unit

Below are a few additional information you can obtain from the plot:

a. Total bolt stretch under 4,000 lbs. external load is OH = 9 units or 45 x 10^{-3} in.
b. Total bolt load when 4,000 lbs. external load is DH = 4.5 units or 9,000 lbs.
c. Bolt load is the same as the external load when it reaches the critical load limit (BF).

Summary: Bolt Strength Information

Strength of a Bolt

- Proof Load – point just before any permanent deformation
- Tensile or Yield Strength –load at o.20 to 0.5% strain
- Ultimate Strength – maximum load without rupture

Proof < Yield < Ultimate

Tensile Stress Area

Tensile stress area of a bolt is used to calculate *proof, tensile or yield strength* of bolts. It is also used to calculate *nominal tensile stress* under a load.

Most common formula for tensile stress area is:

$$\mathbf{A_s = 0.785\ (D - 0.9743/N\ )^2} \quad \text{sq. in.}$$

Where D = Nominal diameter of bolt
N = Number of threads per inch (TPI)

Static Strength of a Bolt

- Calculated bolt strength indicates available clamping force.
- **Proof load** is the maximum usable strength of a bolt.
- Bolt breaks in one of four ways:
  - Bolt body breaks at a stress concentration point
  - Bolt threads strips (thread shear failure)
  - Nut threads could strip (uncommon)
  - Bolt and thread strip together

Static Strength Computation

Find *yield strength* for ¼ -20 UNC class 2A Inconel 600 Bolt (ambient yield stress from table). s = 37,00 psi.

$$A_s = 0.785\ (D - 0.9743/n\ )^2 = 0.0318$$

$$F = 37,000 \times 0.0318$$

$$= 1,178 \text{ lbs.}$$

Contact Stresses

- Washers help avoid embedment and stress distribution. Thicker & harder washers are better.
- Stress variation along the bolt of 8:1 is common.
- Bolt to bolt stress variation depends on spacing and sequence of tightening.

Bolt Stress Contour

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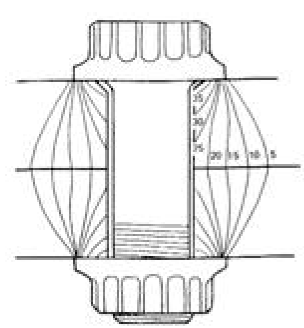


- Stress under nut may exceed yield strength.

Thread Forms and Profiles

- Only a few among hundreds available forms and profiles are common UNJ ACME, WITWORTH, BUTTRESS...
- **Unified Thread Forms** is most common in the Western World
  - Included angle 60 deg.
  - UN is flat bottom
  - UNR form has rounded bottom
  - UNJ has more generous rounding

Basic Thread Profile Standards

- UN/UNR - ASME B1.1, FED Std H 28/2B
- M – ASME B1.13 and B1.18 (for commercial use)
- UNJ – MIL – S-8879F
- MJ – ASME B1.22

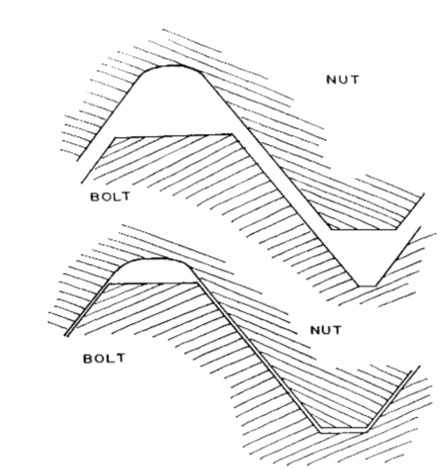
Constant Pitch Thread Series

- All UN/UNJ/M & UNJ/MJ series have constant pitch defined by threads per inch (TPI)
- Standard TPI are:
  - 4, 6, 8,12, 16, 20, 28 & 32 threads per inch.
  - Course (UNC), Fine (UNF), Extra Fine (UNRF) & Special (UNS) types of treads have different TPI.

Thread Allowance, Tolerance, and Class

- Allowance – minimum clearance between male and female thread profiles
- Tolerance – permitted variation of the allowance
- Fit – dictated by the combination of *allowance* and *tolerance*

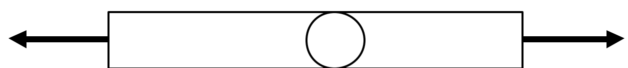
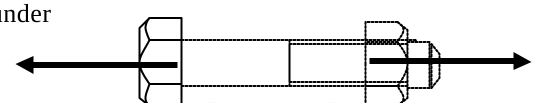
Thread Profiles (Male & Female)

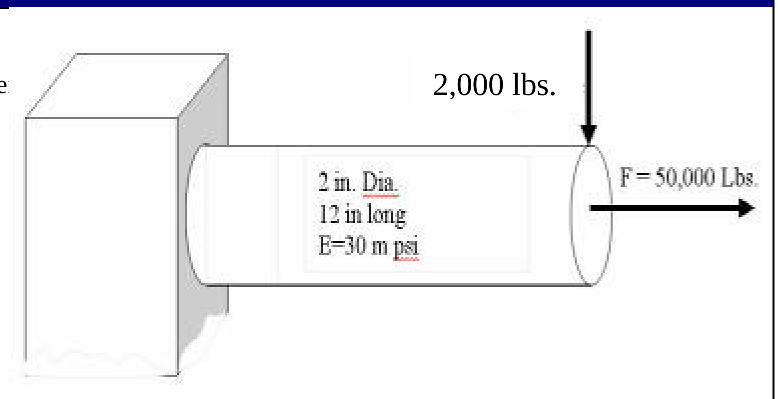
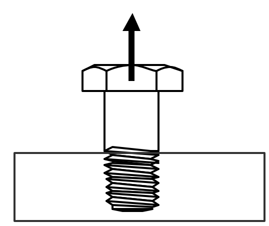


Thread Classes of Fits (Class 1, 2 & 3)

- 1A, 2A & 3A for male threads
- 1B, 2B & 3B are for female threads
- 1A/B makes the loosest fit – used for rough work in dirty environment. Easiest to assemble.
- 2A/B is used mostly for normal applications and are found in the hardware stores.
- 3A/B offers the tightest class of fit. Used for precision applications.

Note: 1A & 2A have same *allowance but* different *tolerance*. 3A has “zero *allowance*”.

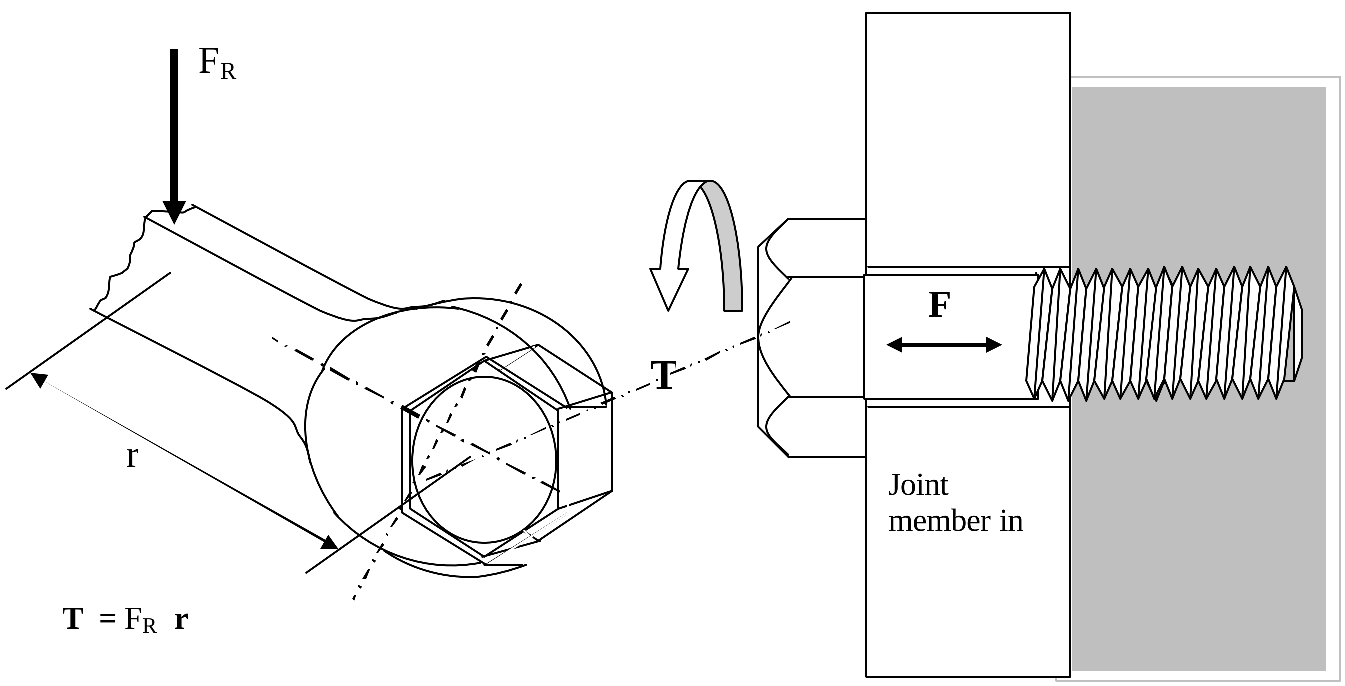
Topics	Group Exercises
Equivalent Stiffness	**A-4** **a:** A mattress for bed uses 30 coil springs of same stiffness to support a human body. If the compression is limited to 1.50 in., what should be the stiffness of the springs that will support an average 300 lbs. body weight?
	b: A suspension is designed using a leaf spring (300 lb/in) and a coil spring (500 lb/in) placed in series. What will be the total deflection under 250 lbs.
Stiffness of a Rod K = AE/L	**c:** A steel rod of 1.25 in diameter is used to hold two walls in place.  What force is developed in the rod when it is it is stretched by 0.200 in. (Given Length of rod = 240 in, E = 30 x 10^6 psi)
Bolt Stretch	**d:** Find the total extension of the bolt under 12,000 lb. load. (E=30mpsi)  Dia(1) = 0.75 in, Dia (2) = 0.68 in L_1 = 4 in L_2 = 2.5 in

Topics	Group Exercises
Axial and Bending Stresses Ix = ? $D^4/64$ J = ? $D^4/32$ s_b = M (D/2)/I t_t = T (D/2)/J	**A-5** **a:** Find the maximum te axial forces as shown. 
A = ? $D^2/4$ P = A x Stress	**b:** Four bolts are used to attach a plate used to lift a machine weighing 350,000 lbs. Based on axial loading, what should be the minimum diameter of the bolts made of materials with stress limit of 45,000 psi?
Strength of Materials Shear Stress	**c:** Find minimum number of thread engagement of bolt required to avoid thread shear. Design strategy: Shear capacity exceeds tensile load Bolt dia. = 1.00 in, Pitch dia =0.85 in, Pitch = 0.078 in. Max. tensile stress = 30,000 psi, max. shear stress = ½ Tensile stress Use thread shear area/turn = ? (Pitch dia) x Pitch 
Bolt Strength Calculation	**A-6** Calculated the proof load and rupture strength of a 1 - 8 UNC 2A bolt made of steel with yield stress = 35ksi. Assume that the rupture point of this material is 1.5 times the yield point. Hint: Calculate stress areas of bolt using the formula $A_s = 0.785\,(D - 0.9743/n)^2$

Section – B

Bolted Joint Design Strategies and Assembly Considerations

Torque and Tension Relationship



The bolt head is turned by use of a wrench. Force (F_R) applied on the wrench handle at a distance (r) from the center of the bolt head produces a torque T.

As the applied torque turns the bolt thread, the engaged threaded portion of the bolt pulls the bolt to the right (Figure at right above, one thread pitch distance each turn). Once the bolt head touches the joint surface, it prevents the bolt from sinking into the surface. Additional torque from this point onward is resisted by bolt (I) stretch, which produced tensile load (clamping force or preload) in the body of the bolt. As the tensile load is generated in the bolt, the friction forces at the contact points, (II) between bolt head and the joint surface, and (III) between the bolt and the joint threads, also increase proportionally. In other word, the applied torque (T) is required to overcome resisting torque arising from:

(I) Bolt stretch, one thread pitch for every turn (T_P)
(II) Frictional resistance between the contacting threads(T_T)
(III) Frictional drag between bolt head (or washer) and joint surface(T_B)

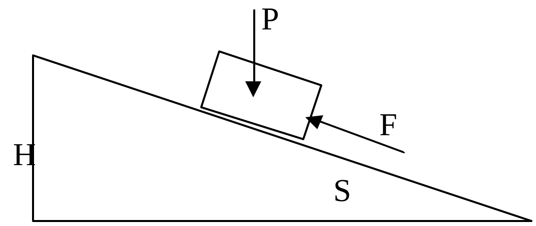
Thus

$$\mathbf{T} = T_P + T_B + T_T$$

Torque necessary to overcome thread pitch

(Ignoring friction for the time being)

In case of a transational force in a system shown at right, the force F Necessary to push the body up (force P is appliedon the body), can be be easily found by considering the conservation of energy.

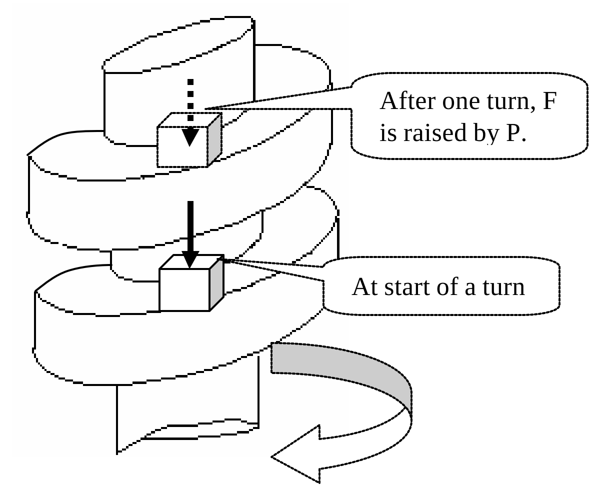


Work done to push the body over a distance S is same as the increase in the energy in lifting the body by a height H against the force P.

That is $\quad F\,S = P\,H$, or $F = P\,(H/S)$

Likewise, for a rotational motion, as in one turn of the thread of the bolt,

P = Thread pitch (distance in one turn)
F = Force in the bolt (tension/clamping force)
T_P = Torque required to overcome pitch
(cause bolt stretch by pitch distance)



Energy spent to complete one turn (angle 2Π) by applying torque is $2\Pi\, T_P$.

Work done in stretching the bolt by a distance P against the force F is FxP.

Thus,

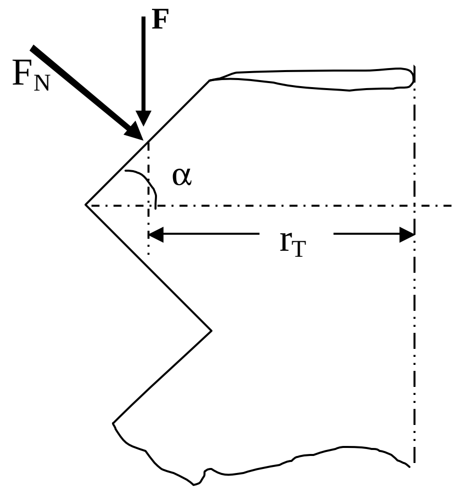
$$2\Pi\, T_P = FxP$$

or $\quad \mathbf{T_P = F\,(P/2}P\mathbf{\ )}$

(II) *Torque necessary to overcome thread friction*

Frictional drag is proportional to the force normal to the thread surface. A component of the force normal to the surface (F_N) resists the bolt force (F).

$$F_N = F / \cos \alpha$$



Friction force between thread which resists the motion is $\mu_T \times F_N$.

μ_T = Friction coefficient between threads
r_T = Effective radius of thread
α = Half thread angle

$T_T = \mu_T F_N . r_T = F \mu_T r_T / (\cos \alpha)$

Or

$$\mathbf{T_T = F\ m_T\ r_T/(Cos\ a)}$$

(III) *Torque necessary to overcome bolt head/washer friction* (T_B)
(Bearing Component)

μ_B = Bearing surface Friction coefficient
r_B = Effective bearing radius

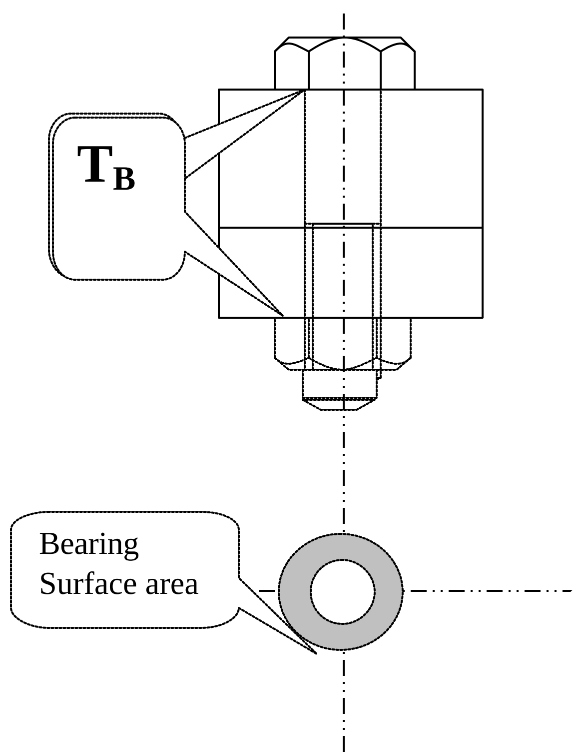
Friction force at bearing surface is $\mu_B \times F$.

Therefore, the torque necessary to overcome the bearing friction is

$$\mathbf{T_B} = \mu_B \mathbf{F} r_B$$

Or

$$\mathbf{T_B = F\ m_B\ r_B}$$



Since

$$T = T_P + T_B + T_T$$

$$\mathbf{T = F\ [\ P/(2P\ ) + m_T\ r_T/(Cos\ a\ ) + m_B\ r_B]}$$

Where

T = Toque applied to the bolt
F = Force in the bolt (tension/clamping force)
P = Thread pitch (distance in one turn)

μ_T = Friction coefficient between threads
r_T = Effective radius of thread
α = Half thread angle
μ_B = Bearing surface Friction coefficient
r_B = Effective bearing radius

The equation above shows that the input torque (T) is resisted by three reaction torques:

$F\ P/(2P\ )$ - 10% (shown next)
Is produced by the inclined thread plane or bolt thread on joint (or nut). This is generally called the bolt stetch component.

$F\ m_T\ r_T/(Cos\ a\ )$ - 40%
Is a reaction torque due to friction between the threads?

$F\ m_B\ r_B$ - 50%
Is the reaction torque generated by the friction between the bolt head/nut and the washer or the joint?

The ratio of T/F can be calculated for a Bolt as

$$T/F = [\,P/(2P) + m_T r_T/(\mathrm{Cos}\ a) + m_B r_B\,]$$

$$= [\,1/(13\mathrm{x}2P) + 0.15\mathrm{x}0.225/(\mathrm{Cos}\ 30) + 0.15\mathrm{x}0.32\,]$$

$$= 0.0122 + 0.039 + 0.048$$

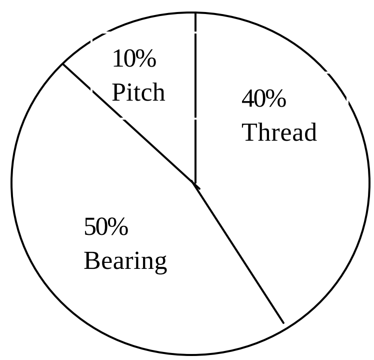
$$= 0.099$$

Example data
1/2 - 13 UNC Thread

T = Toque applied
F = One unit (=1)
P = 1/13 inch
$\mu_T = 0.15$
$r_T = 0.225$ inch
$\alpha = 30^o$
$\mu_B = 0.15$
$r_B = 0.32$ inch

Percentage of torque used to overcome the Resisting torques are as follows.

0.0122/0.099 = 12 %, due to pitch (≅10%)
0.039/0.099 = 39% due to thread (≅ 40%)
0.049/0.099 = 49% due to bearing (≅50%)



Total Torque Breakdown

In addition to torque required to tighten the bolt, there could be torque required to run down the bolt/nut against interference or inserts. These kinds of reaction torque are called *prevailing torque* (T_P).

Example: The torque required to run down a lock nut which has a plastic insert in the thread.

With the prevailing torque included, the torque equation will be:

$$T = F\,[\,P/(2P) + m_T r_T/(\mathrm{Cos}\ a) + m_B r_B\,] + T_P$$

In many common situation the prevailing toque is absent, $T_P = 0$.

The Short-Form of Torque vs. Tension Relation

A close examination of the torque vs. tension equation shows that thread pitch, P, radii r_T and r_B are all dependent on the nominal diameter (D) of the bolt. Furthermore, the friction coefficients are also constant for the material.

Thus,

$$T = F\,[\,P/(2\Pi) + \mu_T\, r_T/(\cos\alpha) + \mu_B\, r_B]$$

$$= F\,[\,K_1 D + K_2 D + K_3 D]$$

$$= F D\,[\,K_1 + K_2 + K_3\,]$$

or

$$\mathbf{T = K\,D\,F}$$

Where
T = Input torque on the bolt or nut (in-lb. or N-mm)
F = Tension in the bolt (lb. or N)
D = Nominal diameter of the bolt (in. or mm)
K = Constant called "Nut Factor" (dimensionless, no units)

K_1, K_2, and K_3 are also constants and $K = K_1 + K_2 + K_3$

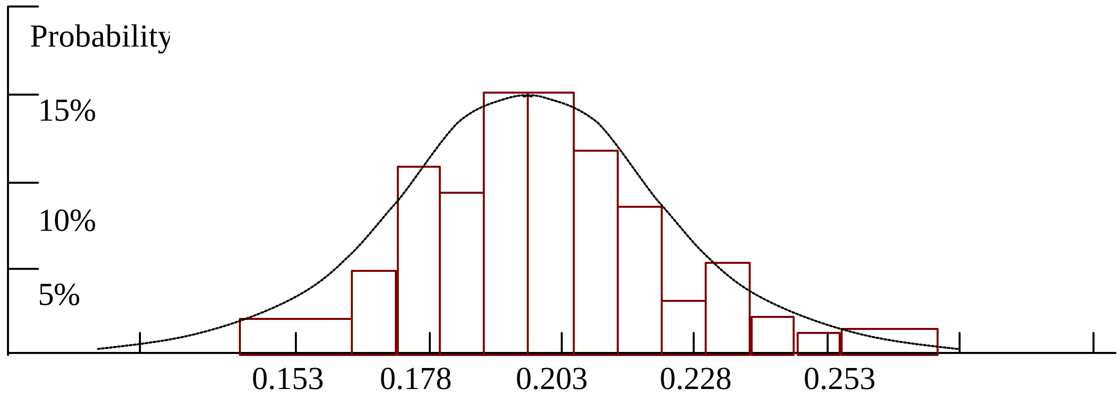
The Nut Factor K is a general-purpose, experimentally determined constant. It

- is not a coefficient of friction (friction has influence)
- is anything and everything that affect the relationship between torque and tension in the experiment – including friction, torsion, bending, plastic deformation of threads, etc.
- can only be determined experimentally by testing a number of samples.

$$T = K\,D\,F$$

is a simple equation and K, the Nut Factor is an all inclusive constant which captures effects of all that affect torque-tension relationship. Prescribing a torque (T) that is needed to achieve a certain tension (F) will have been a simple matter if only K had a unique value for each application. Unfortunately, the Nut Factor varies widely from sample to sample for the same application.

Graph below shows the Histogram of K values reported for as-received steel fasteners from a large number of sources.



Nut Factor (K)

Because of the associated variability, the Nut Factor is defined in terms of a mean value and the scatter. Available experimental data for various materials, procedures, lubricants, types of tool, etc. provide a guideline for design specification. The exact value of K, however, must be determined experimentally using the production application setup.

	Nut Factor (K)		
Fastener Materials & Coatings	**Min. Val.**	**Mean**	**Max.**
Pure aluminum coating on AISI 8740 alloy steel	0.42	0.52	0.62
As received, mild or alloy steel on steel	0.158	0.20	0.267
Machine Oil	0.04	------	0.18
Zinc plate (Waxed)	0.071	0.288	0.52
Zinc plate (dry)	0.075	0.295	0.53

Uncertainty in Assembly Caused by Variability in Nut factor (K)

Automatic nut turner can accurately apply the specified torque during assembly operations (1 –2% accuracy for electric turner). But whether the specified torque will produce the desired bolt tension will depend on the Nut factor (K). As the Nut Factor vary sample to sample, the tension obtained will also vary. As seen with reported experimental results, the range of variability (Max. – Min value) may be several times that of the mean value.

Values of K for
Zinc plate (dry) Min. = 0.075 Mean =0.295 Max = 0.53

Since T = K D F

F = T / (K D)

If a fixed toque T= 1000 in-lb. Is applied on an 1 inch diameter bolt, then the force obtained may be different depending on the value of K as shown below.

High F = 1,000/ (1 x 0.075) = 13, 333 lbs. (With low K)

Mean F = 1,000 / (1 x 0.295) = 3,389 lbs.

Low F = 1,000 / (1 x 0.53) = 1,886 lbs. (With high K)

Observations:

1. Can the bolt withstand the high load? Will it yield or fracture?
2. Is the low load enough for the desired clamp load? Will the joint loosen?
3. What percent will have low load? What percent will fail?

If we were to calculate the toque necessary to obtain a fixed F= 5,000 lb. instead, then the torque necessary will widely vary.

Using T = K D F
Low T = 0.075 x 1 x 5,000 = 375 in-lb.
Mean T = 0.295x 1 x 5,000 = 1,475 in-lb.
High T = 0.53 x 1 x 5,000 = 2,650 in-lb.
(In assembly operation, torque is never varied/adjusted from sample to sample)

Topics	Group Exercises – B	
Torque Breakdown	**B-1** **a:** In an assembly, the torque required to tighten the bolted joint is 500 in.lb. How much torque would be reduced, if the frictional resistance between the nut/bolt-head and the joint is fully eliminated by some means? (Hint use the standard breakdown)	
Torque Vs. Tension	**b:** The bolt of type shown at right is subjected to a torque, T = 1,200 in. lb. What is the expected tension (F) produced in the bolt. Hint: $\mathbf{T = [\,P/(2\pi) + \mu_T r_T/(\cos \alpha) + \mu_B r_B\,]\ F}$ (Ans: F = 12,121 lbs.)	Bolt Data 1/2 - 13 UNC Thread T = Toque applied F = One unit (=1) P = 1/13 inch $\mu_T = 0.15$ r_T = 0.225 inch $\alpha = 30^o$ $\mu_B = 0.15$ r_B = 0.32 inch
	c: If the friction under the bearing surface in the above example is reduced to one third the original value ($\mu_B = 0.05$), what will be the increased bolt tension when the same torque is applied.	
	d: If the friction between the contacting threads in problem B-1b above is increased to twice the original value (μ_T = 0.3.0), what will be the reduced value of the bolt tension when the same torque is applied.	

Topics	Group Exercises – B
Nut factor	**B-2** **a:** The experimentally determined distribution of the Nut Factor for a joint assembly is found to have Avg: 0.30 with Standard deviation = 0.07. If a fixed torque of 800 lb. in. is applied to a ¾ in dia bolt, what will be the expected minimum and maximum values of the force in the bolt? (Hint: Use F = T/(KD), Kmin = avg – 3 Std.Dev. etc., Ans: Fmax= 11,851/Fmin= 2,091)
Torque vs. Tension Relationship T = KDF F = T/(KD)	**b:** The measured value of the tension (F) after assembly of a ¾ in dia. bolt was found to be 3,500 lbs. What would the tension value be if the nut factor is reduced to half its value when the same torque is applied?
F = T/(KD)	**c:** In an assembly is designed to apply a torque (T) to generate a desired tension (F). If the bolt is replaced by another bolt of the same type and material, but 20% larger in diameter, show that this action will reduce the bolt tension by 20%. (Assume that the nut factor is the same)
K = T / (DF)	**d:** Values of torque and tension measured in the ½ in dia. bolt after assembly are: T = 80 lb.in. & F = 1,800 lbs. What is the expected value of the *nut factor* (K)?

Factors That Affect Tension Variability

Given a design, the Nut Factor (hence the torque necessary for a fixed tension) will vary because of the following.

Materials and Design factors

- type of materials
- hardness
- surface finish
- plating/coating
- lubricants
- hole clearance
- contact radii
- thread pitch
- flatness
- perpendicularity

Manufacturing Factors

- Tool accuracy
- Operator skill
- Control accuracy
- Part quality

Operating Environment

- Corrosion
- relaxation
- Gaskets
- Vibration
- Temperature
- External load

Misc. Factors

- Blind holes
- Dirt
- Chips
- Cross-threads
- Burrs
- Wrong size bold
- Metric/English
- Cleaning agent
- Deformed threads
- Insufficient thread depth

Book Values of K

Mild or alloy steel on steel	0.158 - 0.2 - 0.267
Pure aluminum, coated	0.420 - 0.52 - 0.62
Everlube coated threads	0.069 - 0.086 - 0.103
Fel-pro C54	0.080 - 0.132 - 0.230

Ref: Bickford[2], pp.141.

Friction
Operator
Geometry
Tool **A**ccuracy
Relaxation

FOGTAR
A word meaning TROUBLE in Tibetan language.
Ref: John H. Bickford

$T = K D F$

Where
K = empirically determined constant
D = diameter of bolt
F = Bolt tension

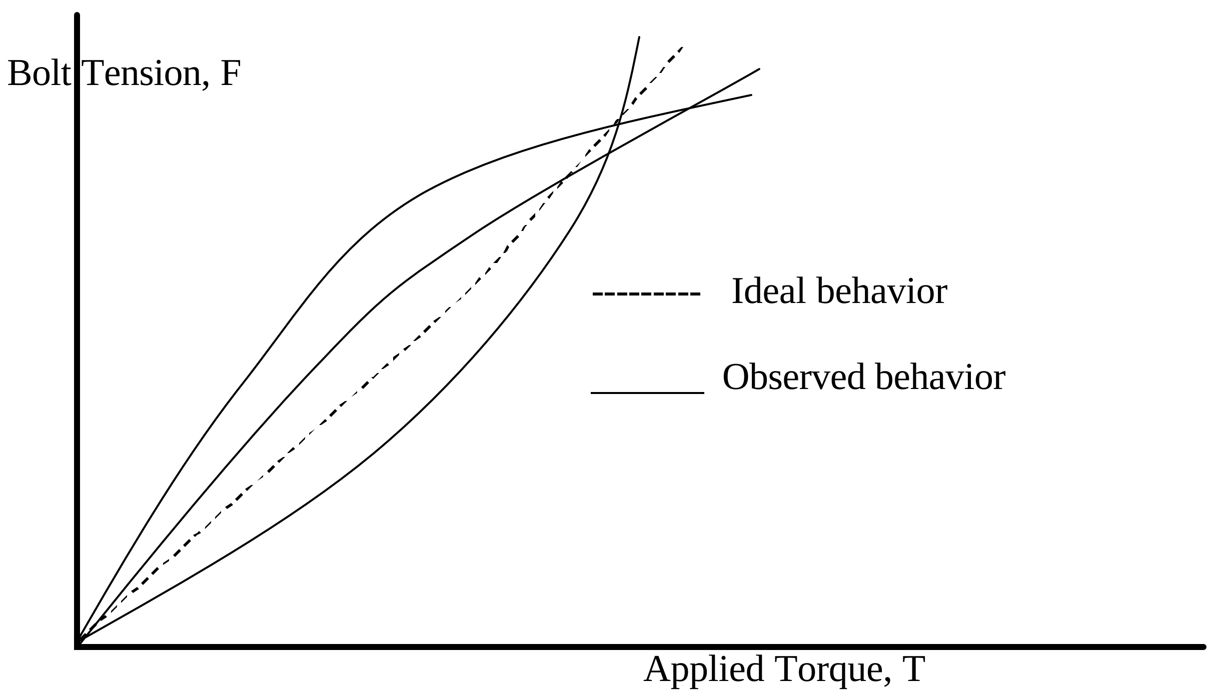
Also $\mathbf{F = T / (K D)}$

Observed in assembly

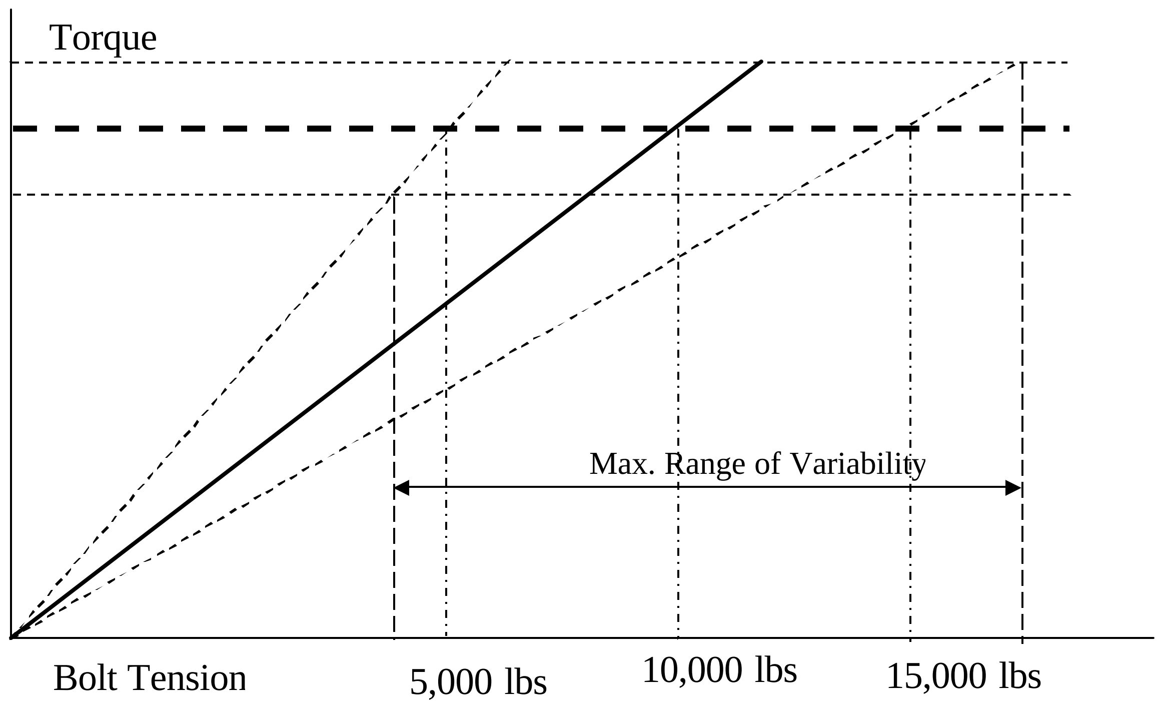
Hold torque fixed – clamp load will vary
Hold clamp load fixed – torque will vary

Join Assembly Challenge

GOAL – How to reduce the variability of the clamp load in assembly where only the torque can be held accurately. (Tension is what we need, but torque is the only practical thing that can be controlled)



Assembly Torque and Tension Behavior



For a fixed value of Torque, Bolt Tension (F) changes as the proportionality constant K depends on several joint parameters.

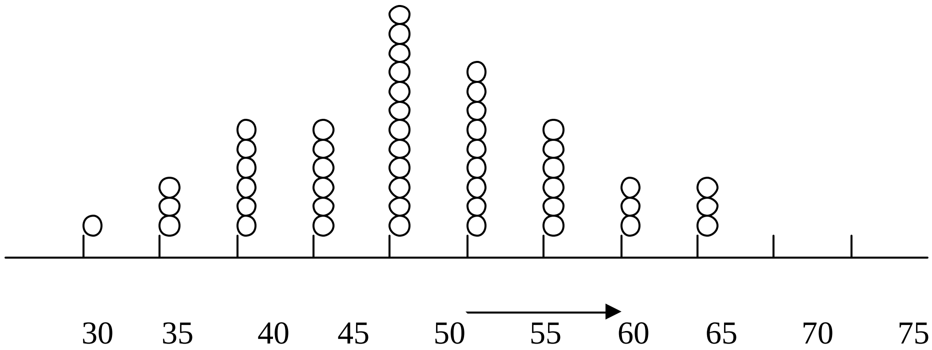
K = f (μ, geometry, tool accuracy, relaxation, operator, etc.)

Further, when **Torque varies due to application devices** (air or electric nut runner), the bolt Tension vary even more.

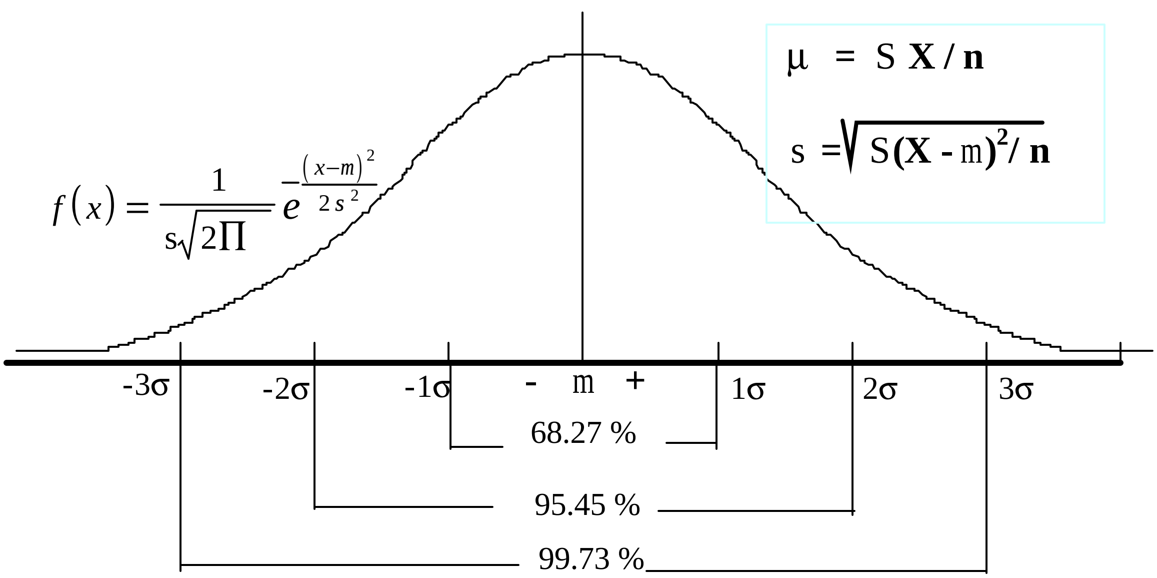
Note:
Prevailing Torque Fastener or Locknuts are used to prevent back-off torque. These are nuts or bolts modified to offer additional friction. Typical modifications include,

- Deflected beam
- Deflected thread
- Out-of-round thread

Process Variation and Process Capabilities



Variations ← Common Cause + Special Cause



Area under Normal Distribution

Data Collection procedure: Take a sample size of 5 and collect data from 20 samples (20 x 5 = 100 samples)

Calculations:

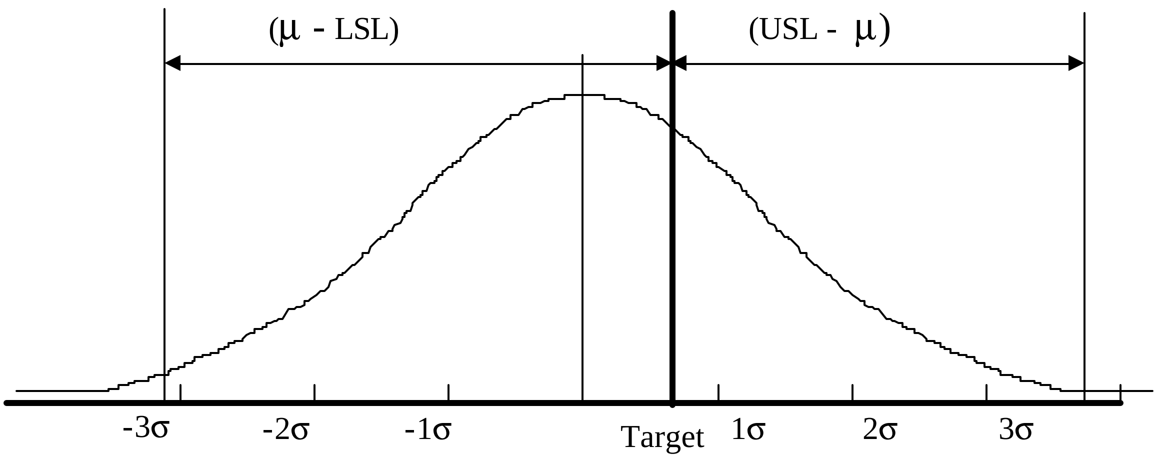
$$\mu = \bar{\bar{X}} = \Sigma\ \bar{x} / 20 = \Sigma\ x / 100$$

$$\sigma = S_x = \bar{R}/d_2 = \Sigma\ R/(20\ d_2)$$

R = Range
d_2 = Constant
= 2.326
for n = 5
Tot. samples = 5 x 20 = 100

Process Performance

C_p = Process Potential and C_{pk} = Process Capability



LSL = Lower Specification Limit , USL = Upper Specification Limit

Definitions:

$C_p = (USL - LSL)/ 6\sigma$

$$C_{pk} = \frac{|m - LSL| \text{ or } |(USL - m)|}{3s} \text{ whichever is smaller}$$

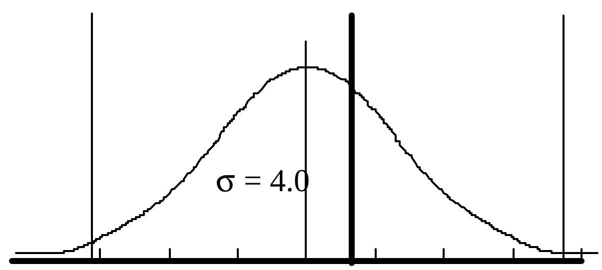
μ = average of the population of samples
σ = Standard deviation population of samples

Example:

$\mu = 75$, $\sigma = 7$, LSL = 60, and USL = 100

$\mathbf{C_p}$ = (100 - 60) / (6 x 7) = 40 / 42 = **0.95**

$\mathbf{C_{pk}}$ = (75 – 60) / (3 x 7) = 15 / 21 = **0.71**



Primary Influencing Factors Affecting Preload

There are two sources:
Tool variability and **Part** variability (+/- 3σ)

Tool variability contributes only a small portion of the total variability that affects preload.

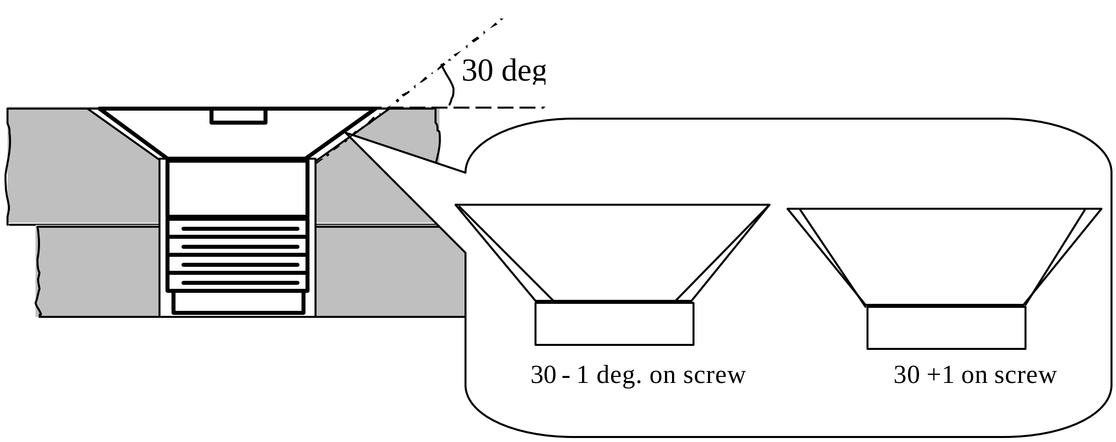
PART VARIABILITY (+/- 3σ) Tool Accuracy ➔	COMBINED **TOOL** + **PART** VARIABILITY (+/- 3σ)			
	Impact Wrench (40%)	Acceptable Tool (15%)	Preferred Tool (10%)	Precision Tool (5%)
40%	55%	45%	40%	40%
30%	50%	35%	29%	31%
20%	45%	25%	24%	22%
10%	42%	20%	15%	11

Observation:
- Part variability contributes most to the total variability.
- Impact wrences have worst effect on preload variability.

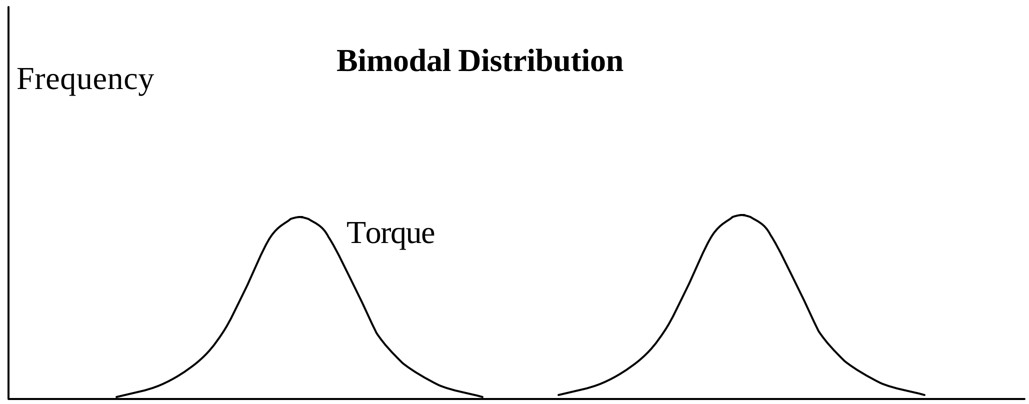
Reduce preload variability by controlling part to part variability.

Total variability, often, is also known and is stable. When possible, attention must be paid to improving and selecting parts that has higher consistency.

Case Studies



Torque applied to a bracket piece screw shown above resulted in bi-modal distribution shown below.



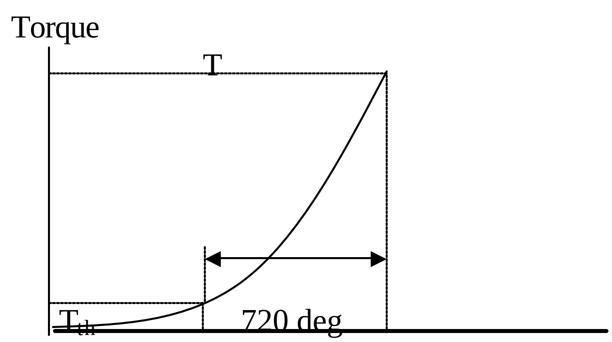
To avoid bi-modal distribution, tolerance on the screw was changed from 30 +/- 1 to 30 – 2, + 0. These forces contact on the upper rim of the screw head and mean torque at a higher range.

Bolt Tightening Strategies

Torque Rate (Joint Rate)

Soft Joint
A joint fastened with a bolt that Takes 720° (2 turns) turn of the bolt to reach the desired torque

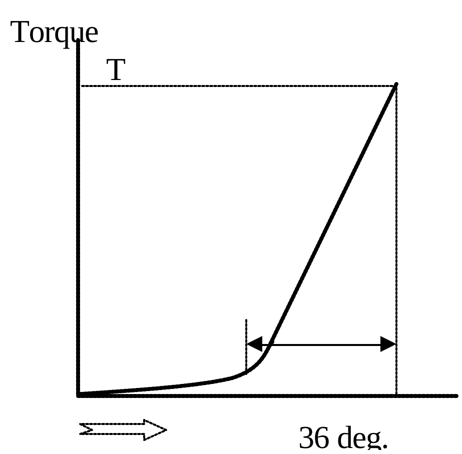
T_{Th} = Threshold torque



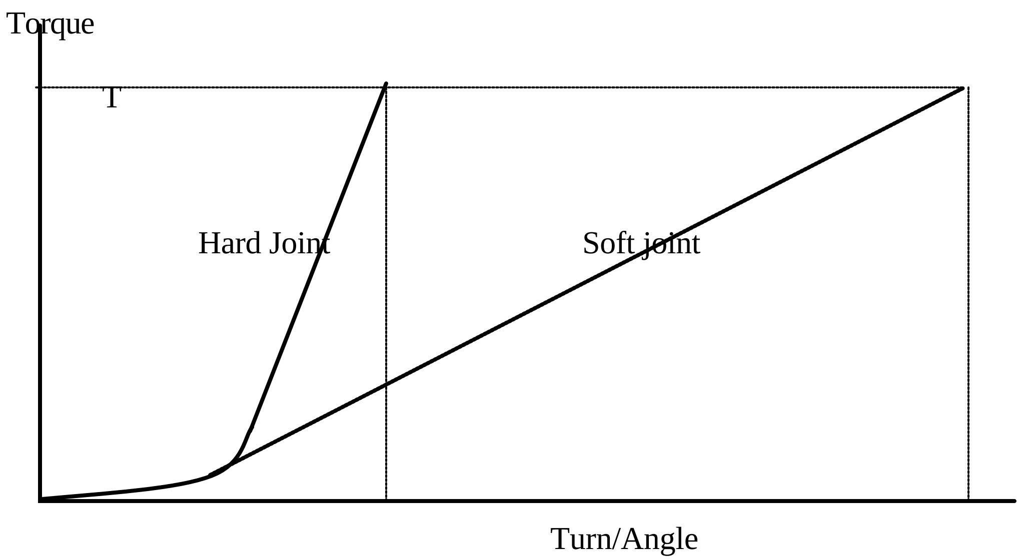
Example:
Tapping screws, valve cover gasket, Antenna mount, pipe joint, instrument Panel, Hose clamps, Engine mount, etc.

Hard Joint

A joint where the bolt reaches the desired torque within 36 ° turn (1/10 turn) of the bolt.

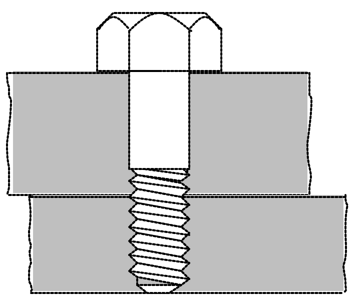


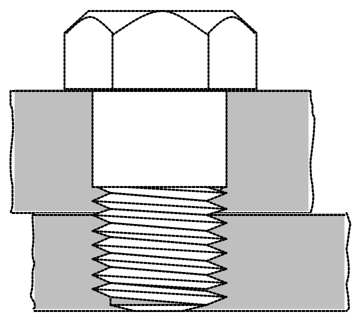
Soft joint

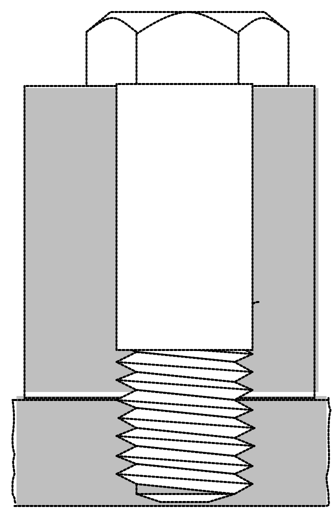


Relative Hardness of Joints

Which among the following joints is hardest?







Length = 2 in.
Nominal Dia = ½

Length = 2 in.
Nominal Dia = 1 in.

Assume that thread pitch and (Material: $E = 30 \times 10^6$ psi.) is same for all three joints.

Length = 4 in.
Nominal Dia. = 1 in

T / Angle = Joint Stiffness

If for a fixed amount of angle (say one turn), the torque required is more, then the stiffness of the joint will be also higher.

$T = KDF$ (F = axial force, D= diameter, K = constant)

Or $T = K D (AE/L) P$ (A = area, E= modulus, P = thread pitch)

Or $\mathbf{T = K E P (P/4) D^3 /L}$

When K, E, and P are fixed, whether a joint is harder or softer depends on whether T is more or less. *(Joint Stiffness α D/L)*

Comparing two joints of same length, but different diameter, the one with bigger diameter will be harder.

When two joints of dissimilar length but same diameter are compared, the one with longer length will be softer.

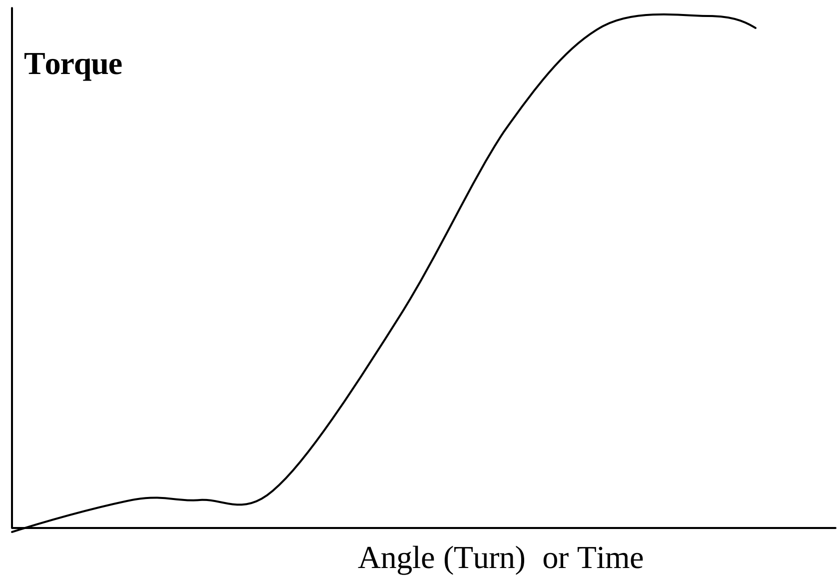
In industrial assembly involving mass production, most joints are assembled by turning the fasteners with power tools (NUT RUNNERS- pneumatic, electric, etc.). Controlling the torque applied by the power tool to obtain the desired bolt tension is achievable by several approaches.

Collected data

Torque, Angle, Time.

Common Plots:

Torque Vs. Angle
Torque Vs. Time



Three Strategies Commonly Used to Control Preload

Torque Rate Monitoring – Torque and Angle data are collected by TORQUE RATE MONITORING SYSTEM which includes a transducer mounted on the power tool (certified air-operated tool). The torque rate observed is compared with pre-established torque limits. In case of PEAK TORQUE monitoring system, the applied torque is held when the peak value is achieved.

(**System** – refers to the power tool and the associated computer hardware and software to collect and process the data)

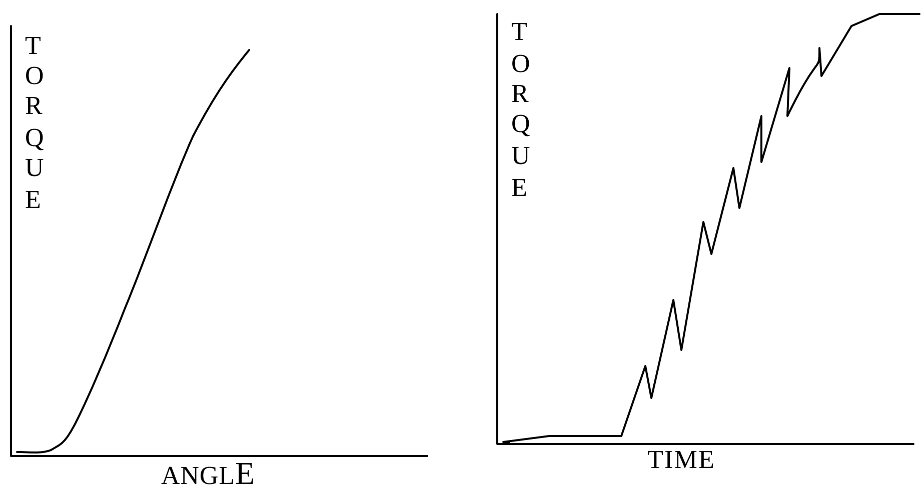
Tension Control – In TENSION CONTROL SYSTEM too torque and angle data gathered are compared with previously established mathematical model. The power tool is turned off when the conditions are met. It can work both in elastic and plastic region of the fastener load.

Yield Control – The YIELD CONTROL SYSTEM senses when the joint has yielded and turns the power tool off.

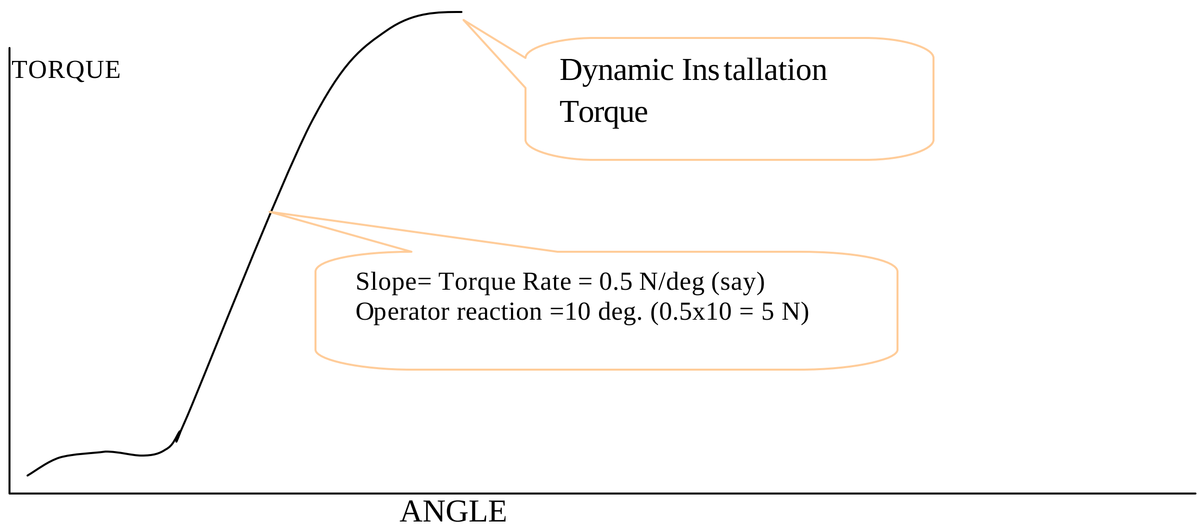
Fasteners are installed with the help of automatic torque wrenches (runners). As the torque limit is approached the motor speed is slowed and finally stalled when the torque limit is reached. The nature of torque vs. angle and torque vs. time plots look slightly different.

Torque vs. Angle diagram depends on material property, friction, geometry, etc. of the joint. The graph bends/flattens past the yield pint as the material property behaves. (A smooth curve)

Torque vs. Time diagram additionally includes the tool. It will bend toward the end of operation as the tool slows down. (a much more wavy line)



Torque that is specified for a joint is known as the **DYNAMIC INSTALLATION TARQUE**. This is the value of the final torque applied during the installation process. This torque is used to estimate the tension generated in the bolt. If only one torque is specified on the drawing, it is assumed to be Dynamic Installation Torque.



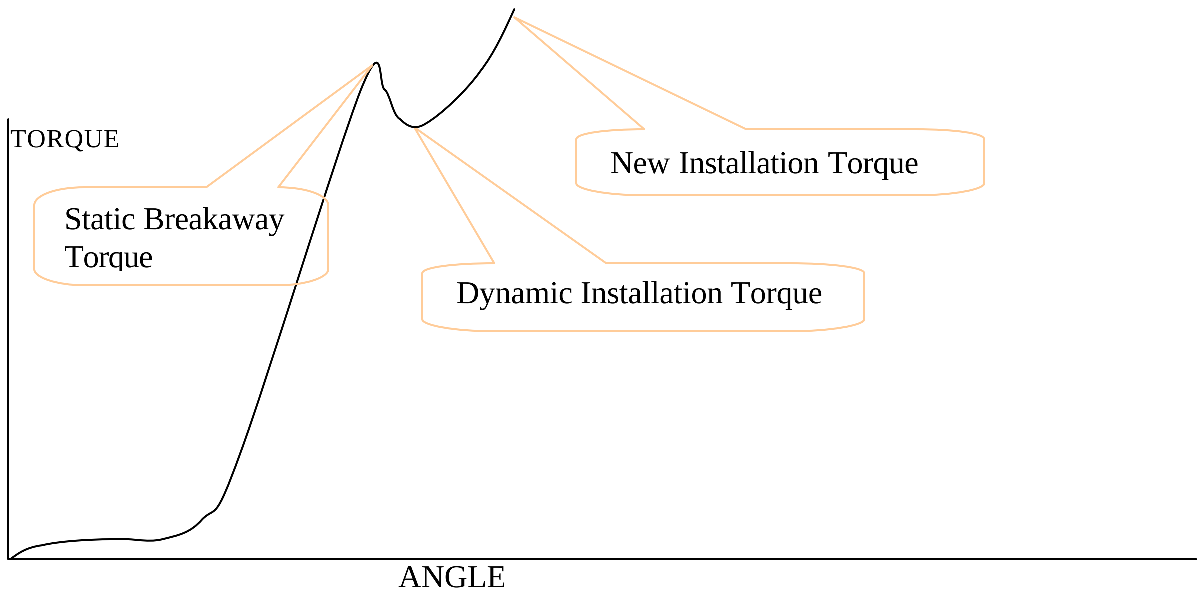
Inspection of Installed Torque

The torque measured by turning a fastener (which is already installed) in the tightening direction until it just begins to move is called the **STATIC AUDIT TORQUE** or BREAKAWAY TORQUE. The torque value falls past the Static Audit Torque, as dynamic friction coefficient is greater than the static value.

For quality inspection purposes Static Audit Torque is determined by taking a statistical sample (usually 100).

- Use electronic torque wrenches instead of conventional audit wrenches
- Consider relaxation and measure after established time delay

If the fastener is turned beyond its original position, the torque may be increased to a new value called the **NEW INSTALLATION TORQUE**. Electronic torque wrenches will record this as the highest value.



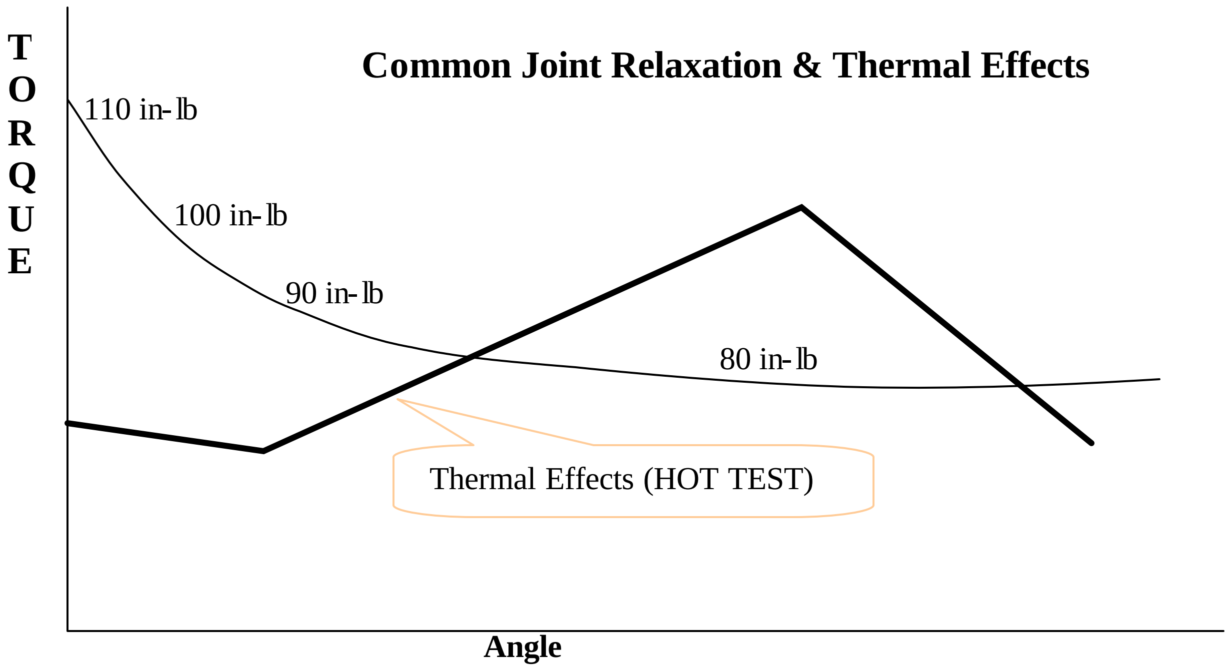
The greater of the Static Breakaway Torque and the New Installation Torque is called the **RESIDUAL TORQUE**. In the above diagram, the new installation torque is the residual torque.

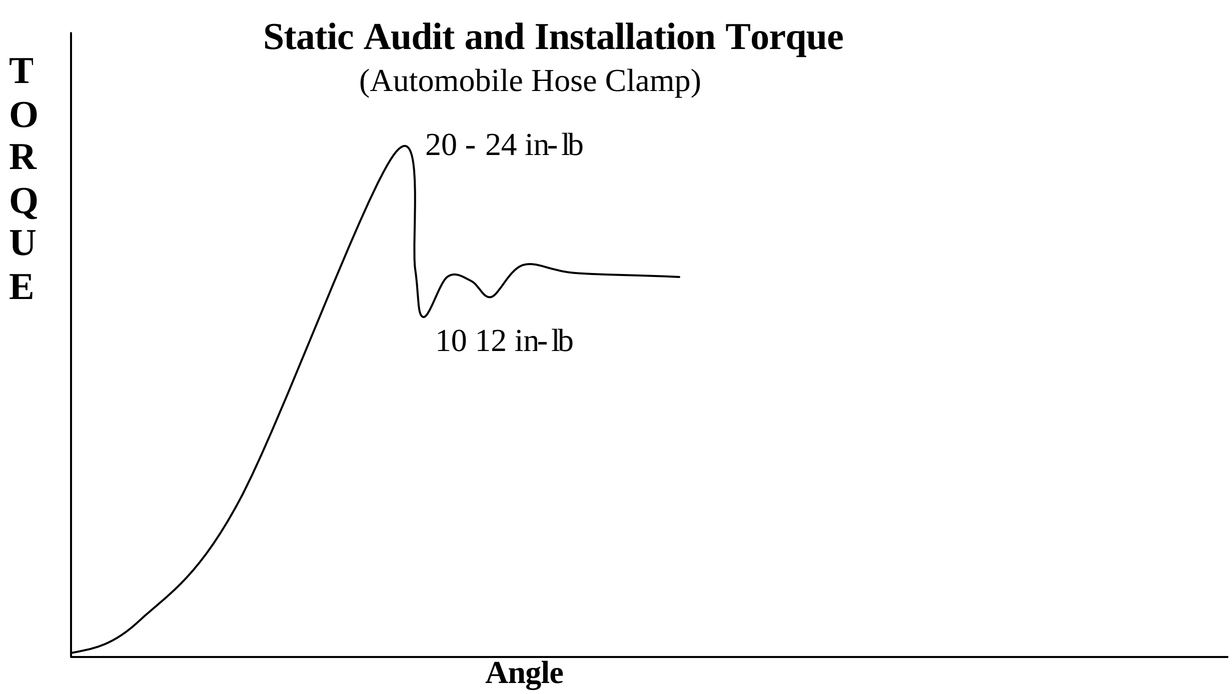
Field Observations

(a)Torque specified = 45 N
Auditor checks the torque to be 52 lbs. (45 + 7 lbs).
Recommends lowering of the power tool.
Soon finds the joint to be loose.

(b) ***How to audit torque for vehicle already assembled a few days later?***

Remember they will have relaxation.
Expect lower torque.





(c) Typical Breakaway (Static Audit) and Installation Torque Data

Data for a certain engine head bolt samples are as follows. In many applications audit torque could be quite different from the dynamic installation torque.

Static Audit Torque

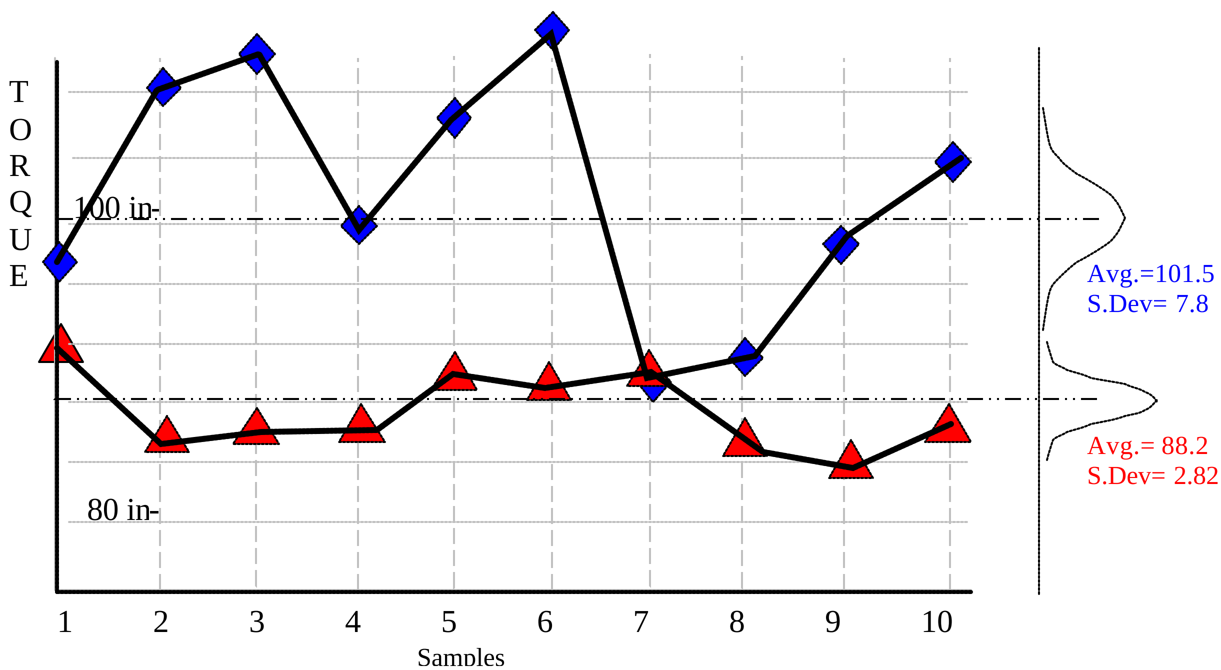
Sample size = 10 97 108 110 98 107 112 89 92 98 104

Average = 101.5 Std. Dev. = 7.835

Dynamic installation Torque

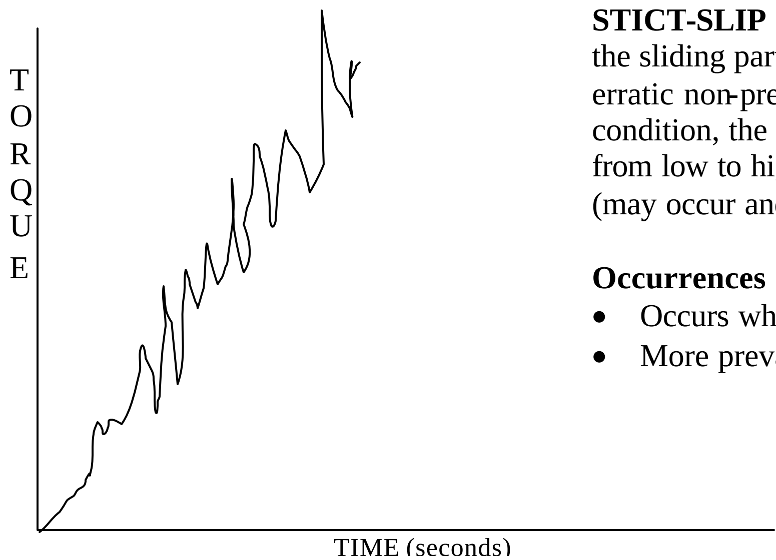
Sample size = 10 92 86 87 88 90 89 93 86 84 87

Average = 88.2 Std. Dev. = 2.82



(d) ***Stick Slip Friction***

Observed application: Connecting rod.



STICT-SLIP FRICTION is a condition where the sliding parts (nut or bolt) stick and slip in erratic non-predictable manner. Under this condition, the torque rate instantaneously changes from low to high and from high to low.
(may occur and go away randomly)

Occurrences

- Occurs when lubrication medium breaks down.
- More prevalent with S2 finish

Problems caused by Static-Slip

- Unreliable static audit torque
- Erratic torque scatter with certified power tool
- Low clamp force
- Less reliable data for tension control

Fixes to prevent Stick-Slip

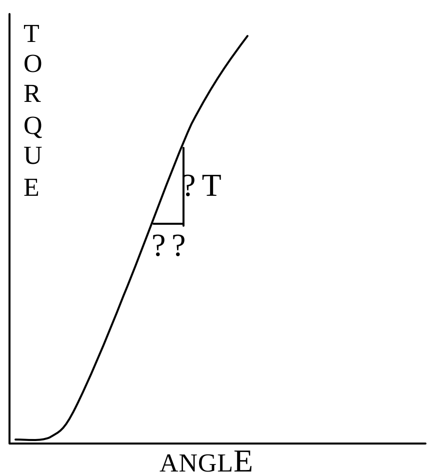
- S (plain) finish with SAE 50 oil
- Electroplated Cadmium
- Hardened Washer

Estimation of Joint Stiffness Using Torque Turn Data

Although, the torque-turn plot may not be linear at all angles, its slope at any point can be established by taking ratio of the experimentally determined incremental values of toque and turn.

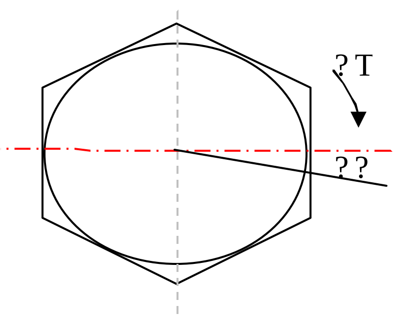
?T = Apply and measure value of torque applied to the bolt (in.lb.)
?? = Measure angle of rotation (in degrees)

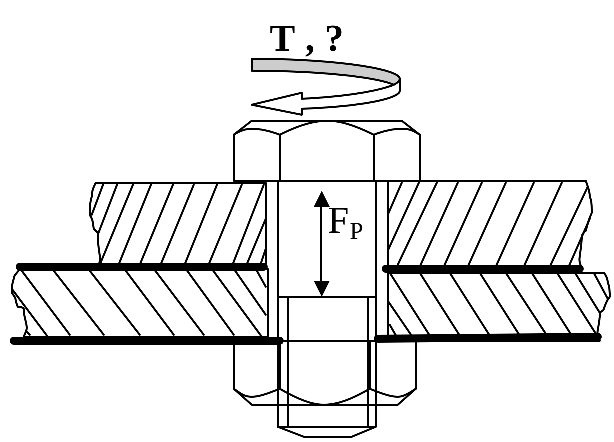
Compute ratio of the two as $\frac{\mathbf{?T}}{\mathbf{??}}$

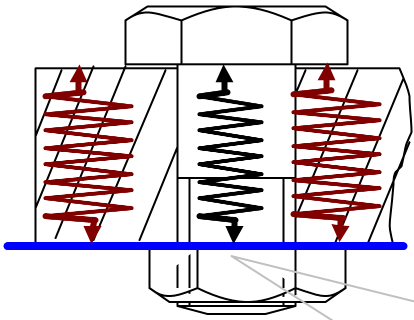


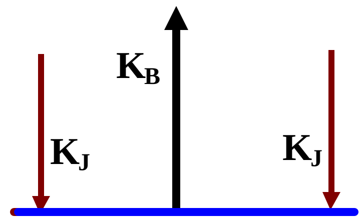
Notations

?F = Incremental preload in bolt
?T = Incremental torque of bolt
?? = Incremental turn of bolt (deg.)
?L = Incremental bolt stretch
K_B = Bolt stiffness
K_J = Joint stiffness
K_{Eqv} = Equivalent Bolt-Joint stiffness
K = Nut factor
D = Bolt diameter
P = Bolt thread pitch
E = Young's modulus of elasticity
A_S = Stress area of bolt
L_{eff} = Effective length of bolt
$\frac{\mathbf{?T}}{\mathbf{??}}$ = Torque to turn ratio
$\frac{\mathbf{?F}}{\mathbf{?X}}$ = Spring stiffness









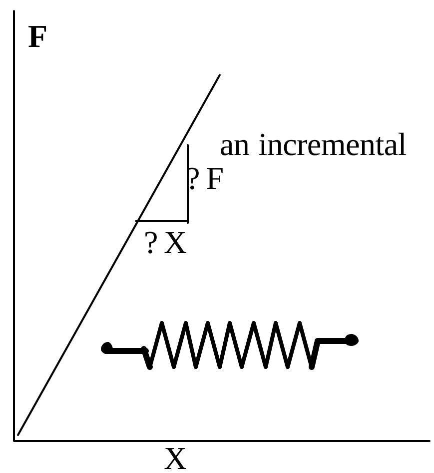
Equivalent stiffness of joint assembly.

$$K_{Eq} = \frac{1}{\frac{1}{K_B} + \frac{1}{K_J}}$$

For a loaded spring,

F = KX, or K = F/X

Similarly for spring of equivalent olt-joint that undergoes an incremental extension 9 ?L) under incremental load (?F) is



?F = K_{Eqv} x ?L or

Since

$$K_{Eqv} = \frac{1}{\frac{1}{K_B} + \frac{1}{K_J}}$$

Thus

$$?F = \frac{1}{\frac{1}{K_B} + \frac{1}{K_J}} \text{ x } ?L$$

Considering the bolt thread geometry, one complete turn (360 degrees) of the bolt head (nut held fixed), the stretch of bolt will be its pitch distance.

360 degrees rotation cause stretch = P
1 degree " " = P/360
?? degrees " " = (P/360) x ??

Substituting the values of ? L in the expression for ?F above, we get

$$?F = \frac{1}{\frac{1}{K_B} + \frac{1}{K_J}} \text{ x } (P/360) \text{ x } ?? \qquad \text{------------} \qquad (1)$$

Again from torque and turn relationship developed earlier,

$T = K\,D\,F$

Taking the incremental values,

$?T = K\,D\,?F$ or $?F = ?T / (K\,D)$

Substituting for ?F in equation (1) above an expression for joint stiffness can be expressed in terms of bolt stiffness and experimental torque-turn values as follows.

$$\frac{?T}{K\,D} = \frac{1}{\frac{1}{K_B} + \frac{1}{K_J}} \text{ x } (P/360) \text{ x } ??$$

or

$$\frac{?T}{??} = \frac{1}{\frac{1}{K_B} + \frac{1}{K_J}} \text{ x } (PKD/360)$$

or

$$\frac{?T}{??}\;\frac{1}{K_J} = \frac{PKD}{360} + \frac{?T}{??}\;\frac{1}{K_B}$$

or

$$K_J = \frac{\dfrac{?T}{??}\,K_B}{\dfrac{PKD}{360}\,K_B - \dfrac{?T}{??}}$$

(Ref.1, Bickford, page 310, expression 8.9 or 5.30 has a missing term)

The value of the bolt stiffness can be computed and substituted in the above equation and solve for the joint stiffness.

$\mathbf{K_B = A_S \times E / L_{eff}}$

Where

E = Young's modulus of elasticity
A_S = Stress area of bolt
L_{eff} = Effective length of bolt

Utility of above equation:
Find joint stiffness when torque-turn data and bolt stiffness values are available.

Example:
Given D = ½ in. P= 1/10 = 0.10, L_{Eff} = 3.00, K=0.20, E = 30 x 10^6 psi

From experiments - ?T = 250 in.lb. ?? = 8^o

FIND the joint stiffness.

Solution:

Stress area of bolt, A_S = 0.785 (0.50 - 0.974/10)^2 = 0.127 Sq. in.

K_B = 0.127 x 30 x 10^6 /3 (effective) = 1.27 x 10^6 lbs/in.

PKD/360 = 0.10 x 0.20 x 0.50/360 = 27.8 x 10^{-6}

?T/?? = 250/8 = 31.25 in.lb/degree

K_J = (31.25 x 1.27 x 10^6) / (27.8 x 10^{-6} 1.27 x 10^6 - 31.25)

= 39.69 x 10^6 / 4.056

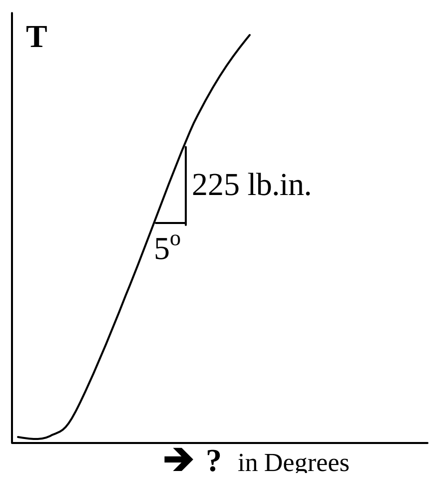
= 9.785 x 10^6 lbs/in

Exercise:

The effective length of an UN ¾ - 12 bolt is 2.75 in. Assume that the nut factor and Young's modulus are K=0.16 and E = 30 x 10^6 psi.

Experimental data for the joint torque and turn is as shown.

Determine the joint stiffness. (Ans. 4.97 x 10^6 lbs/in)



Solution:

A_S = 0.785 (0.75 - 0.974/12)^2 = 0.351 Sq. in.

K_B = 0.351 x 30 x 10^6 /2.75 = 3.83 x 10^6 lbs/in.

PKD/360 = (1/12) x 0.12 x 0.75/360 = 20.8 x 10^{-6}

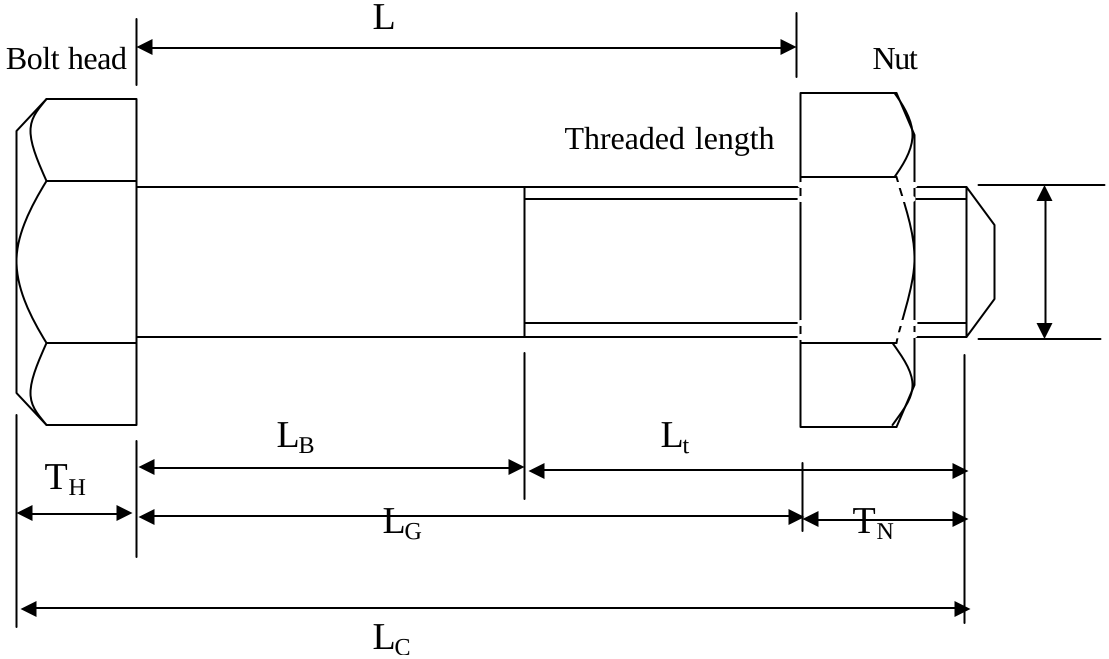
?T/?? = 225/5 = 45 in.lb/degree

K_J = (45x 3.83 x 10^6) / (20.8 x 10^{-6} 3.83 x 10^6 - 45)

= 172.35 x 10^6 / 34.66

= 4.97 x 10^6 lbs/in

Bolt Geometry



L_C = Total Length

L = Nominal bolt length ($L_B + L_t$)

L_B = Body length ($L - L_t$)

L_t = Threaded length

L_G = Length of body and threaded section

T_H = Length of head (Thickness)

T_N = Length/height of nut

D = Nominal diameter

L_{Be} = effective body length ($L_B + T_H/2$)

L_{te} = Effective threaded length ($L_G - L_B + T_H/2$)

Bolt Identification

Inch Series

½ - 13 NC 3 A

Male, B-Female Thread

Class of fit

NF = National Fine 1 – Very loose 2-free fit*
NC = National Course 3 – Close fit 4 – Snug fit
5 – Interference fit
*Most Ford applications

# of Threads per inch 1/13 = pitch

Nominal bolt diameter

Metric Series

M 10 x 1.5p – 6H

Position of thread (efgh)
Lowercase for external thread
Upper case for internal thread

1, 2, …9 amount of tolerance
1 – smallest, 9 – larger

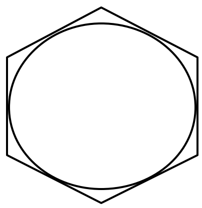
Pitch in m.m.

Outside diameter in mm

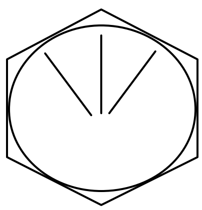
Metric

Bolt Grade Markings

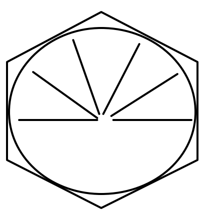
ASTM and SAE Grade Markings for Steel Bolts.
(Based on materials, hardening and strength)



No Marking Grade 1 and 2

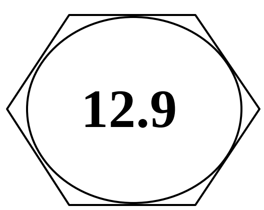


3 marks + 2 Grade 5

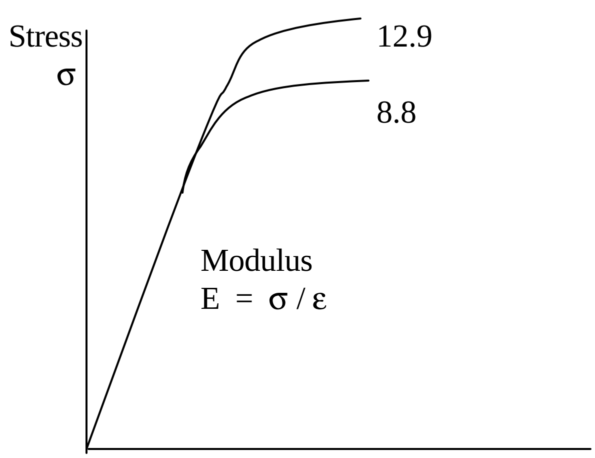


6 + 2 Grade 8

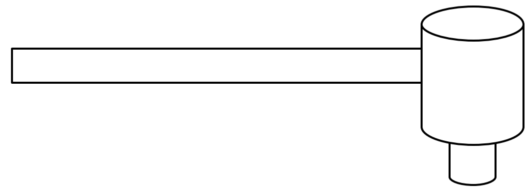
Metric Grade Marking



12 → 1200 Mpa Ultimate Strength
.9 x 1200 = 1,080 Mpa Tensile Strength



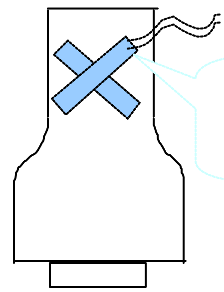
Torque, Angle, and Tension Measuring Devices



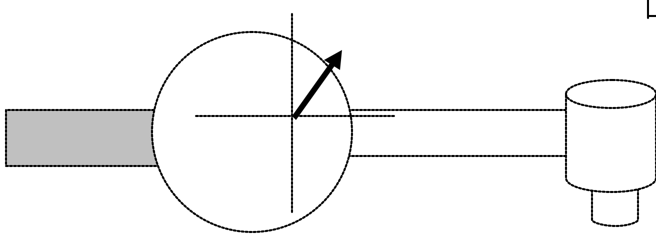
Torque Measuring Tools

1. **Reaction type**

- Electronic
- Mechanical (Dial or Clicker)



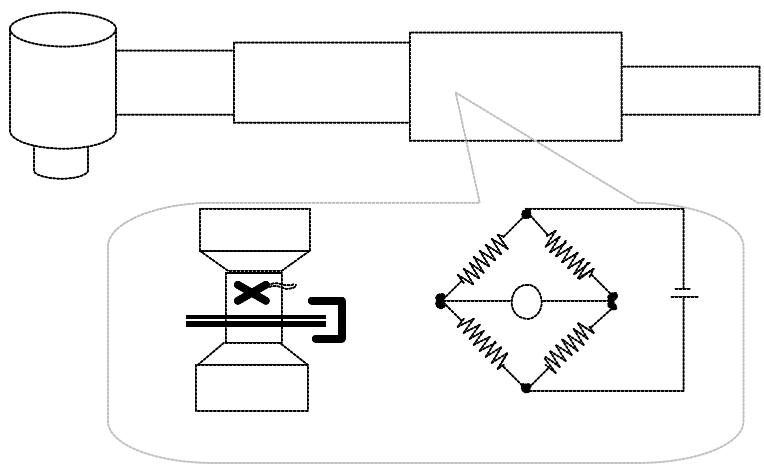
Typically used for torque audit in the plant. Individual pressure regulator for each tool desirable.

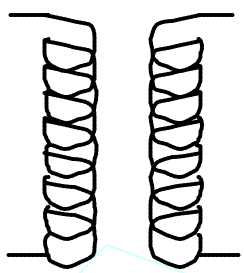


Inclined Plane

2. **Rotary type**

- Slip Ring
- Rotary Transformers

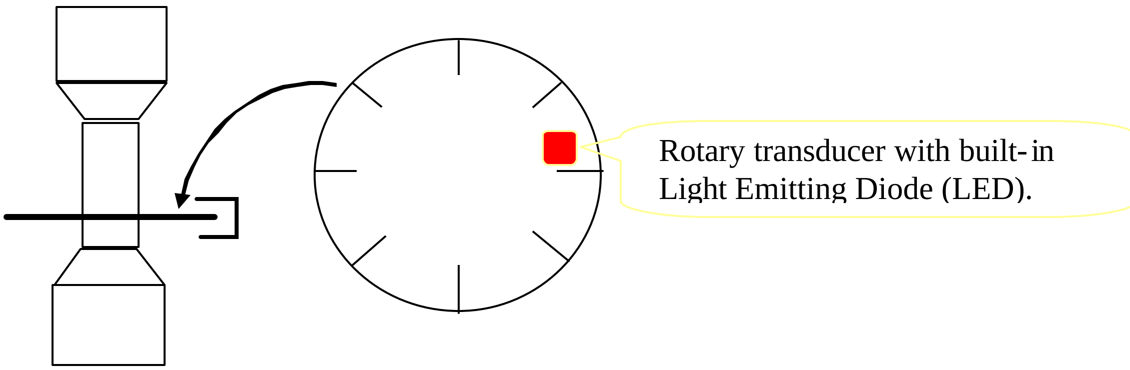




ROTARY TRANSFORMER

More delicate instrument. Used primarily for certification of power tool in the laboratories.

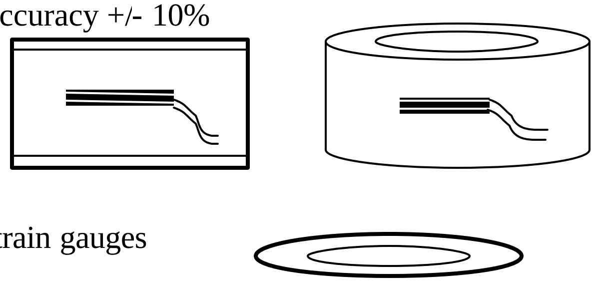
Angle Measuring Transducers



Tension Measuring Equipment

Load Washer
(for clamp load in bolt)

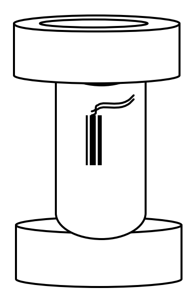
Accuracy +/- 10%



Strain gauges

Hardened steel washers

Load Cell



- Are not influenced by friction
- Need to machine the part to accommodate the Load Cell.

Bolt Converted to Load Cell
(STRAINCERT BOLT)

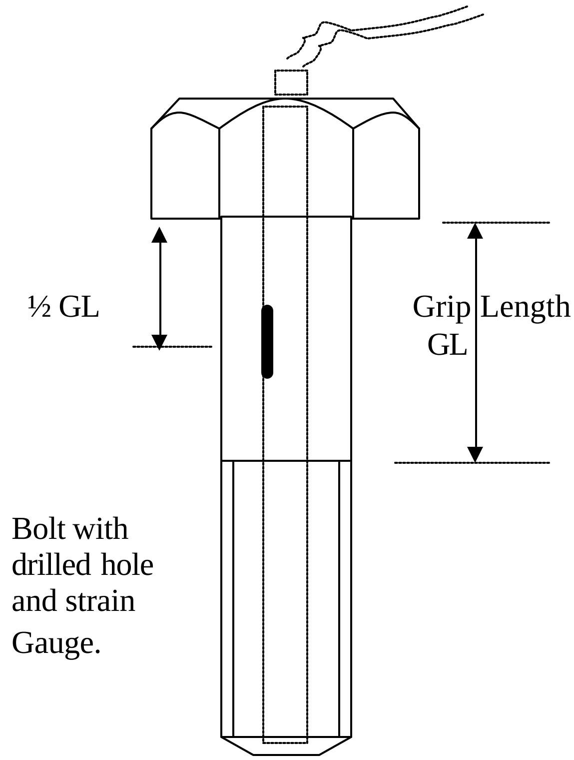
Accuracy within +/- 1%

It changes the stress characteristics of the bolt. The drilled hole is small enough such that

Crossection > X-section at core dia.

Used for measuring loads of Engine Cylinder bolts.

Straincert Bolt

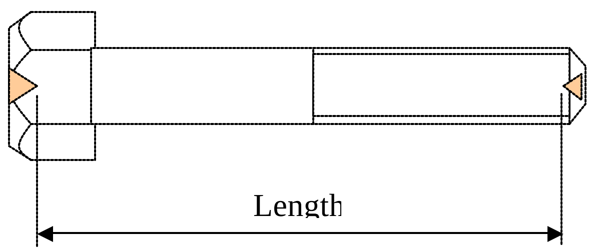


Length Change Method
(Physical Measurement)

L_f = final length (loaded)
L_o = free length (unloaded)

$\Delta L = L_f - L_o$

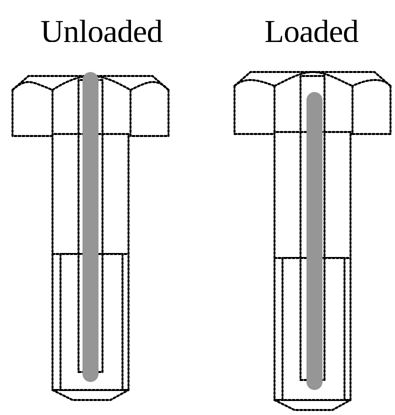
Load $P = \Delta L \ (EA/L)$



Drilled Hole Length

Measure length of drilled hole by a thin rod.

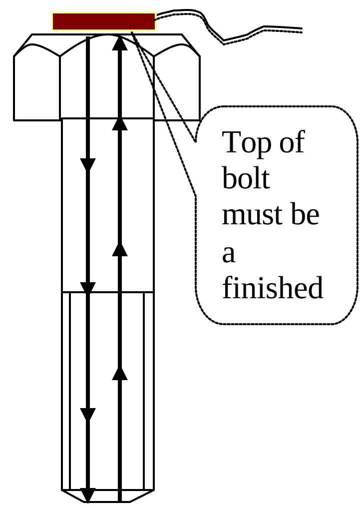
Expensive. Not much used in automotive.



Ultrasonic

A direct way to measure the clamp load.

Send ultrasonic sound to read the length of the bolt.

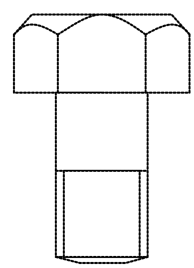


Accuracy = +/- 4 % (of clamp load)

$\Delta L = L_f - L_o$

Load $P = \Delta L \ (EA/L)$

Ultra sound measurement is preferred for short and stubby bolts as physical length measurement might be inaccurate due to smaller change in length.



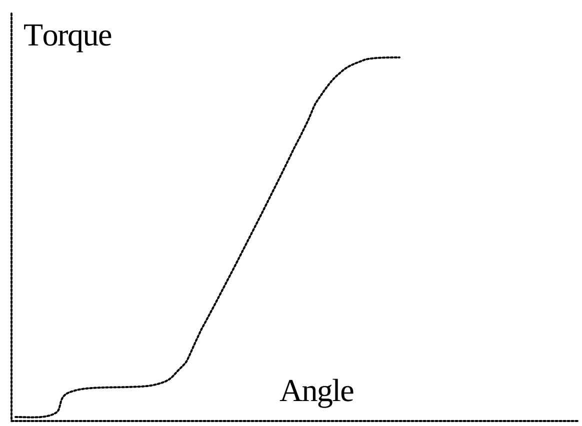
Torque Rate Measuring Devices

Computerized instruments measure both torque and angle of bolt rotation.

Older device weigh about 150 lbs.

Newer portable system weighs 30 lbs.

Data available in digital form for plotting And reduction purposes.

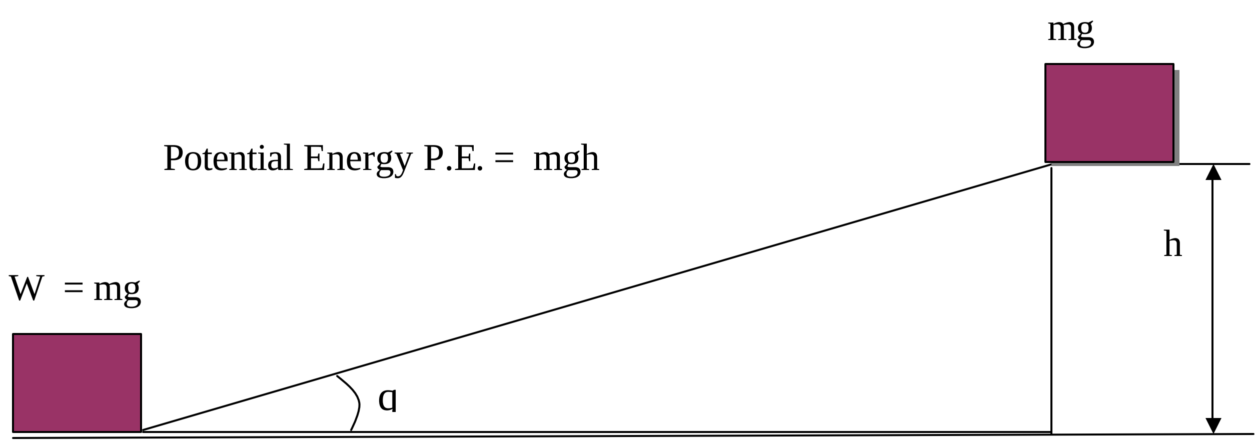


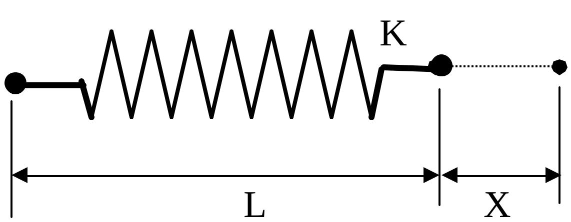
Torque Scatter Due To Tool

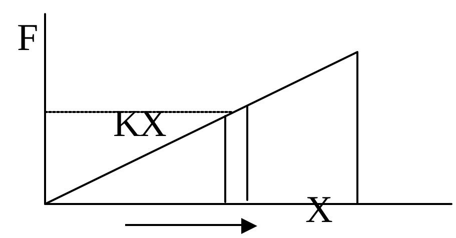
Conservation of Energy

During operations the tool stores and converts potential energy into kinetic energy, and back and forth. The linear and rotational inertias of the tool components cause variability in applied and measured characteristics.

Potential Energy (P.E.) = Work done

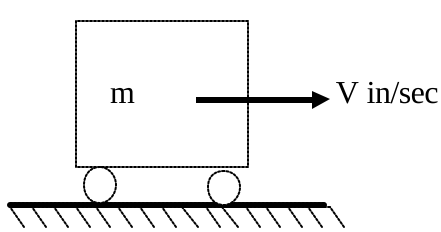


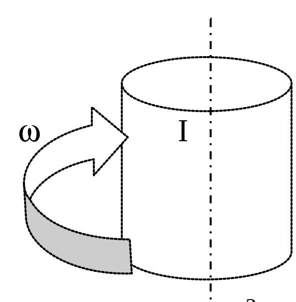




Kinetic Energy (K. E.)

Area under the Force Vs. Displacement P.E. = ½ K X^2





K.E. = ½ m V^2 K.E. = ½ I ω^2

M = Mass I = Mass moment of inertia ω = Angular velocity

Standardized Torque and Tension Values

(How Table values of Torque vs. Tension are calculated)

Torque T = KDF

Where K = nut factor (depends on lubrication and cleanliness)
D = bolt diameter
F = bolt tension

Torques necessary, for bolts of various sizes, to achieve a fixed stress (25,000 psi) limit have been calculated as follows:

$$T = KDF, \quad \text{but } F = (\pi/4\, D^2) \times 25{,}000$$

Or $$T = K.D.\ \pi/4\, D^2 \times 25{,}000$$

Or $$T = \pi/4 K.D.^3 \times 25{,}000$$

Assuming K = .2, $\boxed{T = 10^3 \times 3.93\, D^3}$ in lb.

Therefore the table value (T) for a ½" bolt
Should be $T_{1/2in} = 3.93 \times 10^3\ (1/2)^3 \cong 491$ in lb. [Table value: 433 in.lb for ½-32 UN]

If the bolt is of harder material with strength 60,000 psi instead of 25,000 (used in table values), and the nut is cleaned/lubricated such that K = .137 instead of .2, then the revised torque becomes (for ¼ - 20 bolt, T = 39.75).

$$T = \frac{K_n}{.2} \times \frac{s_n}{25{,}000} \times T_T = \frac{.137}{.2} \times \frac{60{,}000}{25{,}000} \times T_T$$

or $$T = 1.64 T_T = 1.64 \times 39.75 = 65.3 \text{ in lb}$$

Example:

For a ½-20 size bolt the table value of torque is 399.8. If the bolt is of hardened material (strength = 45,000 psi) and the nut factor is 0.15, determine the correct torque.

$$T = \frac{(0.15)}{.2} \times \frac{45{,}000}{25{,}000} \times 399.8 = 539.7 \text{ in lb}$$

Note: When the thread friction is reduced, less torque is needed to produce the same tension. But the need for higher stress required higher torque, which overcomes the reduced torque due to friction.

$T \propto$ friction (nut factor in the bolt)

$\propto$ Stress

Similarly, bolt tension (Preload) can be recalculated based on the table values for known torque and nut factor:

$$T = K\,D\,F$$

Or $E_N = T_N / (K_N D)$ (1) F_N = New Preload (calculated)

T_N = New Torque

From the table

$K = .2$ and $F = F_T$

Thus, from Eqn. (1) $F_T = T_T/.2D$ . . . (2)

$$\frac{F_N}{F_T} = \frac{T_N\,.2D}{K_N..D\;T_T}$$

or

$$\boxed{F_N = F_T \frac{.2}{K_N} . \frac{T_N}{T_T}}$$

Example:

The table values for 5/16-28 bolt is F_T = 1,515 lb. and T_T = 90.8 in lb. If the nut factor is K = .12, Calculate the tension for 120 in lb torque:

F_N = 1515 x (.2 / .12) x (120 / 90.8) = 3,337 lbs.

Bolt Strengths
(Standard Table values of Torque vs. Tension)

English 25,000 psi = 172.4 Mpa

Size	Series	Tensile Strength area (in^2)	Preload For 25 ksi (lbs) x 10^3	(in-lb) Torque to achieve 25 KSI stress
0-80	UNF	.00180	.045	.54
1-64 .	UNC	.00263	.065.8	.96
12-32	UNEF	.0270	.675	29.2
¼-20	UNC	.0318	.795	39.75
¼-28 .:	UNF	.0364	.910	45.5
½ - 13	UNC	.1419	3.548	354.8
½ - 32	UN	.173	4.325	433
½ - 16 :	UNF	.151	3.775	378
1 – 8	UNC	.606	15.150	3,030
1 – 12 :	UNF	.663	15.58	3,316

Metric Series

Size	Area (mm^2)	Preload 172.4 Mpa (K_N)	Torque for 172.4 Mpa
M1.6 x .35	1.27	0.219	.070
M2 x .4 :	2.07	0.357	.1428
M6 x 1 :	20.1	3.465	4.158
M12 x 1.75 :	84.3	14.533	34.9
M24 x 3	353.	60.96	292.6

Overview of Bolted Joint Design

Designing a bolted joint design is an iterative process, in that the designer generally relies on trial and error, past experience and personal judgment to make some design decisions. Obviously, experience and knowledge of the designer play an important role. With increases experience, the designer is able to make better decisions about the effect of certain design parameters.

However, regardless of the size, application or operating parameters of a joint, there are some steps which are commonly undertaken. These steps include:
Following are many of the critical factors that apply to all bolted joint designs regardless of the size, application or operating parameters.

1. Define purpose

Determine and define the following:

- desired function of joint
- environmental conditions
- cost targets
- size and operating parameters
- desired life
- critical nature
- potential failure modes
- any other factors involved influencing joint function.

2. Design joint.

Determine the layout of the joint, including joint members, size, shape and material(s).

3. Estimate service loads

Estimating service loads is a difficult, but very important step, especially in critical joints. The static and dynamic loads to be considered include weight, pressure, shock, inertial affects, thermal affects, etc. Load intensifiers, such as prying and eccentricity, should also be considered.

4. Select bolts

After the joint geometry and service loads are established, the bolt size, number and strength can be determined. As part of the bolt selection, you should identify

- *material*
- *diameter*
- *thread pitch*
- *length*
- *tensile strength*
- *head style*
- *drive style*
- *thread style*
- *hardness and*
- *Anticorrosion coating/plating.*

5. Determine required minimum and maximum preload & clamp load.
Preload at assembly should be selected such that the desired clamp necessary for proper function of the joint throughout the life can be achieved.

Preload > clamp load after relaxation

Minimum clamping force > force necessary to overcome vibration loosening, joint separation, slippage, fatigue, leakage and other similar type failures.
Maximum clamp force < force that can cause bolt yielding, joint crushing, stress cracking, fatigue failure, tensile failure or other similar failures in service.

6. Determine tightening methods and assembly torque
There are different fastener assembly methods and tightening strategies available. A proper method and strategy must be selected based on accuracy of achieving preload (P), nut factor (K), and required tightening torque. Commonly desired preload accuracy is:

Torque: ±35%
Torque-Angle: ±15%
Torque-to-Yield: ± 7%

$T = KDP$

Suppose that we wish to use a UNC 3/8 – 16 bolt with K factor of 0.20. The yield strength of the fastener is 5,000 lbs. minimum. If we use 80% as the design preload, that is 5,000 x 0.80 = 4,000 lbs.

The assembly torque then becomes: T = .20 x (3/8) x 4,000, or T = 300 lb.in

7. Release joint design.
Review and modify joint material, bolt preload range, bolt selection, tightening methods, etc. if necessary.

8. Experimental Joint Data
Joint design is not an exact science. Testing joint designs to ensure that the application achieves the desired level of reliability is very important. Experimental verification of the long term performance is essential for critical joints.

Bolted Joint Design Strategy

Goal → Reliable Joints [Comes from Sound design + quality part + Proper assembly]

Considerations for design vary depending on the design. But following checklist generally apply to all.

Checklist of reliable designs (bolted joints):

Mechanism – Bolted joint is a clamp. Unreliable joint is similar to an "inadequate clamp".

Reliability – Strong enough to give structural integrity, resist separation, prevent slip, overcome misalignment.

- enough number of bolts
- appropriate material strength
- acceptable resistance to fatigue
- acceptable relaxation and creep
- acceptable temperature relaxation

Bolt & Joint geometry – minimize stress concentration (bolt-head-to-shank fillets, thread run out, etc.)

In-Service clamping force – preload modified by service loads and thermal effects. Must be enough to minimize self-loosening.

Design Steps

1. Description of joints: Establish the purpose of the design. Identify the operating condition of the equipment in which the joint is located. Speed, vibration, impact, temperature, desired life, etc.

2. Preliminary design: Rough layout of sizes and shapes of various parts.

3. Estimate of Loading: Determine (estimate) service loads by considering – weight, pressure, inertia, thermal effect, etc.

4. Select Bolt – Nominal dia.
   - Static strength
   - Upper tension limit
   - Stripping strength of threaded area.
     (a must if threaded engagement is short)

5. Required Minimum Clamping Force

Clamping force > external load
(must always)

The amount by which clamping in-service force should exceed external load depends on:

- accuracy of bolt tightening
- correctness of the estimate of external load
- relaxation over time
- unknown overloading

Also:

Joint Slip – high tension needed to resist shear load.
Self Loosening – transverse load causes slip between joint members and threaded surfaces.
Pressure loads – Example: cylinder, boiler.
Joint Separation – Under gravity held part clamp load is needed only for alignment and can be very low.

Fatigue – in this case minimum clamp load calculation is quite complicated .

6. Upper Limit of Clamping Force

Loosening (slip leak) $\leq$ Clamping force (F_c) $\leq$ upper limit

Generally, we want the maximum clamping force the parts can stand.

Upper limit $\leq$ Bolt yield strength (current trend in automotive application is to just go past yield)

$\leq$ Thread stripping strength (worse case upper limit)

$\leq$ Allowable bolt stress and Assembly Stress limits by codes, standard, bias, company practices.

$\leq$ Torsional Stress Factor

$\leq$ Shear Stress Allowances

Other limits may be imposed by:

Flange rotation, Gasket crush, Stress cracking (for service load over 50% yield), combined load.

(Specifying TORQUE based on CLAMPING road required)

Joint Assembly and Behavior
[Assembly viewpoint, determining what torque to specify)

Now that we understand that CLAMP LOAD is what we are after, but the only practical way to obtain it is to apply the TORQUE, which is full of uncertainty. So, how do we determine what torque should we specify after the joint is designed (both and joint members)?

Theoretical approach:

1. Goal – Secure Correct Clamping Force

2. What clamping force is "right" for the joint?

   (a) acceptable range at application
   (b) expected range of tension obtainable after assembly
   (c) what happens after assembly
      - time, environment, application, temperature, etc.

3. Assembly Preload – Generally not known
   (a) Past experience – considered best guide
   (b) Ask the designer – second best
   (c) Unimportant joint and no prior knowledge
      - use the table value

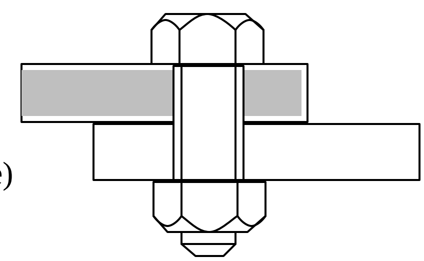
   (d) Concerned effort – calculate bolt torque analytically
      $T = K.D. (S_y.A)P$
      P = % of yield stress used for loading
      K = constant, D = bolt dia., Sy = yield strength

4. Anticipated Preload in service. . . etc.

Typical Target Preload (As % of Yield Strength)

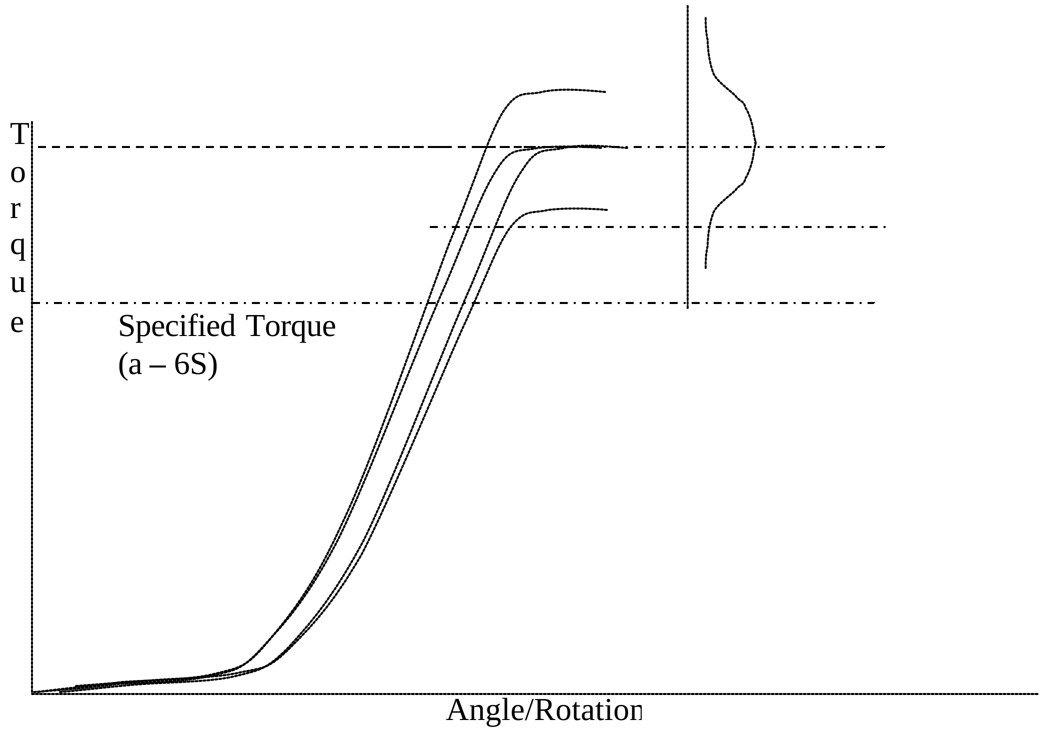
Preload as % of Yield Load	Common Situations
25% (if σ_y = 30,000 psi, then Bolt Stress = 7,500 psi)	Foundations, anchor bolts non-gasketed joints
40%	Gasketed joints in routine service
50% - 60%	Average non-gasketed joints "normal" safety concerns. Max. for gasketed joint.
70 – 75%	Upper limit for non-gasketed joints with previous "low preload" problems.
85 – 95%	Joints with previous "low preload" problem occurred consistently. Where it is unwise to reach load.
100%	Structural steel bolts tightened by turn-of-nut procedures. Joint facing fatigue, self-loosening, where service load can be predicted with accuracy.

Experimental approach:



Case 1: Threaded Bolt

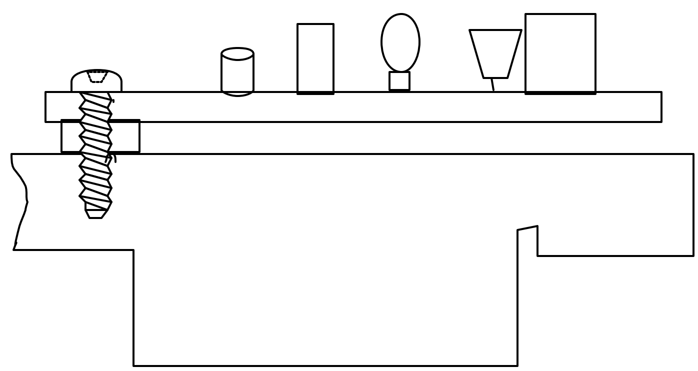
1. Tighten and measure yield torque for a set (10 or more) samples.
2. Calculate average(a) and standard deviation(s) of Yield torque.
3. Determine relaxation factor (torque multiplier to compensate for loss of torque).
4. Specify TORQUE at six standard deviation below the average yield torque (= a – 6s)



Case 2: Self-Tapping Screws

Assumption.

- Steel Screws
- Aluminum or Plastic joint members
- Failure occurs due to thread stripping

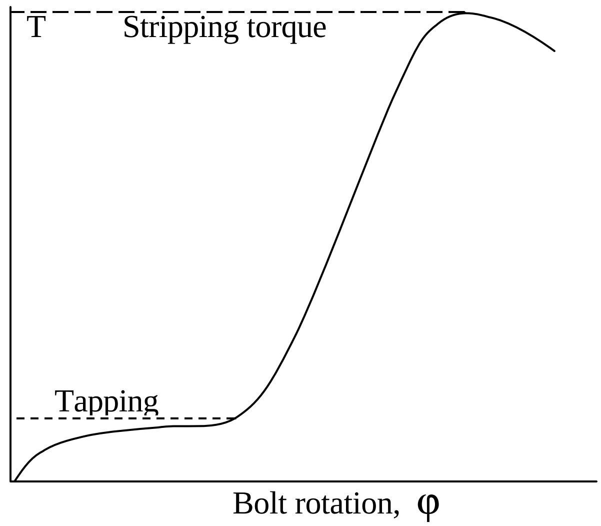


Steps:

1. Procure and prepare 10 or more samples for tests.

2. Take at least two torque measurements for each sample.

(a) Maximum tapping torque
(b) Thread stripping torque



Sample #	(a)	(b)
Sample 1	30 N.Cm.	120 N.Cm.
Sample 2		

Etc.

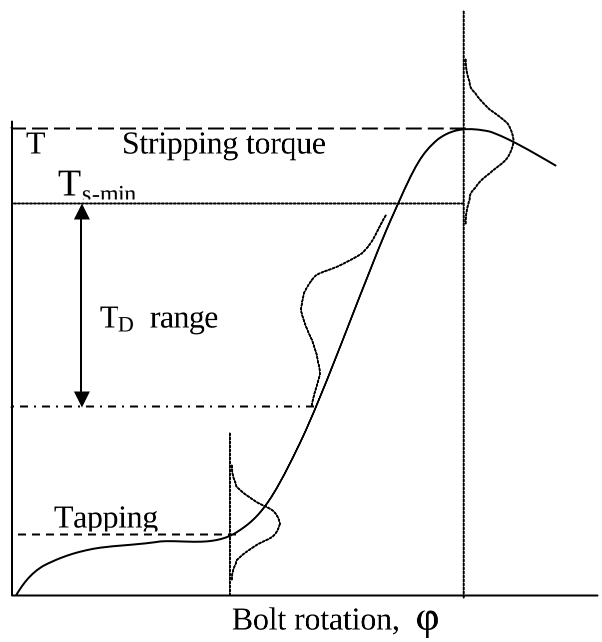
3. Calculate AVERAGE (μ) and Std. Deviation (σ) for each data set.

μ_t = Average tapping torque
σ_t = Std. Deviation of tapping torque

μ_s = Average stripping torque
σ_s = Std. Deviation of stripping torque
$T_{s\text{-}min}$ = minimum value of stripping torque

$= \mu_s - 3\sigma_s$

T_D = Design (specified) torque



4. Specify torque as 80% of minimum stripping torque.

T_D $= 0.80 \times T_{s\text{-}min}$

Alternate approach.
Assuming that the specified torque will have Standard Deviation same as the stripping torque (σ_s), the specified torque should be $3\sigma_s$ below the $T_{s\text{-}min}$.

That is

$T_D = T_{s\text{-}min} - 3\sigma_s$

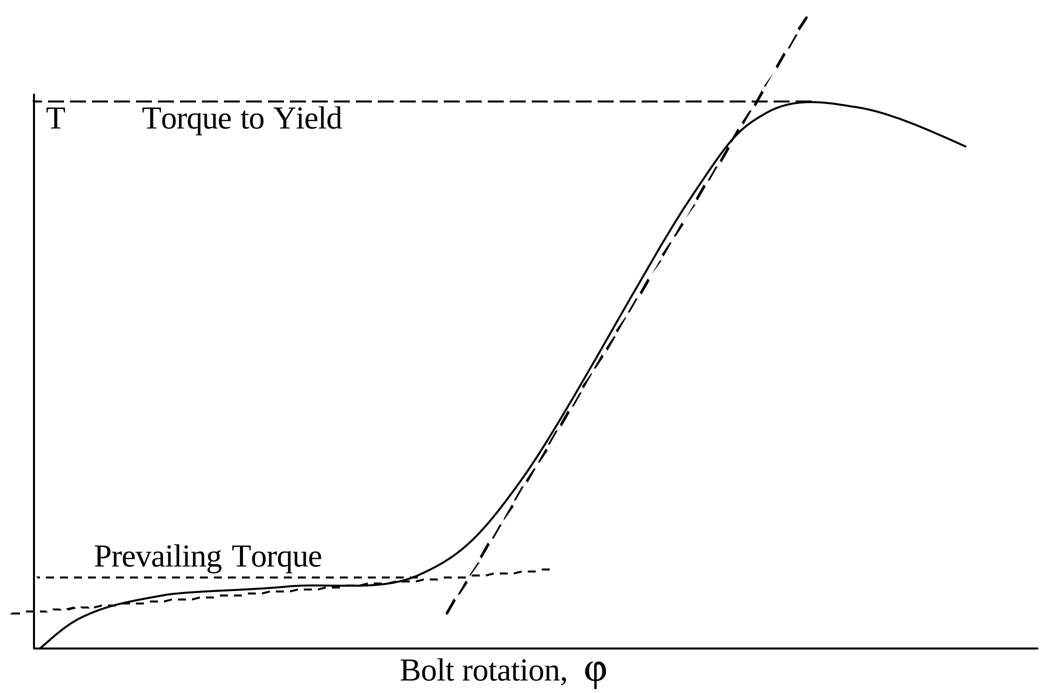
The above value represents the upper limit of the design torque. The lower limit should be above the upper range of the tapping torque.

Case 3: Bolted Joint (nut or threaded part. Conventional approach)

1. Procure and prepare 10 or more samples for tests (destructive)

2. Find maximum Prevailing Torque and Toque to Yield data for each sample.

Prevailing torque should be found by intersection of two straight line idealized to represent the initial torque build up portion and the elastic range of the torque Vs. angle graph as shown below. The Toque to yield is found by the tangent to the top part of the torque vs. angle plot.



3. Find the Average and Standard Deviation of the sample data (10 or more)

4. Specify torque at 80% of the minimum Torque to Yield.

5. Find relaxation factor by conducting additional tests. Measure Static Audit Torque for each of the above test samples at 2 min. 10 min, 1 hr. ½, 1 day, 5 days, and 1 month time interval. Plot data, Audit Torque vs. Time. Find the torque at stable (horizontal) portion of the curve. Find Relaxation factor, C_R as the ratio of the Audit toque at the beginning to that at the stable condition (do not use if no stable value is found). Modify the design torque by multiplying by the Relaxation factor.

Generalized Hooke's Law

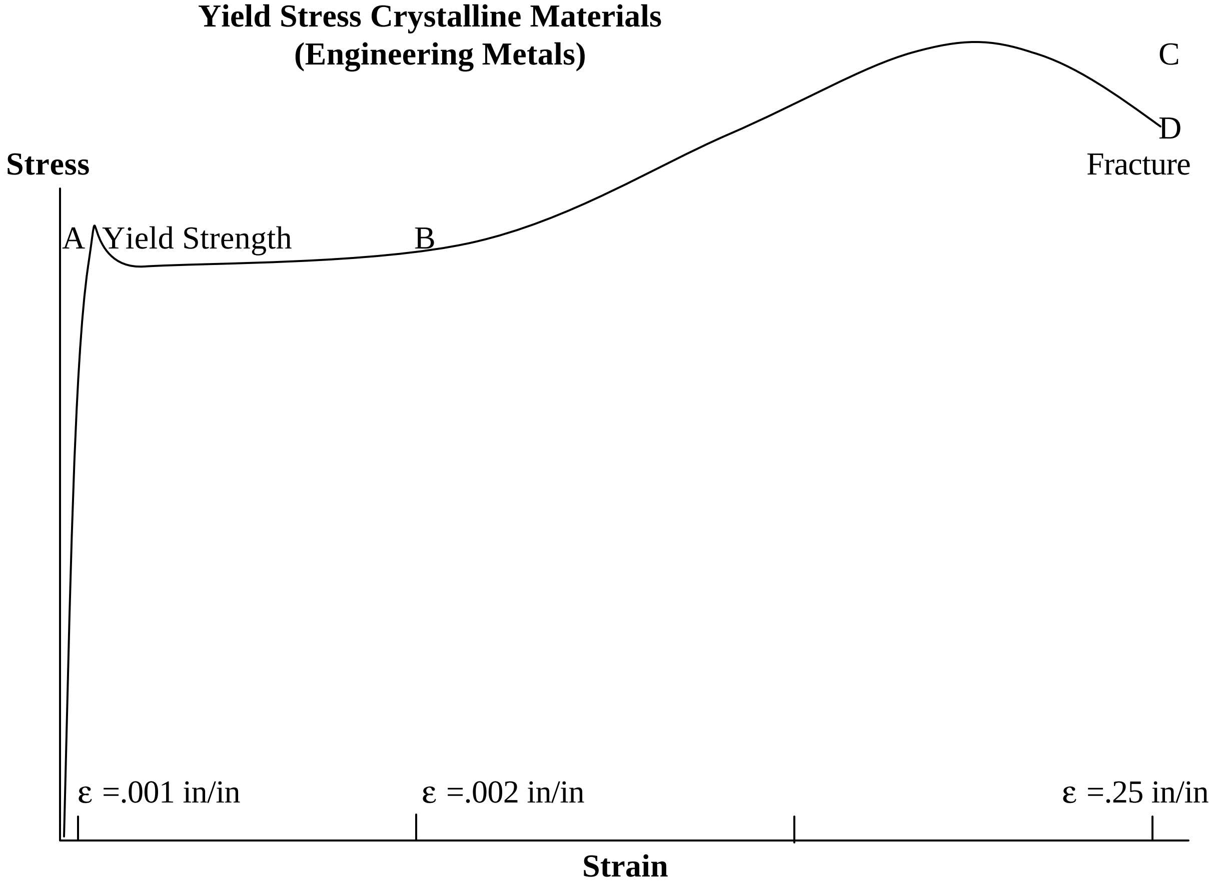
Linear Strain $\varepsilon = \sigma/E$ $\sigma \propto$ proportional limit
or $E = \sigma/\varepsilon$ and $\sigma = E\varepsilon$

Shear Strain

$\gamma_{XY} = J_{XY}/G$

or $G = J_{XY}/\gamma_{XY}$ or $J_{XY} = \gamma_{XY} G$

Yield Stress Crystalline Materials
(Engineering Metals)



Steel	1020 (hot rolled)	σ_u = 67,00 psi,	σ_y = 45,000,	BHN = 137
	1010 (hot rolled)	σ_u = 51,00 psi,	σ_y = 29,000,	BHN = 101

Mechanical Properties of Steel

Hot Rolled Steel (SAE 1020)	High Carbon Steel (SAE 1090)	Aluminum 1100.0 annealed
$E = 30 \times 10^6$ psi	$E = 30 \times 10^6$ psi	$E = 10 \times 10^6$ psi
$G = 12 \times 10^6$ psi	$G = 12 \times 10^6$ psi	$G = 3.8 \times 10^6$ psi
$\mu = .27$ (friction)	$\mu = .27$	$\mu = .33$
$\sigma y(.2\%) = 36{,}000$ psi	$\sigma y(.2\%) = 67{,}000$ psi	$\sigma y(.2\%) = 3{,}500$ psi
$\sigma_u = 65{,}000$	$\sigma_u = 122{,}000$ psi	$\sigma_u = 11{,}000$ psi
$w = .283$ lb/in^3	$w = .283$ lb/in^3	$w = .098$ lb/in^3
$\propto = 6.5 \times 10^{-6}$	$\propto = 6.5 \times 10^6/°$ F	$\propto = 13.1 \times 10^6/°$ F

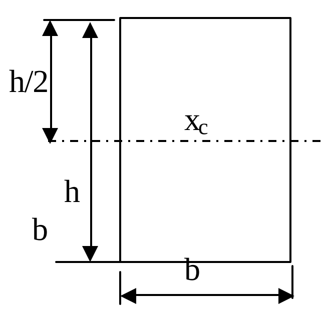
E = Young's Modules, G = Torsional modules, μ = Poison's ratio, ε_t/E_{axial}
$\propto$ = Thermal Expansion

Properties of Plane Areas

Area $A = bh$

$Ix_c = bh^3 / 12$

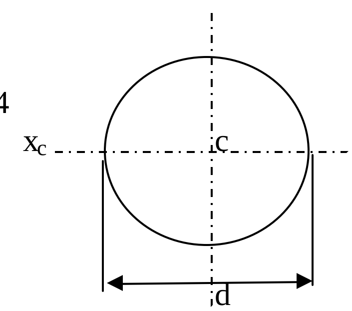
$Ix_b = bh^3 / 3$



$A = \Pi d^2 / 4$

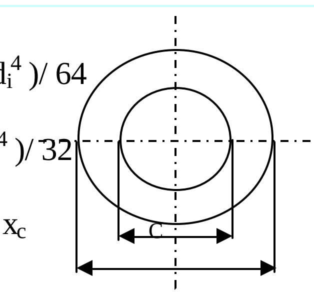
$Ix_c = \Pi d^4 / 64$

$J_c = \Pi d^4 / 32$



$Ix_c = \Pi (d_o^4 - d_i^4)/ 64$

$J_c = \Pi (d_o^4 - d_i^4)/ 32$

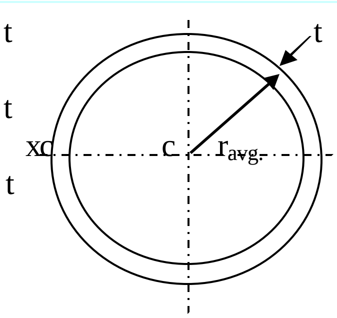


$A = 2\Pi r_{avg.} t$

$Ix_c = \Pi r^3_{avg.} t$

$J_c = 2\Pi r^3_{avg.} t$

$t << r_{avg}$



TORQUE CONTROL TOOLS

The tools for application of torque are selected based on

- Productivity
- Cost
- Quality Reliability

Manual Torque Wrenches

Speed – Slow
Accuracy - + / - (2 to 20%)
Incorporates Gauge, Dial, Click, and Electronic

Torque Multipliers

Accuracy – inadequate

Hydraulic Wrenches

Can produce high torque
Usually slower

Impact Wrenches

Speed – very fast
Accuracy - +/- 50%
High maintenance
Lower operator reaction force

Air Nutrunners

Speed – slower (certified) faster (non-certified)
Accuracy- 5 – 18% (certified), up to 40% (non-certified)
Stall – dependent on operator influence
Clutch – controlled by operator reaction (Air shut off)

Electric Nutrunners

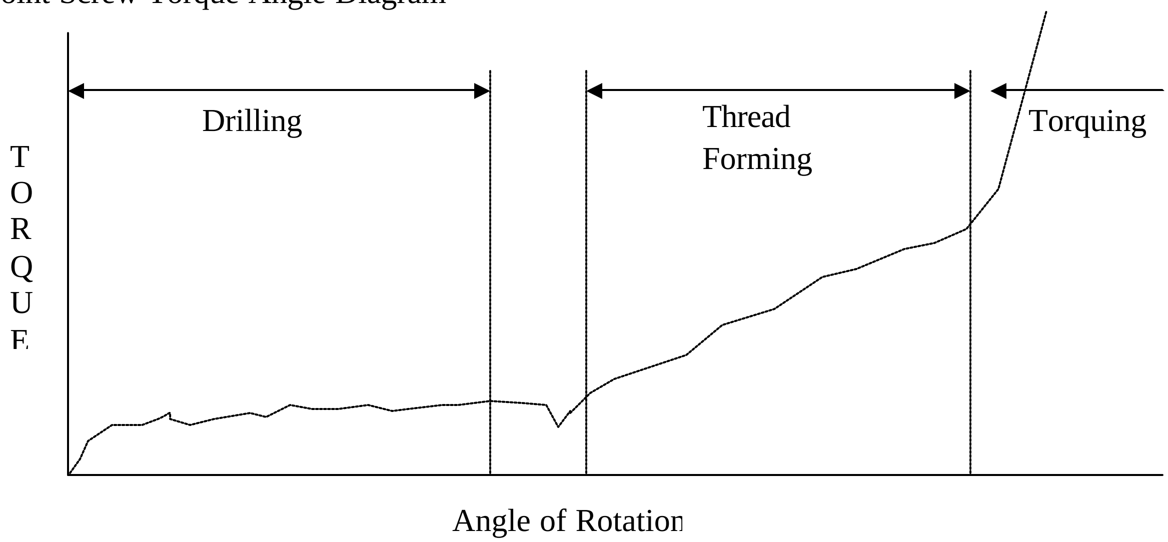
Speed – usually slower
Cost – uses lower energy
Accuracy – more accurate with feedback control

Impulse Torque Wrenches

Speed – quicker

Factor Influencing Power Tools

Drill Point Screw Torque Angle Diagram



Sources of Variabilities

Drilling
- Dril point – shape and hardness
- Force applied
- Torque
- Metal thickness & hardness

Thread Forming
- Torque – depends on hole size, screw O.D.
    - Metal thickness

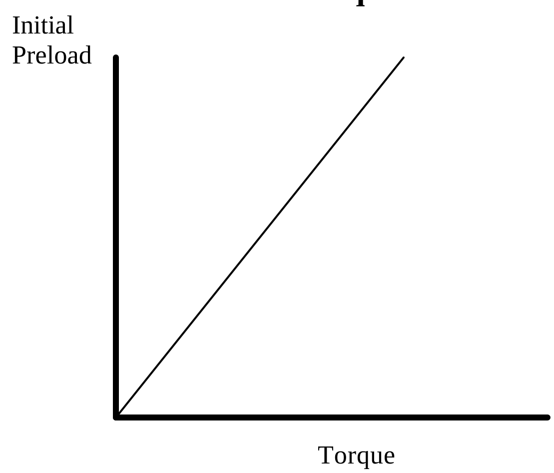
Torquing
- Friction – Screww finish, materials (steel, aluminum, plastic)
- Geometry – anle of driving

Summary: Joint Assembly

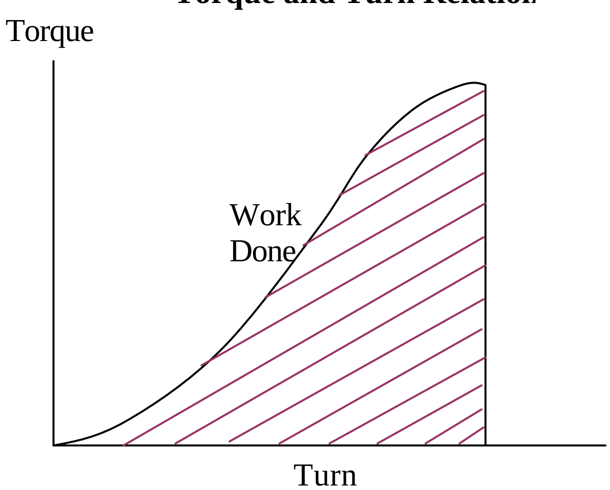
Joint Assembly Considerations

- The main idea in a joint is to obtain proper clamping load. To secure proper clamping load, the joint must be well assembled.
- Clamping for is called the preload
    - *Initial Preload* – load while tightening
    - *Residual Preload* – what's left after assembly
    - *In*-service Bolt tension – what's left after some time in service

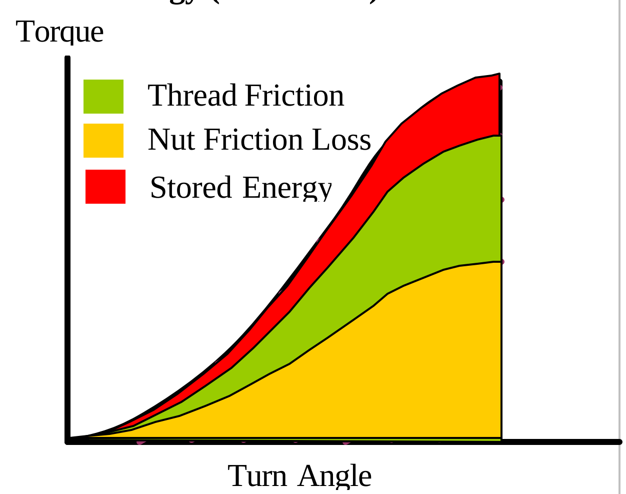
Load and Torque Relation



Torque and Turn Relation



Loss of Energy (about 90%)



Reasons for Loss of Clamp Force

Short Term relaxation due to:

- Contact *embedment*
- High spots yielding between surfaces
- Softer parts
- Poor tread engagement
- Large fillets
- Oversized holes
- Misaligned holes

- Can be avoided by longer bolt.
- 2 – 11% without and 10 – 30% with gasket.
- 100% with thick gasket and short bolts.

Elastic Interactions (Cross-talk between bolts)

It is a phenomenon in which tightening one bolt affects the tension of another. It is an unavoidable situation present in some joints with multiple bolts.

- Is affected by joint gasket stiffness.
- Can be compensated by bolting pattern.
- The net result is that it loosens bolt that were previously tightened.
- It is not a concern in many joints.

Topics	Group Exercises
Joint Design Theoretical Approach	**B-3** **a:** Find the joint design parameters based on the given conditions. The bolt is to withstand a maximum load of 12,000 lbs. Yield stress in bolt = $45,000 Nut Factor, K = 0.20 Bolt dia. = ? Clamping Load= ? (Use 50% of yield stress) Torque = K D (Stress x Area)).50
Experimental Approach Assembly Torque Specification	**b:** Specify the assembly toque for an electric torque wrench for the tapping screw assembly with the following data. Use toque limit at 80% of the minimum stripping torque. Avg. stripping torque = 60 in.lb. Stand. Dev.. = 3 in.lb. (Ans: 40.8 in.lb.)
Flange Joints	**c:** Design a joint using class 150 threaded flanges to connect two 2 in. diameter pipes carrying steam at 300 psi. Specify the torque required for the bolts to keep the leakage within limit. (Assume gasket factor, M = 1 and nut factor, K = 0.32, 4 5/8 dia. bolts) Hint: See example at page C-9, use Table at page C-7 for 2 in dia. bolts.)

Section – C

Gasketed Joints for Leak Prevention

Most of today's liquid or gas used by industry is carried by tubes. Many such tubes are connected to each other by flanged ends and joined by bolts. Preventing leaks in jointed tubes is the most important and difficult job.

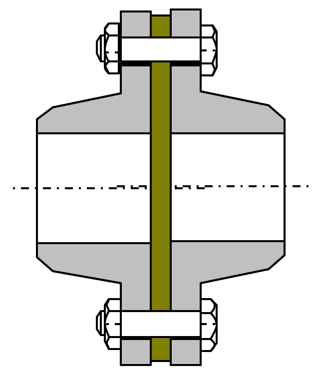
Example application of gasketed joints is:

- Pressure vessel
- Pipe lines
- Process systems
- Etc

Gasketed Joint:

Gasketed joint includes a third member in addition to bolt and joint members.

1. Bolt
2. Flange
3. Gasket



Purpose of gasketed joint is to prevent leak. A gasket between the joint members prevents leaks. Theoretically leaks can never be eliminated, but minimized. With proper bolt tensioning the joint can be designed such that the leak is so slow that it is extremely difficult to detect.

Causes of Leak

- Loss of contact pressure
- Distortion of flange due to non-uniform tightening
- Rough or damaged flange surface
- Small leak at initiation often erode or corrode leak path that increase leak rate over time.
- Creep and relaxation of gasket materials and effect of temperature change

MECHANICAL BEHAVIOR OF A GASKET

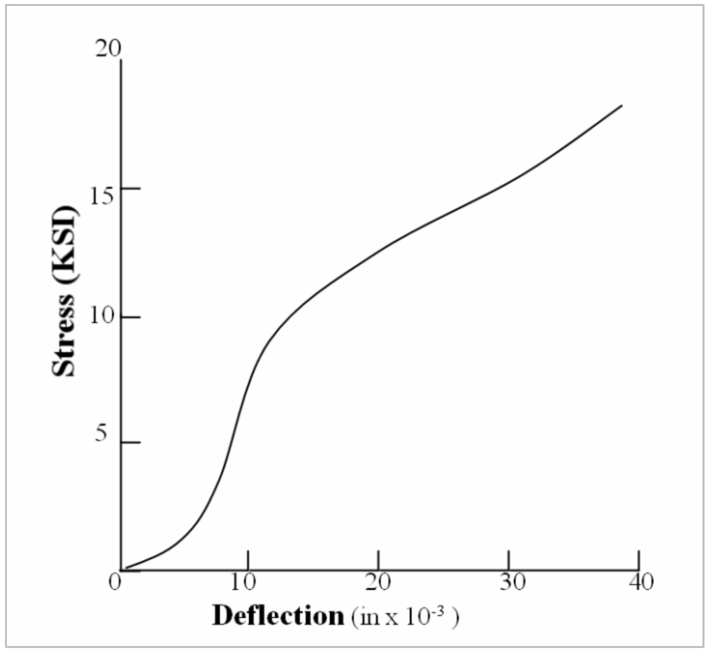
Understanding mechanical behavior of gasket joint is essential to determining causes of leak and remedies to prevent it.

Gasket Stiffness – The load-deflection characteristic of a gasket is reported in terms of compressive stress and deflection. Assuming that the gasket in use is subjected to uniform stress, or load-deflection, the stiffness can be calculated from the measured values. Gasket parameters are usually determined by compressing a small gasket under hypothetically loaded platens.

Key Observations

1. Gasket force-deflection curve is non-linear. This changes the stiffness value depending on the applied force (prediction becomes difficult)

2. Gasket inserts a third spring in series with the joint.



K_J = Joint stiffness, K_F = Gasket stiffness and K_G = Gasket stiffness, then

$$\frac{1}{K_J} = \frac{1}{K_G} + \frac{1}{K_F}$$

(This represents two springs in series)

3. The spring constant of the gasket is much smaller than that of joint and/or bolt. It being of lower value dominates the elastic behavior of the joint assembly. (Two springs in series results in equivalent stiffness to be lower than the lower of the two.)

Examples:

(i) $K_F = 4$ and $K_G = 1$ then $K_J = 1 / [1/1 + 1/4] = 4/5$ (lower than 1)

(ii) $K_F = 12$ and $K_G = 3$ then $K_J = 1 / [1/3 + 1/12] = 12/5$ (lower than 31)

4. Gasket is not fully elastic. When deformed, it takes some permanent set. It also exhibits different behavior under dynamic loading (called hysteresis).

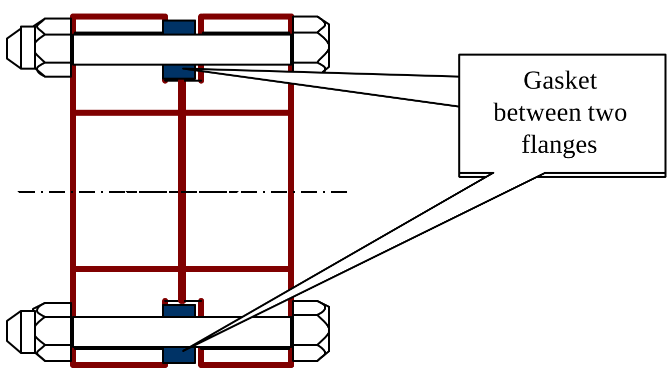
5. The thermal expansion coefficient of gasket is normally different from that of flange or bolt materials. Thus, the temperature changes causes thermal load (potential for leak)

6. Some joints such as raised-face flange, the joint stiffness is calculated with rotational stiffness not readily available for gasket. (includes error)

EFFECT OF CREEP AND RELAXATION ON GASKET BEHAVIOR

Pure Creep – is the loss of thickness under a constant compressive stress load. Creep is reported as thickness change in percentage at preloaded condition (constant load).

Pure Relaxation – is the reduction in compressive stress when the gasket is subjected to a constant deflection. Relaxation is normally expressed as percentage reduction of the compressive stress.



Creep Relaxation – is a combined effect of creep and relaxation on gasket.

In a gasketed (flanged joint), the gasket is compressed by tightening the bolt. As the gasket creeps:

- it becomes thinner

- bolt relaxes which cause
    - Loss of elongation
    - Loss of tension
    - Reduction in clamping force

- Compressive stress on gasket reduces

Example of Creep Relaxation:

In certain joint, the preload in bolt created elongation $10x10^3$ in. and gasket compression of some (X) amount. After a time, T hours, the elongation of bolt is found to be $8.5x10^3$ in., which represents a loss of $1.5x10^3$ in. or 15%. Find the loss of bolt tension.

This loss of elongation ($1.5x10^3$ in.) has also occurred in the gasket.

From the loss of bolt elongation, gasket creep can be calculated as

Gasket creep = $1.5x10^3$ in / (X)

This is much less than 15% as X is much larger than the original bolt elongation.

Loss of bolt tension = 15

Since P = (?L/L) A. E and (?L/L) = 15%

(A and E are crossectional area and elasticity modulus respectively)

Gasket Creep Relaxation Data (Page 658, Bickford)

Gasket Type	Thickness	Temperature	Type of Test	Time (Hr)	Loss (%)
Compressive Asbestos	N/A	75 Deg. F	C-R	24 1000	21% 28%
Cork	0.062	200 Deg. F	C-R	18	68%
Flexible Graphite	N/A	75 Deg. F	C-R	24 1000	11 12

C-R = *Creep Relaxation*, Loss *(%) is of clamping force or one of its equivalents (bolt tension, bolt stretch, or gasket thickness).*

Factors that Affect Creep

Cyclic Loading – Gasket s are found to creep more under cyclic loading than static loads.

Material Plasticity – Plasticity in gasket material is desirable as it allows better mating with the flange surface and prevent leak. This means that some creep is desirable and unavoidable.

Initial Thickness – Creep relaxation is directly proportional to the thickness. A 1/4 in thick flexible graphite gasket will relax about twice as much as a 1/8 in. thick gasket made of the same material. For design selection,

- a gasket should be as thick as it needs to be
- but as thin as possible.

Note that ASTM F 38 B test reports creep data on 1/32 in thick gasket and must be modified for thicker gaskets.

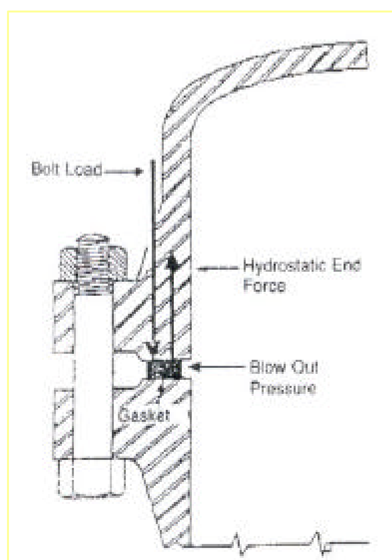
Initial Load – The amount of initial stress and preload on the gasket affects creep rate. Generally, higher load produces higher amount of creep.

Time and Temperature – Most creep relation occurs in the first 15 – 20 minutes of the load application creep increases at higher temperature. Creep rates become very slow after 18 – 24 hours.

Bolt Stiffness - The stiffness of the bolt does not have influence on gaskets creep-relaxation properties, however, it affects the relationship between loss of gasket thickness and the corresponding loss of clamping force. Long, flexible bolt will result in less loss of clamping force than a short, stubby bolt.

GASKET STRENTH - The P x T Factor

Gasket must have proper strength (tensile and crushing) to resist being torn apart blown out of the joint. Blowout resistance of a given gasket material is measured by clamping a test gasket between a pair of flanges and then subjecting it to higher pressure and temperature. Typically gasket and flange assembly is heated to certain level of temperature then the contained pressure is increased until blowout occurs. Every material is found to withstand a certain level of service temperature and pressure. After several such tests, a (P x T) rating is assigned to the gasket.



A gasket material with temperature limit of 300 Deg. F and a contained pressure limit of 1000 psi may not operate when both are at their limits. If the P x T value for this material is 250,000, then at a proposed service pressure of 1000 psi, it can withstand 250 Deg. F. If the desired service pressure is 1200 psi, then the service temperature could be as high as 250,000/1,200 = 208 Deg. F. P x T factor data for a few gasket materials is shown in the table below.

Table 19.2 Mechanical Properties of Sheet Gaskets

Type of gasket	Reported range of properties						
	Creep-relax[a] (%)	Tens. str.[b] (psi)	Compress. (%)	Recovery (%)	$P \times T$ × 1000	Max P (psi)	Max T (°F)
Compr. asbestos	16–30	2600–5000	7–17	40–50	350	1200–3700	650–1100
Compr. aramid fiber	18–25	2200–2800	7–17	40–50	350	900–1200	700
Compr. carbon fiber	14–15	1500–1800	7–17	50	700	2000	900
Pure PTFE	10–55	1625–2000	6–55	20–40	50–350	100	500
PTFE/st. st. core	20	5000	4–9	45–50	NA	2500	500
Rubber and synthetic rubbers	NA	1000–2600	25–75	NA	15–20	100–150	250–350
Flexible graphite	<5–5	900	40	20	700	2000	932–5432[c]
Inorganic-filled PTFE	30	2000	8–16	40	NA	1200	500
Vegetable fiber sheet	NA	1000–2000	25–55	40	40	200	212–250

[a] At room temperature.
[b] Across the grain (i.e., weak direction).
[c] Highest in a neutral or reducing atmosphere; lowest in an oxidizing atmosphere.
Source: All data taken from manufacturers' catalogs: Refs. 40, 47, 48, 49, and 59.

Ref.1, Page 661, Table 19.2

[For more details on gasket behavior and thermal effects, read Ref. 1, Pages 670 -673.]

LEAKAGE BEHAVIOR OF GASKET – m and y Factors

For design of flanged connections Section VIII of ASME Boiler and Pressure Vessel Code provides two factors, m and y. These are known as classical view of leakage behavior and are considered very simplistic.

y or seating stress factor – is the initial gasket stress or the surface pressure required to preload or seat the gasket such that there is no leaks in the joint when pressurized.

and

m or maintenance factor – is the ratio of contact pressure to the contained pressure. Experiment has shown that the liquid or gaseous pressure contained by a gasket is proportional to the amount of residual pressure, which is the pressure on gasket after the system is pressurized, is proportional to the residual contact pressure exerted by the joint surfaces on the gasket. (m factor may be different for different types of gasket.)

Gasket Factors from ASME Boiler & Pressure Vessel Code

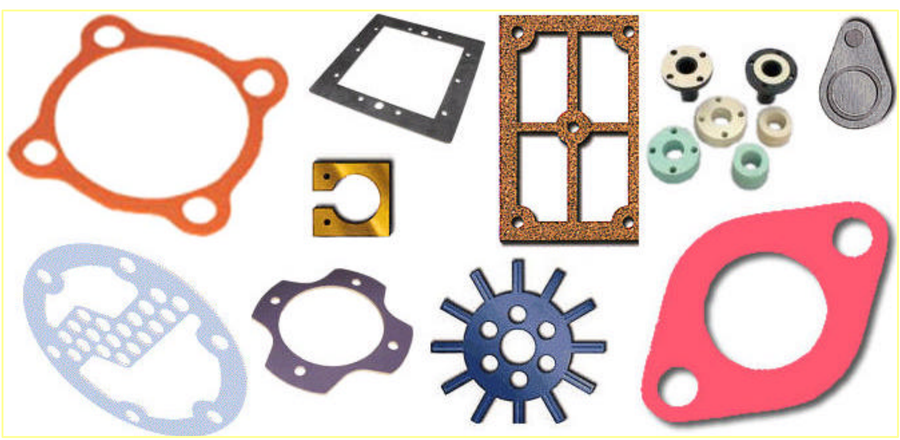
Type of Gasket	Maintenance Factor, m	Min. Seating Stress, y psi
1/8 in. Asbestos	2.00	1,600
Soft Aluminum – flat	3.25	8,800
Elastomers with cotton fabric	1.25	400

[*Advance studies in Gasket behavior including Tightness Parameter and new Gasket Factors are beyond the scope of this course. Interested participants are encouraged to read Rf. 1, pages 676 – 699 and consider attending special course of Flanged Joints offered by the ASME*]

GASKET SELECTION

Selecting proper gasket can be a complex task. For suitable gasket, one must have knowledge about the materials, configuration, strength, and other mechanical and physical properties. General guidelines about selection should be based on the following gasket properties:

1. Chemical resistance
2. Heat resistance
3. Tensile & Crush strength
4. Seating stress
5. Cost
6. Resilience (stiffness/recovery)
7. Pressure of contained liquid or gas
8. Nature of contained fluid 9explosive, carcinogenic, benign, etc.)
9. Configuration of joint
10. Permeability of gasket.
11. Lubricity
12. Creep characteristic
13. Thermal conductivity
14. Shape and size of gasket



Presently, there is a number of software available for selecting suitable gasket. These software incorporates gasket date in regards to seating strength or stress, crushing strength and similar hard to find information. Users input data about his/her applications: flange size, pressure ratings, gasket contact area, bolt length, service temperature and pressure, the type of fluid to be contained, type of bolt, etc.

Such software program out put includes:

- Recommendation for seating stress (including assembly torque)
- Total clamping force the torque will generate on the flange
- Working stress on the gasket
- Maximum allowable gasket stress
- Gasket crushing strength
- Etc.

SIMPLISTIC DESIGN ASSMEBLY GUIDELINES

What toque should one use for assembly?

- Determine the bolt's yield strength
- Find the crushing strength of the gasket (in terms of bolt stress)
- Use smaller of the above two to compute assembly torque.

(This method ignores thermal expansion, flange rotation, stress corrosion, etc.)

Other known practice include:

- Use of the y factor for the gasket, which gives minimum seating stress, obtained from ASME Code or from gasket manufacturer to compute torque
- Calculate toque from a table of "allowable stress" from the Code

Safety Check – Before proceeding with the assembly, divide the desired force by the "tensile stress area" of the bolt to assure that the stress is below the yield strength of the bolt at service temperature.

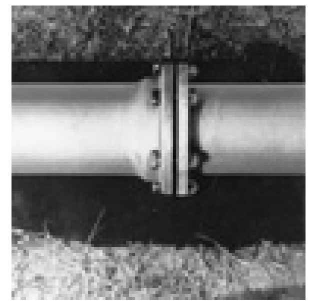
Calculate torque using:

$$T = K\,D\,F \text{ in. lb.}$$

Where D = nominal diameter of bolt (in)
F = target preload (lb)
K = Nut factor (dimensionless, steel on steel = 0.2)

Topics	Group Exercises
Gasketed Joint	**C-1** **a:** A gasket materials has a P x T factors as 280,000. Determine: (i) Maximum acceptable service pressure when the service temperature is 280 Deg. F (ii) Allowable service temperature if the service pressure needs to be 1,200 psi.
Gasketed Joints	**b:** Determine the assembly torque for a gasketed joint of the following description. 4 steel bolts, ½ in. dia, Yield stress = 35,000 psi. Gasket area = 3.5 Sq. in., Crushing strength of gasket = 9,200 psi. (Use T = K D F, with K = 0.30 for stainless steel bolts.)

GASKETED FLANGED JOINT DESIGN



The purpose of the flanged joint is mainly to join two section of the pipe, seal ends of tubes, or cover containers of various shapes. Although most flanged joints are circular, as shown for piping joints, rectangular and shapes of complex nature for valve, turbine, engine cylinder heads of internal combustion engines, are common. The fluid contained in such joined devices may be gas or liquid, which may be stationary or flow at high speed, and at wide ranging pressure and temperature (External or internal pressure range: vacuum to very high, temperature rage: cryogenic to 1000^{o}C). The common objective in all these applications is to control the leakage.

Flanged joints can be of two types:
- with gasket
- without gasket (metal to metal contact for sealing)

Objectives and Design Challenge

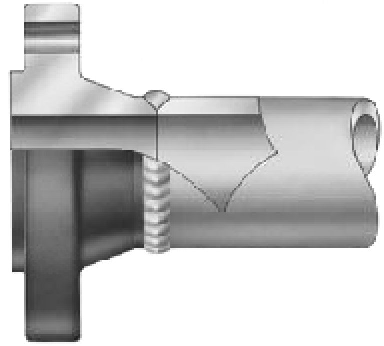
For a given characteristics of the gasket, determine economical dimensions of the flanges and the bolts that assures a desired level of sealing. The sealing level desired is obtained when leakage is controlled within acceptable limit. Specific design goals are:
- Establish dimension of the flanges
- Determine number of bolts & its tension (Prescribe torque necessary based on the torque vs. tension relationship.)

Types of Flanged Joints

For most applications choosing the standard joint design is the most efficient approach. Standard flanges and gasket of corresponding sizes are readily available in the market place. The available standard joints are specified in detail in national standard like ASME B16.5, and international standard like ISO 7005.

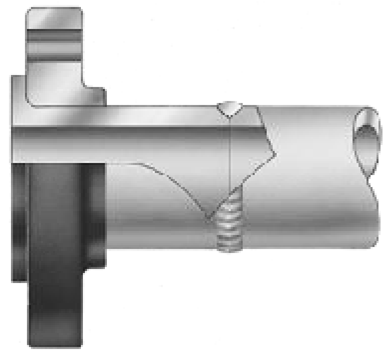
There are several designs of flanges available to suit applications of all kinds. The more common among the flange designs are shown below.

WELDING NECK FLANGE



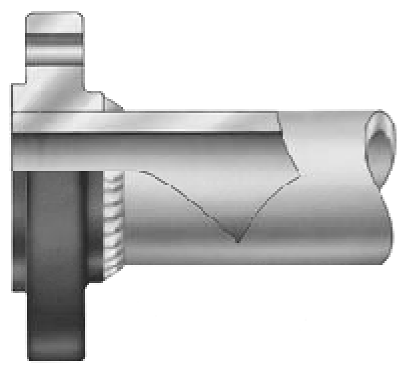
- Common choice for heavy duty applications.
- Designed with a hub on the backside tapering to a diameter that will match the pipe.
- Flanges are bored to match the inside diameter of the mating pipe so there will be no restriction of product flow.
- Reinforced hub gives it an extra rotational stiffness.

LAPED PLATE FLANGE



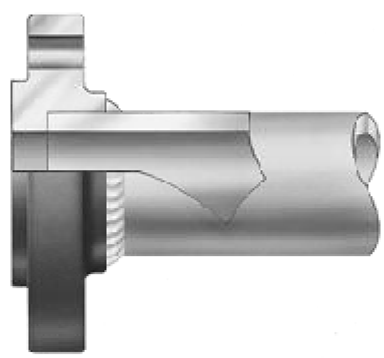
- Flange rotation is mostly decoupled from the pipe end hub and the gasket.
- Used in applications where the joint must be frequently disassembled for cleaning or where there is a need to facilitate bolt alignment.

SLIP-ON WELD PLATE FLANGE



- Designed to slide over the outside diameter of the pipe.
- Flanges are attached to the pipe by fillet welding at the hub and at the end of the pipe inside the flange.
- Not normally used in high stress applications.

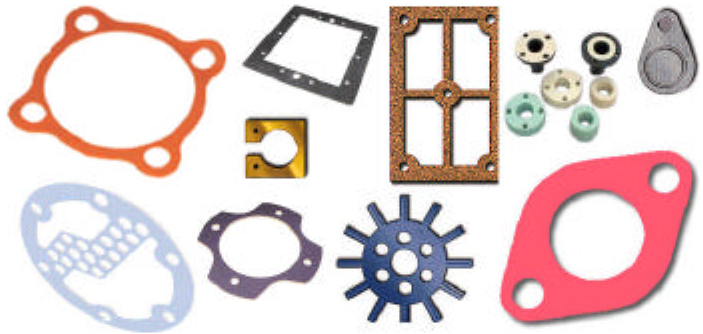
WELD-SOCKET PLATE FLANGE



- Similar to slip-on flanges except they have a bore diameter equal to that of the matching pipe.
- Have a counterbore from the hub side slightly larger than the outside diameter of the matching pipe which provides a "socket" into which the end of the pipe is inserted.
- Flange is attached to the pipe by a fillet weld at the hub.

Types of Gaskets

Gaskets come in different shapes to suit the flange geometry and sealing requirements.

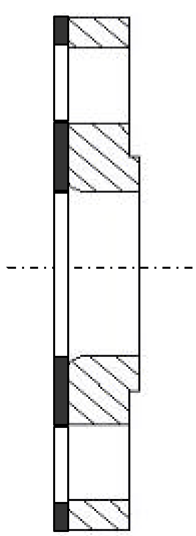


Commercially available gaskets are made of different materials to satisfy different performance characteristics. Below is a product line from one manufacturer.

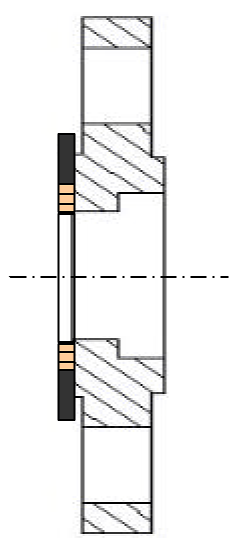
- Adhesive Backed Gaskets
- Automotive Gaskets
- Cork Gaskets
- Custom Gaskets
- Die Cut Gaskets
- EPDM Gaskets
- Felt Gaskets
- Flange Gaskets
- Foam Gaskets
- Gasket Material
- Gaskets
- Neoprene Gaskets
- Oil Resistant Gaskets
- Plastic Gaskets
- Rubber Gaskets
- Silicone Gaskets
- Sponge Rubber Gaskets
- Urethane Gaskets

Additionally, depending on how the gasket is utilized in the joint, it is grouped under different names.

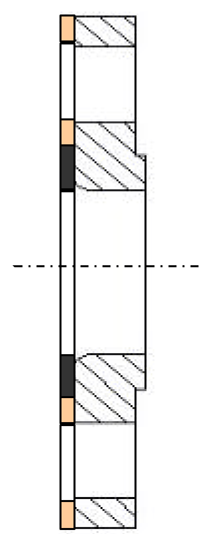
Full-face gasket



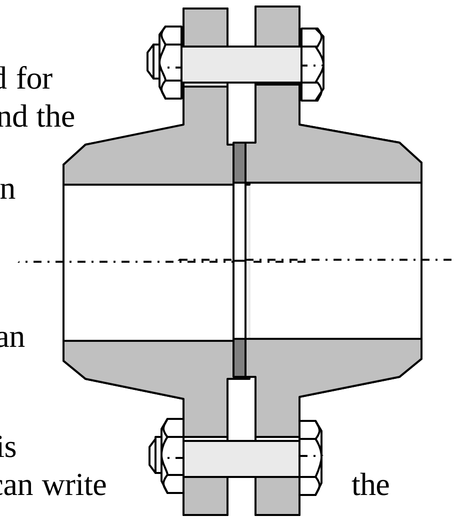
Controlled compression spiral wound gasket, centered off bolts



Controlled compression gasket, extending beyond bolt circle



Analysis of Flanged Joints – Simplified Model

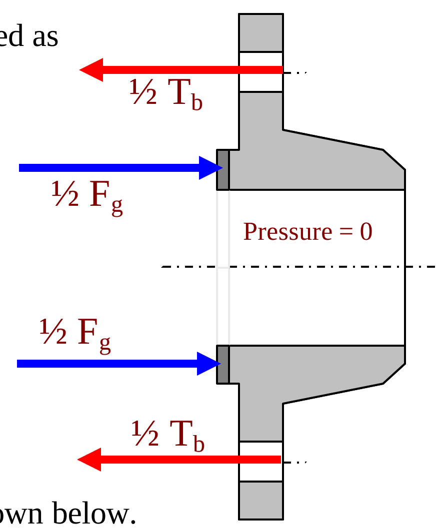


The taper-hub weld-neck flanges shown here, is coomonly used for high duty applications. In a simplified view , the flages, bolts and the gasket assmbly can be treated as linear springs. This would be the case when the flange rotation due to moment produced when the bolts are tightened, is ignored.

Remember the goal of the gasketted joint is to make sure that the stress in the gasket is sufficient to keep the leak rate below an acceptable limit (no leak may not be a practical goal)

From free-body diagram of flange, we find that when the joint is assembled, before the fluid in the tube/pipe is pressurized, we can write the an axial force equation for equilibrium condition:

$T_b = F_g$ ----------- (Eq. C-01)



Also the spring rates for the gasket and bolting can be expressed as follows.

Axial stiffness f the bolting:

$K_g = E_g A_g / L_g$ ----------- (Eq. C-02)

Compressive stiffness of gasket

$K_b = E_b A_b / L_b$ ----------- (Eq. C-03)

The above is derived from the relationships of axial spring stiffness and stress vs. strain relation of material properties shown below.

For axial spring,

$F = K X$ or $K = F/ X$

From stress and strain relation

Stress $(F/A) = E$ Strain (X/L)

Or $F = E A (X/L)$

Substituting F in the expression for K,

$K = E A (X/L) / X = EA/L$

Where K =stiffness, X = Deflection, F = Force, L = Length, and E = Modulus of elasticity

The stifnness values of gasket and bolting are key characteristics as they can be used readily to controll forces by controlling the displacement. Also from known values of area and the compressive stress limit of, say gasket, the allowable displacement can be detremined.

Notations:

T_b = Force (Tension) in the bolts (lbf)

A_b = Crossectional area of the bolt (sq.in.)

E_b = Modulus of elasticity of bolt material (psi)

L_b = Effective clamp length of the bolt (inch)

K_b = Axial stiffness of the bolting (lbf/in)

F_g = Compressive force in the gasket (lbf)

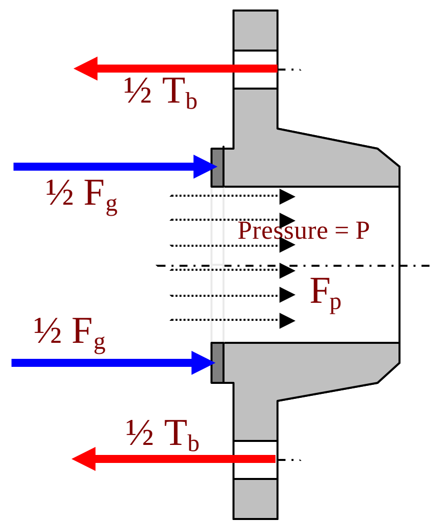
A_g = Gasket contact area (sq.in.)

E_g = Modulus of elasticity of gasket at no load (psi)

L_g = Thickness of gasket (inch)

K_g = Compressive stiffness of gasket (lbf/in)
P = Internal pressure (positive or negative, psi)

F_p = Force due to pressure in the tube



When the tube is pressurized, the equilibrium equation gives:

$$T_b = F_g + F_p \qquad \text{----------} \qquad \text{(Eq. C-04)}$$

Design Steps:

1. From the pressure in the pipe and its geometry, force due to the pressure (Fp) can be calculated.
2. Based on the selected gasket (code) geometry and properties, force in the gasket (Fg) can be calculated.

   After the gasket seating is pressurized, there would be a tendency to separate the joint and reduce the gasket seating stress. The operating bolt force must resist the separating pressure and maintain a sufficient gasket load to retain seal. As per the ASME Boiler and Pressure Vessel Code design rule, the bolt force component on the gasket (Fg) is also dependent on the pressure in the tube as per the following relation:

   $$T_b = F_g + F_p$$

$$T_b = (0.785 D_g^2 P) + (2W_g \times 3.14\, D_g\, M\, P) \quad \text{(total load on the bolt)}$$

Where D_g = Gasket diameter, W_g = Effective width of the gasket, M = Gasket factor (experimentally determined)

Note: Load required on the bolt is gasket area times pressure, plus presuure on TWO times the effective width of gasket times a gasket factor (M, constant). In reality, the gasket sealing will be affected by flange rotation and its rgidity which are ingnored in this calculation.

3. Total load in the bolt now can be calculated using Equation C-04
4. Based on the required bolt load, using the torque vs. tension relationship, the torque necessary to obtain the tension can be calculated.

Tightening strategies and their preload accuracy are:

Torque: ±35%
Torque-Angle: ±15%
Torque-to-Yield: ± 7%

Determine torque for achieving desired preload using the following equation:

T=KDF (Torque = K factor x screw diameter x pounds of tension induced in bolt)

For example, a 1/4"-20 socket screw with UNRC threads and a K factor of 0.21 has yield strength 5,073 lbs. minimum. Using 75% of that, 3,805 lbs, gives us the pounds of tension induced in the bolt. Consequently, our formula becomes: T = .21 x .25 x 3805, or T = 200 lbin

Standards and Codes for Flanged Joints

The quickest and most efficient way to design the joint is to select the standard joint design. The details of the standard joints are specified in national standards like the ones shown below.

- ASME B16.5 – 1988, Steel Pipe Flanges and Flanged Fittings (Formerly ANSI B16.5 and then ASME/ANSI B16.5)
- International Standard ISO 7005, Metallic Flanges, Part 1:Steel Flanges.
- ASME B16.47 – 1990, Large Diameter Steel Flanges: NPS 26 Through NPS 60.

The joint design codes provide procedures and steps that allow the user to design gasketed joint that is structurally sound and meets the leakage control specification. Through a series of formulas and charts, the code offers key data like:

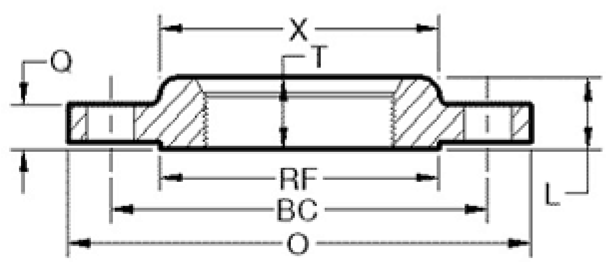
- Dimension of flanges and bolt
- Face details – dimension, finish, and permissible defects.
- Bolt-hole templates
- Weld geometry and dimensions.
- Material specifications.
- Ratings - maximum allowable non-shock pressure/temperature limits for each size and duty class
- Etc

Code Terminologies

Standard flange designs are grouped based on their application suitability. On ASME B16.5, the joints are grouped in Class 150 (lightest duty design) through Class 2500 (heaviest duty designs. On the other hand, the international standard ISO 7005 categorizes the flange designs in PN 2.5, the light duty, to PN 420 for the heaviest duty designs. Flange design for ASME Class 150 is shown below

Threaded Flange

1/2" through 24" NPS
Class 150 through 1500
(to 12" NPS max. for Class 2500)
(Larger Sizes available on request)



Threaded Flange

Threaded Pipe Flanges - Class 150

Nom. Pipe Size	Flange Dia. (0)	Flange Thick (Q)	Hub Dia. At Base (X)	Raised Face Dia. (RF)	No. Hole	Dia. Bolts	Dia. Of Bolt Circle (BC)	Length thru Hub (L)	Thread Length (T)
1/2	3-1/2	7/16	1-3/16	1-3/8	4	1/2	2-3/8	5/8	5/8
3/4	3-7/8	1/2	1-1/2	1-11/16	4	1/2	2-3/4	5/8	5/8
1	4-1/4	9/16	1-15/16	2	4	1/2	3-1/8	11/16	11/16
1-1/4	4-5/8	5/8	2-5/16	2-1/2	4	1/2	3-1/2	13/16	13/16
1-1/2	5	11/16	2-9/16	2-7/8	4	1/2	3-7/8	7/8	7/8
2	6	3/4	3-1/16	3-5/8	4	5/8	4-3/4	1	1
2-1/2	7	7/8	3-9/16	4-1/8	4	5/8	5-1/2	1-1/8	1-1/8
3 *	7-1/2	15/16	4-1/4	5	4	5/8	6	1-3/16	1-3/16
3-1/2	8-1/2	15/16	4-13/16	5-1/2	8	5/8	7	1-1/4	1-1/4
4	9	15/16	5-5/16	6-3/16	8	5/8	7-1/2	1-5/16	1-5/16
5	10	15/16	6-7/16	7-5/16	8	3/4	8-1/2	1-7/16	1-7/16
6	11	1	7-9/16	8-1/2	8	3/4	9-1/2	1-9/16	1-9/16
8	13-1/2	1-1/8	9-11/16	10-5/8	8	3/4	11-3/4	1-3/4	1-3/4
10	16	1-3/16	12	12-3/4	12	7/8	14-1/4	1-15/16	1-15/16
12	19	1-1/4	14-3/8	15	12	7/8	17	2-3/16	2-3/16

14	21	1-3/8	15-3/4	16-1/4	12	1	18-3/4	2-1/4	2-1/4
16	23-1/2	1-7/16	18	18-1/2	16	1	21-1/4	2-1/2	2-1/2
18	25	1-9/16	19-7/8	21	16	1-1/8	22-3/4	2-11/16	2-11/16
20	27-1/2	1-11/16	22	23	20	1-1/8	25	2-7/8	2-7/8
24	32	1-7/8	26-1/8	27-1/4	20	1-1/4	29-1/2	3-1/4	3-1/4

* Data used for example calculation shown below.

ANSI/ASME B16.5-1996 Class 150

Pipe Size	Outside Diameter	Inside Diameter	Number of Bolt Holes	Bolt Hole Diameter	Bolt Circle
1/2"	3-1/2	0.88	4	5/8	2-3/8
3/4"	3-7/8	1.09	4	5/8	2-3/4
1"	4-1/4	1.36	4	5/8	3-1/8
1-1/4"	4-5/8	1.70	4	5/8	3-1/2
1-1/2"	5	1.95	4	5/8	3-7/8
2"	6	2.44	4	3/4	4-3/4
2-1/2"	7	2.94	4	3/4	5-1/2
3"	7-1/2	3.57	4	3/4	6
3-1/2"	8-1/2	4.07	8	3/4	7
4"	9	4.60	8	3/4	7-1/2
5"	10	5.66	8	7/8	8-1/2
6"	11	6.72	8	7/8	9-1/2
8"	13-1/2	8.72	8	7/8	11-3/4
10"	16	10.88	12	1	14-1/4
12"	19	12.88	12	1	17
14"	21	14.14	12	1-1/8	18-3/4
16"	23-1/2	16.16	16	1-1/8	21-1/4
18"	25	18.18	16	1-1/4	22-3/4
20"	27-1/2	20.20	20	1-1/4	25
24"	32	24.25	20	1-3/8	29-1/2

Note:
This table is intended for estimating purposes only. Refer to the ASME B16.5-1996 standard for complete engineering and applications information. All dimensions are in inches unless otherwise noted. Inside diameter and thickness dimensions are application specific. Recommended bolts are 1/8" smaller than the nominal bolt hole diameters shown.

Example Flange Joint Design

Consider a Class 150 3 in. diameter tube from the table above subjected to 250 psi.

Bolts: 5/8 in. dia, 4 bolts (Gasket width is obtained from RF data in the Table in page C-7)
Gasket width: (5.00 – 3) = 2 in., Effective width = 2.0 x 0.5 = 1.00 in.
Gasket Dia: 3 + 1.0 = 4.00 in. M (gasket factor) =1.0

Then, the total load on 4 bolts

$$\mathbf{T_b = (0.785D_g^2\ P) + (2W_g \times 3.14\ D_g\ M\ P)}$$

$$T_b = (0.785 \times 4.00^2 \times 250) + (2 \times 0.625 \times 3.14 \times 4.00 \times 1 \times 250)$$

$$= 3{,}140 + 3{,}925$$

= 7,065 lbs. (tension in bolts, F in toque vs. tension formula)

Tension in each of the four bolts = 7,065 /4 = 1, 766 lbf.

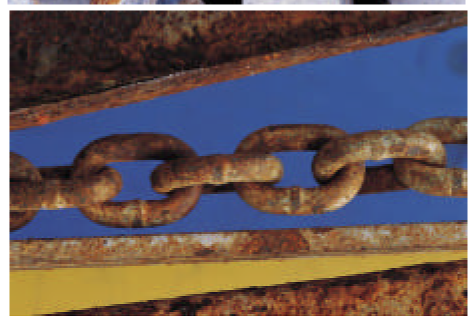
Assembly Torque:

Assuming, Nut factor, K =0.25, **T = KDF**

Or Torque required, T = 0.25 x (5/8) x 1,766 = 276 in. lb.

Topics	Group Exercises
Flange Joint	**C-2** **a:** Specify bolt torque required in a flanged joint to keep the leakage within limit. Use a class 150 threaded flanges to connect two 5 in. diameter pipes carrying steam at 225 psi.. (Assume gasket factor, M = 1 and nut factor, K = 0.27) Hint: See example at page C-9, use Table at page C-7 for 5 in dia. pipes.)
Flange Joints	**b:** Design a joint using class 150 threaded flanges to connect two 2 in. diameter pipes carrying steam at 300 psi. Specify the torque required for the bolts to keep the leakage within limit. (Assume gasket factor, M = 1 and nut factor, K = 0.32, 4 5/8 dia. bolts) Hint: See example at page C-9, use Table at page C-7 for 2 in dia. pipes.)

Section – D
Causes and Prevention of Bolt Failure



Corrosion

Effect of Corrosion

- accelerates mechanical failure like fatigue and thread stripping
- initial buildup of corrosion/rust. Can increase tension in bolts (increased joint dim.).
- excessive corrosion can lead to total loss of clamping load as joint weakens.
- It could cause sudden/unexpected loss of clamping force due to mechanism of hydrogen embrittlement or stress corrosion cracking.

Science of Corrosion

ALL METALS have electrical potential which

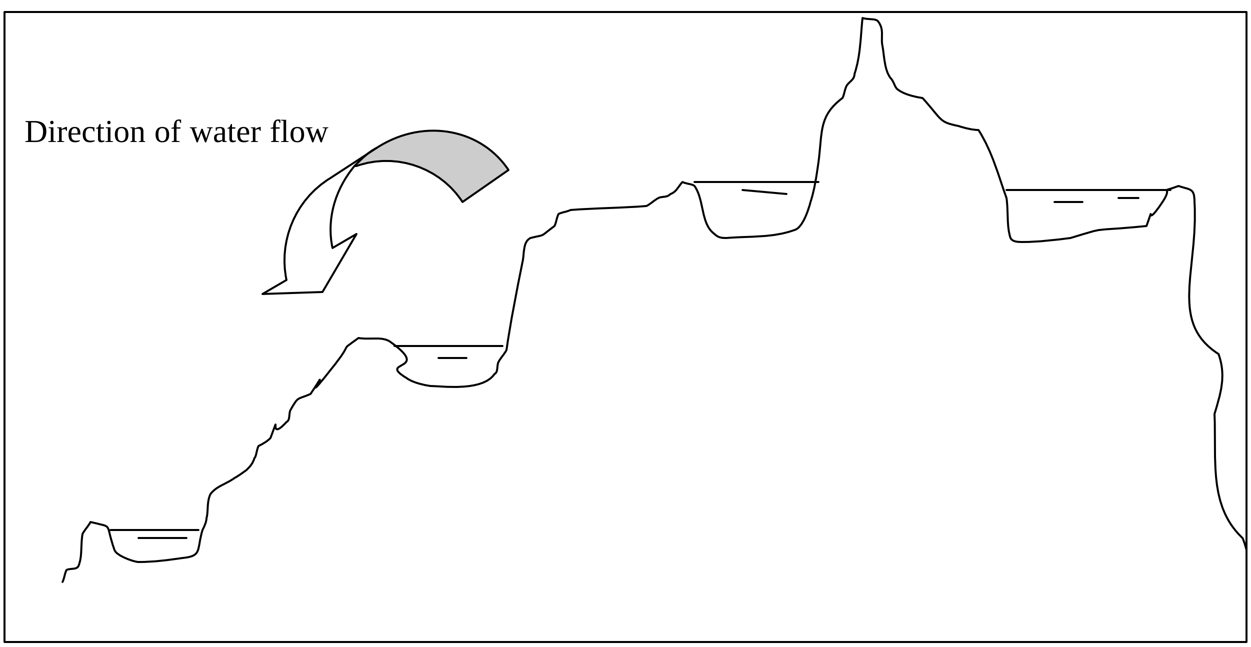
- Depend on atomic structure
    - Indicates how easily it gives out or absorbs electron

Analogy

For water trapped in higher elevation in the mountain, potential energy available is dependent on the height of the lake.

*Elevation = Electrical potential

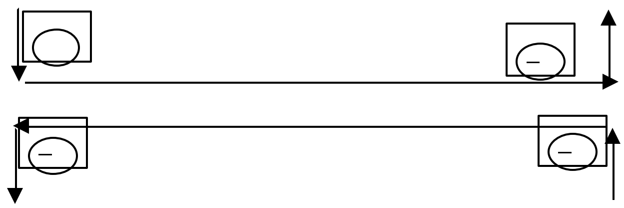
Elevation determines which way the water flows.



Anodic materials – gives out electrons <u>more</u> readily
Cathodic materials – absorbs electron <u>more</u> readily

- Anodic and Cathodic are relative terms.
- No material can be just anodic or Cathodic.
- Any material can serve either function.

Example: **Steel (anode)** in presence of **Brass (cathode)**

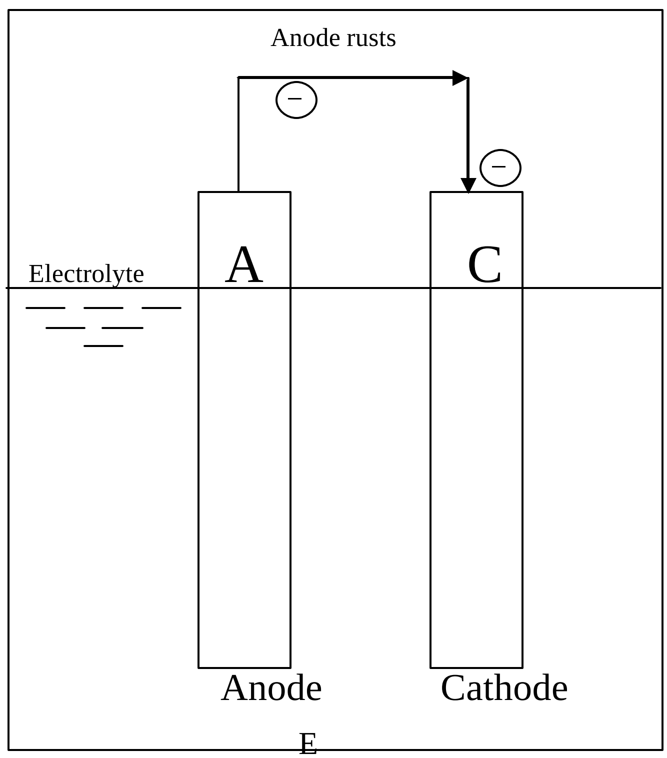


But **Steel (cathode)** in presence of **Magnesium (anode)**

<u>A</u>node – gives electron <u>A</u>way
<u>C</u>athode – <u>C</u>ollects electron

An electrical battery is formed when two metals of different electrical potentials are connected by a conductor and a liquid.

Under this condition
MORE ANODIC OF THE TWO METALS WILL CORRODE.



Electrical Battery
A – Anodic metal, C = Cathodic material

Depending on the electronic potential, metals are arranged in a **galvanic series table**.

ANODIC – Most likely to corrode (least noble).

- **Magnesium**
- Zinc
- Aluminum 1100
- Cadmium

- Aluminum 2024-T4
- Steel Iron
- Cast iron
- Chromium iron (active)

Higher location on the mountain indicates higher electrical potential (ANODIC) – more likely to lose electrons and Corrode

* Nickel resist cast iron
* Types 304 and 316 stainless steel (active)
* Tin
* Nickel (active)

* Inconel (active)
* Hastelloy Alloy C (Active)
* Brasses
* Copper
* Bronzes

Active – easily gives out electrons

Passive – Less likely to lose electrons (Due to an oxide layer.)

- Monel nickel
- Nickel (passive)

- Silver
- Titanium
- Graphite
- **Gold**

ANODIC – Most likely to corrode (Least Noble)
CATHODIC- Least likely to corrode (Most Noble)

Conditions that influence electrical potential (Anodic or Cathodic status – likelihood of corrosion)

- Metals can be in active or passive status.
  Passivity reduces likelihood of corrosion by the presence of an oxide layer

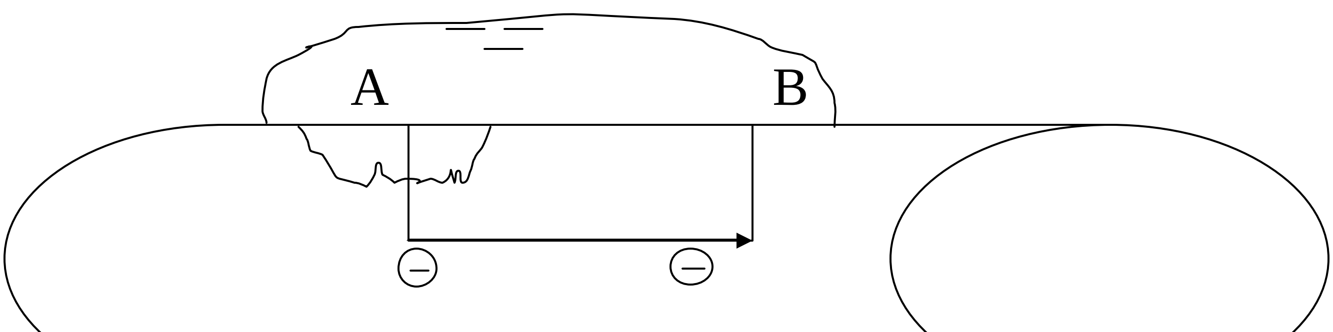
- Increased stress level makes more Anodic

- Higher temperature – makes more Anodic

- Some materials are more Anodic in the presence of impurities and near the grain boundaries.

- Oxygen in electrolyte (liquid which connects the two metal/area) also makes metals more anodic – making it easier to give out electrons.

Above reasons make it possible for a single body to act as both Anode and Cathode.



Steel Rod exposed to moisture

A – an area of high stress concentration
B – an area exposed to moisture

Areas A and B are connected by moisture (acting as electrolyte)

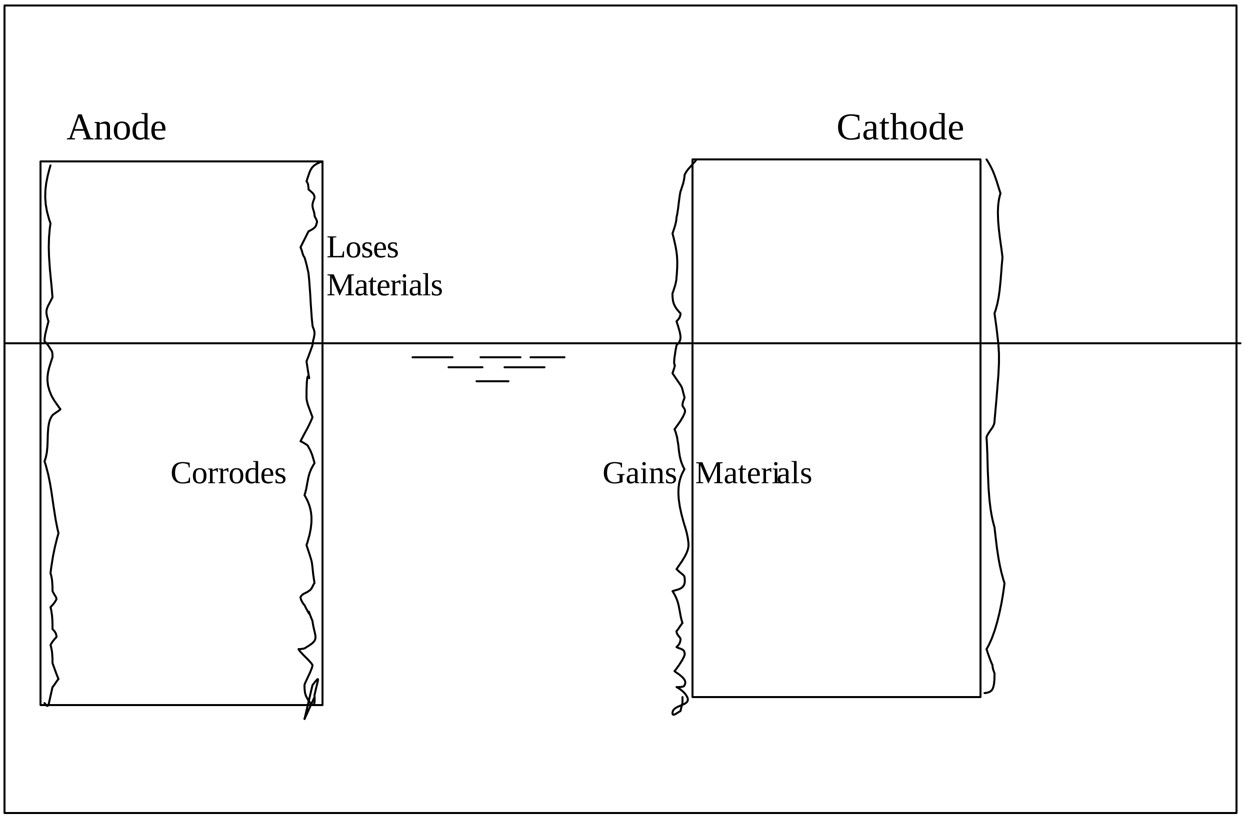
Essential conditions for Corrosion

- Anode
- Cathode
- Electrolyte (conducting liquid)

- Metallic connection

A body will only CORRODE when it is connected to another metal of different electrical potential and both are immersed or wetted by some solution (forms miniature batteries).

Anode in the battery provides electrons, which flow to cathode. In this process the **anode is gradually destroyed/corrodes**. The cathode, on the other hand, collects electrons/materials, which plates out on its surface.



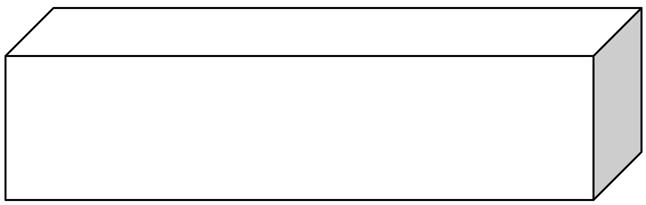
Two Metal Corrosion

Corrosion is inevitable when two metals are different and connected together in the presence of a fluid.

Example:
Steel bolts on Aluminum towers exposed to seawater.

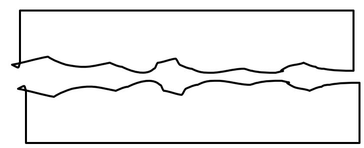
The likelihood or severity of corrosion depends on how far apart the two metals are in the galvanic Series.

Single Metal/Part Corrosion



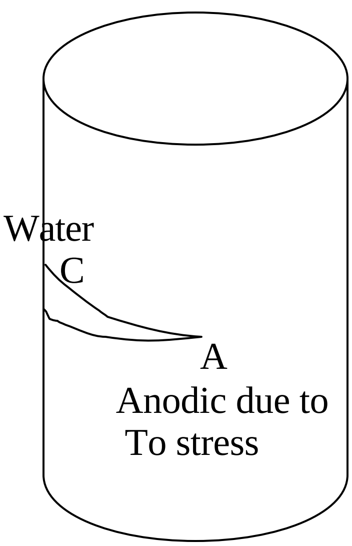
Broken oxide film makes
Localized area more anodic

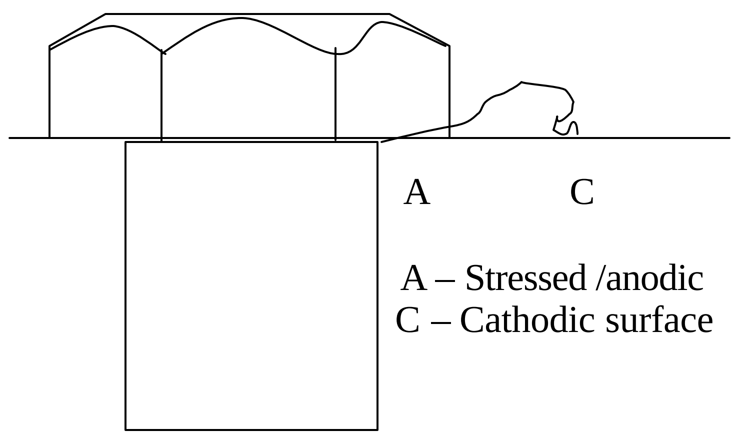
Fretting Corrosion



Rubbing of oxide coated bodies.
Loss of coating – becomes ANODIC

Crevice Corrosion





A – Stressed /anodic
C – Cathodic surface

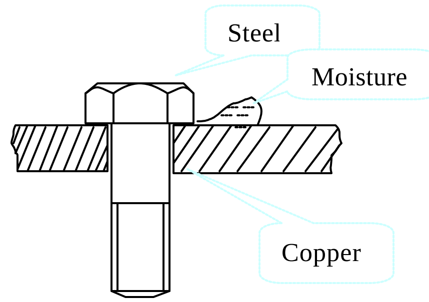
Chemistry of Corrosion

Assume **Steel Bolt** on C**opper Joint** materials exposed to slat water.

1. Positive particles of metal (steel) dissolve in moisture absorbing oxygen & hydrogen forming *ferrous ions.*

2. Negative charged electrons flow through steel to copper into the moisture where they combine with oxygen and water to form *hydroxyl ions.*

3. Hydroxyl ions combine with ferrous ions producing *iron oxide* (rust), the corrosion product.



Stress Related Corrosion

Sudden fastener failure takes place due to combinations of

- tensile stress and corrosion
- tensile stress and absorbed hydrogen

Failure mechanisms

- Hydrogen embrittlement
- Stress embrittlement
- Stress corrosion cracking
- Hydrogen-assisted stress corrosion

Hydrogen Embrittlement (HE)

Atomic hydrogen gets trapped under plating of fasteners. When stressed, hydrogen migrates to point of stress concentration. Pressure created by the hydrogen initiates crack propagation – subsequently, failing the fasteners.

It is not corrosion related. However, it often occurs in “corrosion resistant” bolts.

How to prevent:

- After plating, bake fasteners properly to get H out.
- Use coatings that do not include electroplating
- Use softer materials for fasteners. Harder materials are more susceptible to failure.

H.E. is mostly associated with carbon and low alloy steel.

Strategies for Corrosion Reduction

Philosophy: Eliminate condition that forms corrosion "battery". Remove one or more of the four necessary conditions/ingredients for the battery to function.

1. The anode,
2. The cathode,
3. Electrolyte, and
4. The metallic connection between the anode and the cathode.

Commonly suggested design tips:

1. Keep joints dry. Provide drainage or remove electrolyte.

2. Select fastener material that is closer in the galvanic series. (Use Brass or Bronzes for Copper). Better design practice is to use identical material.

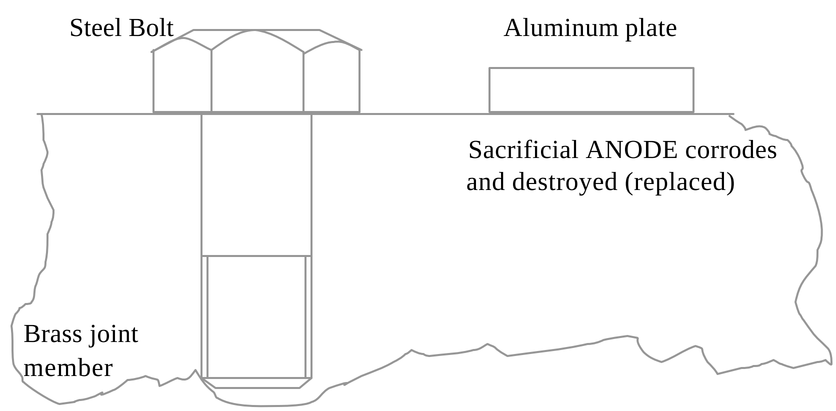
3. Since anode is destroyed in corrosion, design it to have larger dimension.

4. Break the metallic connection between the two parts by insulation (paint, coatings, gasket, and spacers).

5. Include a third sacrificial anode, which reverses the flow of current in the battery.

Galvanic Series

Aluminum
.
.
Steel
.
.
Brass



6. Attach an external battery in opposite polarity.

7. Seal cracks and crevices (paint, putty, and plastic washers).

8. Reduce stress concentrations by designing FILLETS, POLISHED or SHOT PINNING SURFACES, distribute load evenly, avoid bending load, use conical/crowned washers.…

9. Use inhibitors in electrolyte, which retards its capacity to transport electrons.

10. Use materials that resist electrolyte. Industrial Fastener Institute (IFI) publishes list of materials with resistance to different solution/materials.

    Example: Type 304 stainless steel has excellent resistance to fresh water, wine, sulfur, turpentine. It has poor resistance to sulfuric acid or zinc chloride.

11. When none of the above can be implemented, consider replacing the bolt periodically.

Combating Corrosion

Most fasteners are typically highly stressed, intricate, and small parts. Even a slight corrosive attack can cause detrimental effect on such parts. Therefore, proper measures must be taken to guard against corrosion by using special materials, coatings, and finishes.

Use fasteners made of corrosion-resistant materials:

Industry		Materials preferred
• Aerospace	→	Titanium, Inconel
* Automotive	→	Aluminum, plastics
* Marine	→	Monel, Silicon Bronze
* Petrochemical	→	Stainless Steel (all kinds)

Coatings of Fasteners

Coating fasteners to reduce corrosion is most common. (90% of all carbon steel bolts are coated).

Coating resists corrosion by

- providing a barrier which isolate bolt from the corrosive environment
- including “passivation/inhibition” that slows down the corrosion process
- acting as “galvanic” or sacrificial layer that can reverse the direction of the current in the battery.

Coating Quality and Side Effects

Quality -

For barrier protection, coating must be perfect. A slight break in coating (paint, cad plating,…) can create a localized anodic region that will create a battery.

Sacrificial coating need not be perfect.

Side Effects - Impact on bolt performance

Coatings may be used to produce many desirable effects

- reduces friction between thread, collar and the part
- reduces torque necessary to overcome frictional drag
- good coatings reduce variability of friction and improve torque-tension relationship
- change appearance of fasteners, matching colors to hide it. Or even make fasteners stand out as design accents. (Non-structural purpose)

Common Types of Coating

Organic Coating

- Made from plant or animal matters.
- Contain carbon compounds
- Can provide twice the corrosion protection of things such as cadmium and zinc plating. They also come in different colors.
- Eliminates (not present) hydrogen embrittlement

  Application method – dip, spin, spray, paint.

Paints (organic):

Alkyd and phenolic paints were popular before. Fluorocarbons and other polymers are used now. Zinc-rich paint is used for structural steel where entire joint, rather than bolt alone, is pained.

Phos-oil coating (organic):

Mild acids like zinc phosphate and manganese phosphate in a tumble bath gives the bolt a porous surface which provides an excellent base for retention of oils, waxes or other lubricants.

Phosphate + oil
Phosphate + paint + oil
Phosphate + zinc rich paint

Common practices

Solid Film Organic Coatings:

Produce "a thin layer of slippery paint".

Teflon-S
Stalgard
Xylan
Emralon
Everlube

↑
Fluorocarbons

Trade → names

Used on

* Carbon Steel
* Stainless Steel
* Aluminum

Chemicals → Molybdenum disulfide, graphite, or polytetrafluoroethylelne (PTFE)

In general the above organic fluorocarbons can be used in applications which involve changes in temperature.

-450°F (-268° C) to 400° F (+204° C)

Inorganic or Metallic Coatings

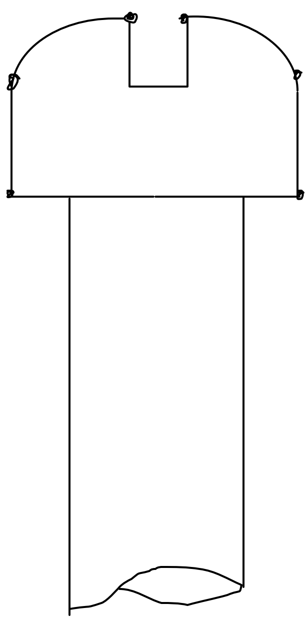
Inorganic materials do not contain any plant or animal matters. It includes metallic and ceramic compounds.

Application method – electroplating, hot dipping, vacuum deposition, mechanical plating.

Electroplated Coatings:

Common **materials** – Cadmium, Zinc (less expensive), Nickel, chromium, silver (more expensive).

Builds up on sharp edges
Rather than filling cracks.



Cadmium – better for marine environment

Zinc – better choice for most industrial applications.

[cadmium plating used to use cyanide rinse in the plating process which was considered as an environmental hazard.]

- **Zinc** is cheaper than **Cadmium** but not as lubricious.
- **Zinc** also doubles the torque-tension scatter.
- Also, produces dull “white dust”.

Hot Dip Coatings

Two materials are applied this way.

Aluminum – fasteners coated with aluminum are called “aluminized”

Zinc – Zinc coated fasteners are called “galvanized”.

- Inexpensive, used for high-strength fasteners like ASTM A325 structural steel bolts (often galvanized).

- The hot dip process is difficult to control – resulting in thickness variation. **Threads are undercut** to provide room for coating – but needs to be recut with a tap or due after coating has been applied.

- Lower stripping strength (thread) than mechanically galvanized one, but **more corrosion resistance** (greater thickness).

Mechanical Plating

In these process ductile metals such as zinc, cadmium, or tin are cold-welded onto the metal substrate by mechanical energy. Fasteners that mechanically plated with zinc are called **“mechanically galvanized”.**

Mechanically plating VS	**Electro-plating Hot dipping**
• Uniform coating thickness	* Thicker and non-uniform
• Absence of hydrogen embrittlement	* hydrogen embrittlement present
• Tempering – absent. No detempering needed	* Detempering may be needed
• Higher durability out of newer process which combines aluminum and zinc for coating threads. ½ mil thickness. No need for retap.	* Not done
• Phosphate coating over zinc & aluminum has been seen to provide better corrosion resistance.	* NA

Commercial Fastener Coatings

Type	Trade name	Manufacturer	Rating	Temp.
*Aluminum in Ceramic binder	SermaGard	Teleflex	1500 hr.	1200°F
*Cadmium	--	3M	320 days	--
*Cad/Tin (50/50)	--	3M	2000 days	--
*Cad/Zinc(50/50) Chromate	--	3M	6720 days	--
*Fluorocarbon (PTFE)	Teflon	DuPont	--	-450 +400°F
*PTFE plus Binding resin	Emralon 305	Acheson Colloids Co.	500 hr.	-100 +400°F
*Molydisulfide + epoxy binder	EcoaLube 642	E/M Corp.	500 h.	-365 +500°F
*MoS_2/graphite/ Phenolic resin	Electrolube E-40	Electrofilm Inc.	--	-459 + 450°F
*Surface alloy of nickel	Sanbond	AMCA Int'l.	--	+600°F
*Silver + indium + binder	Lube-Loc	Electrofilm Inc.	--	-459 + 450°F
*Zinc, plain	--	3M	192 days	+500°F

Ref: An Introduction to the Design and Behavior of Bolted Joints – John H. Bickford

Miscellaneous Coating Processes

Ion Vapor Deposition (IVD) – High-strength steel and titanium are coated with aluminum using this process:

- High resistance to stress corrosion cracking

High purity Nickel coating – This material is alloyed with carbon or alloy steel fasteners

- Good resistance to Severe acid and alkalines
- No buildup
- Allows additional coating of zinc or cadmium to enhance corrosion protection and increase lubricity.

Composite Coating:

Mixed – organic and inorganic materials

<u>Corrosion Resistance Rating</u>

ASTM B117 procedure – expressed in hours before significant damage by <u>salt spray.</u>

[Salt may not be your concern]

Modes of Joint Failure

The main function of a bolt is to clamp two or more joint members together. The objective of the assembly functio n/process is to introduce tension in the bolt that produces clamping force. A joint is considered to have failed when there is loss of claming force in the bolt.

There are a number of ways that **bolts can lose the clamping force** and the joint fail.

a. Mechanical Failure of the bolt

- too much tension during installation (excessive torque applied). (<u>This should never happen if the joint is designed* right)</u>
- Mechanical failure due to elevated temperature
- Corrosion destroyed the bolt
- Stress corrosion cracking failure
- Fatigue failure

*Design refers to proper material, quality, dimension, heat treatment, application specs. Etc.

- Poor heat treatment in low cost bolt – head may be pulled off.
- Bolt head welded to threaded rod –
- Bolt designed without proper fillet radii-.

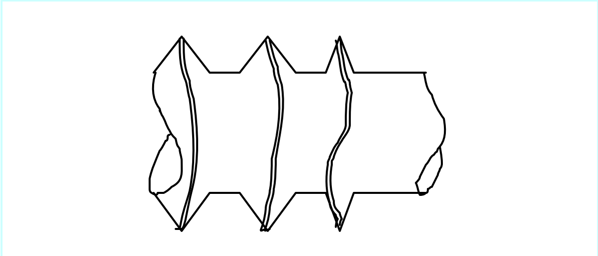
b. Stripping of Threads

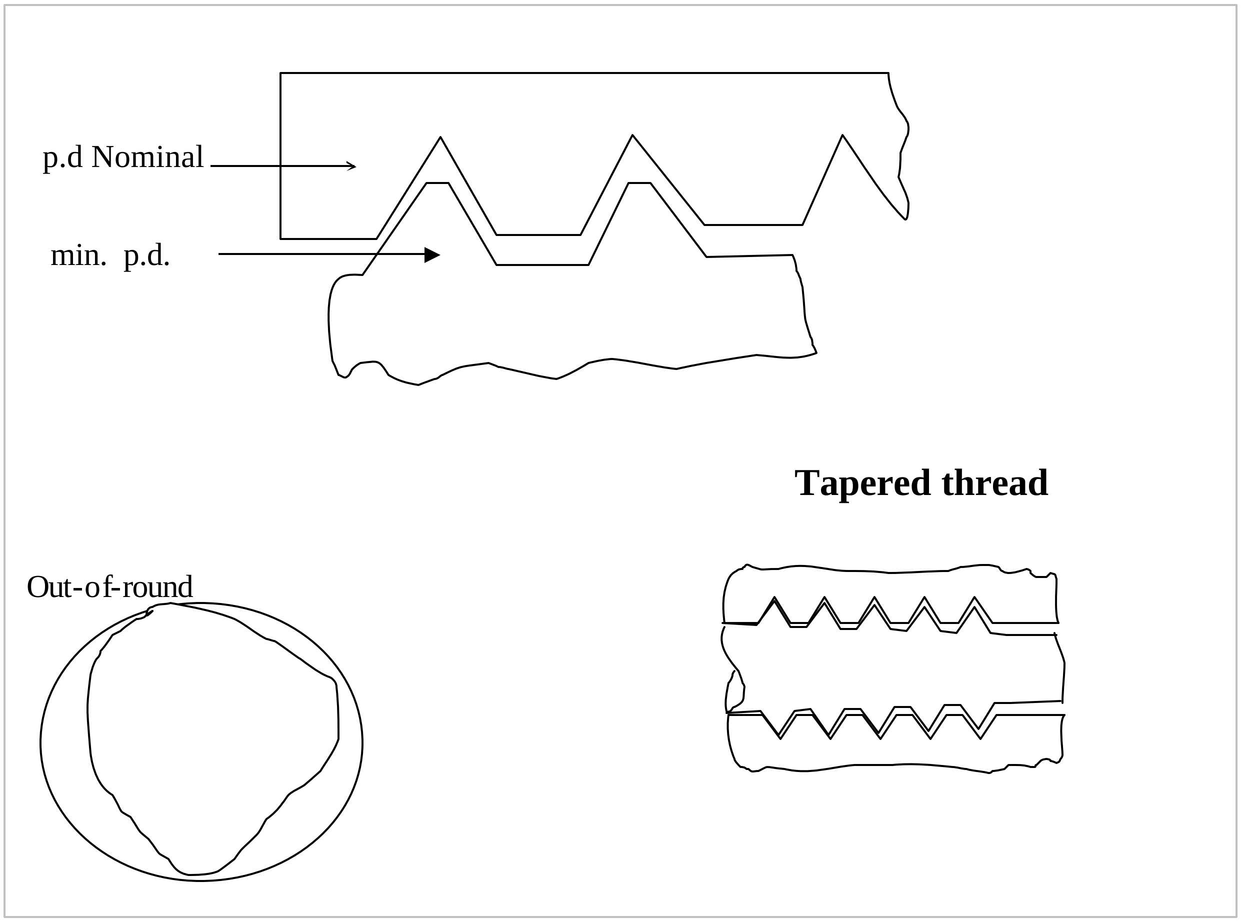
Threads may strip when proper contact and shear resistance area is reduced. This would generally happen when thread geometry is too far from the standard (ANSI B.1.1) -–pitch diameter of thread.

Say: Nominal pitch Dia = .5889(pd)
Tolerance = +0 - .0035

If the nut is at nominal, but the bolt is at minimum dia. allowed by the standard, this bolt will have 15% less thread strength. The thread strength goes down to 50% when bolt p.d. is lowered by 20 mils.

- flank angle
- tapered bolt and/or nut
- irregular helix angle →
- out-of-round bolt/nut.





c. Lost Bolts (self-loosened)

There can be no clamp load when the bolt is absent. A common cause of missing bolt is a phenomenon called self-loosening caused by:

- Vibration
- Temperature and pressure cycle
- Reversing loads in a direction right angle to bolt axis
- Carelessness
- Not installed due to hole misalignment
- Not installed as considered overdesigned.

d. **Loose Bolts**

Bolt not having appropriate tension is quite common. Loose bolts may cause “self-loosening” or Vice versa. The resulting failures are:

- Joint leakage
- Joint slip
- Cramping of machine members (viz. bearing misaligned)
- Fatigue failure
- Self-loosening

e. **Bolts Too Tight**

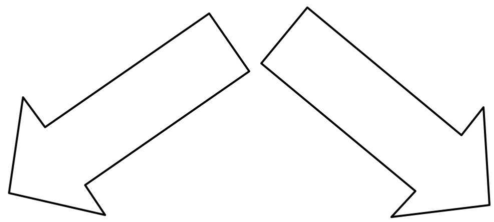
Excessive bolt tension/load can cause:

- Crushed gasket
- Damaged joint surface (galling
- Encourage stress corrosion cracking

Which Failure Mode should we be concerned about?	**Conditions for Failure**	**Importance of Correct Preload**
It depends on the industry segment/type of applications. **Auto:** - self-loosening - corrosion - leakage (head gasket) - vibration loosening **Petrochemical:** - leakage from gasket - corrosion **Structural Steel Industry** - joint slip - corrosion - fatigue **Aerospace:** - fatigue	Each type of failure is initiated by a limited number of “essential conditions” **Corrosion:** - An anode - A cathode - An electrolyte - A metallic connection between anode and cathode	Absence of proper preload has been seen to contribute to most modes of failure. Improper preload causes: **Corrosion** - higher load→anodic - lower load→leakage **Stress Corrosion Cracking** - raises stress level **Fatigue -** Higher load **Mechanical failure -** Higher load **Self-loosening -** Lower load

Mechanical Failure Modes of Bolted Joints

JOINT FAILURE



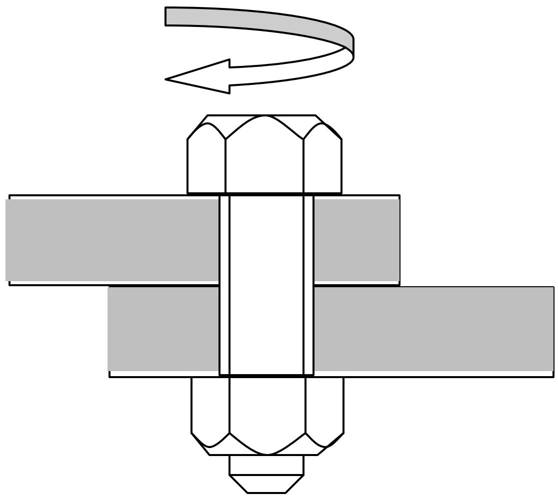
Bolt Failure

- Bolt head-Shear
- Bolt head-tensile
- Bolt body-tensile
- Bolt body – crush
- Thread –Shear
- Nut thread – Shear
- Bending at lead thread
- Galling (metal to metal cold
  Welding, at threads)

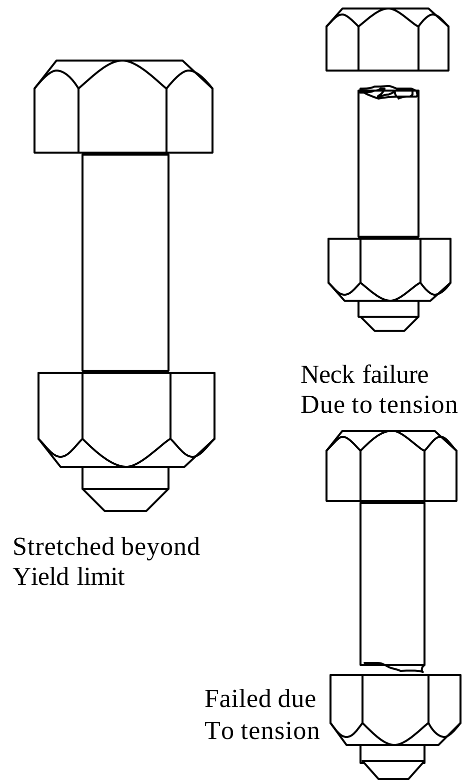
Joint Part Failure

* Crush of contact area
* Compression under bolt or nut
* Shear

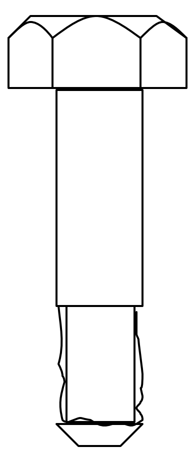
Bolt Under Torque



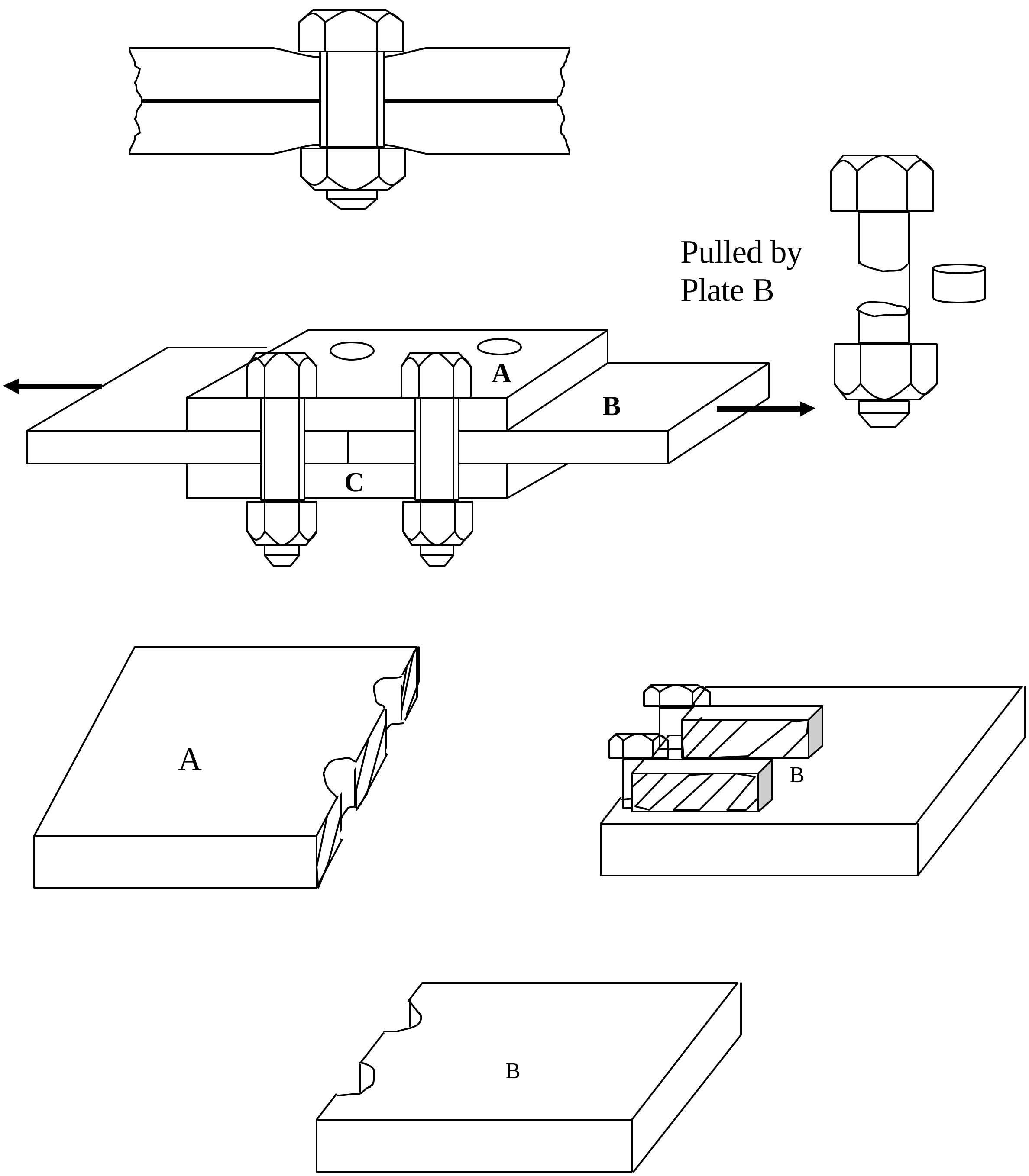
Bolt Failure under Excessive Torque



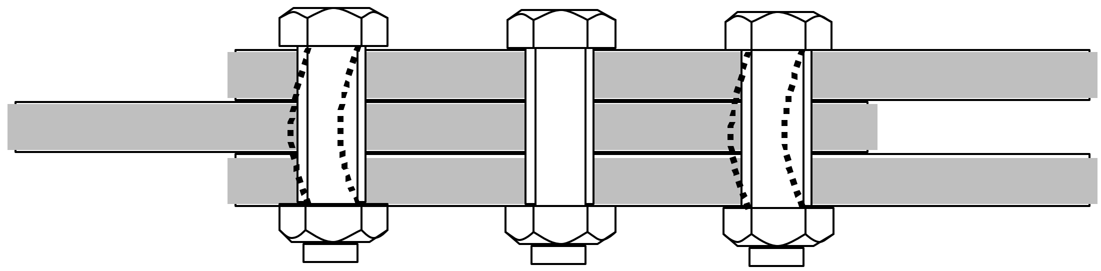
Stripped Thread Shear Failure



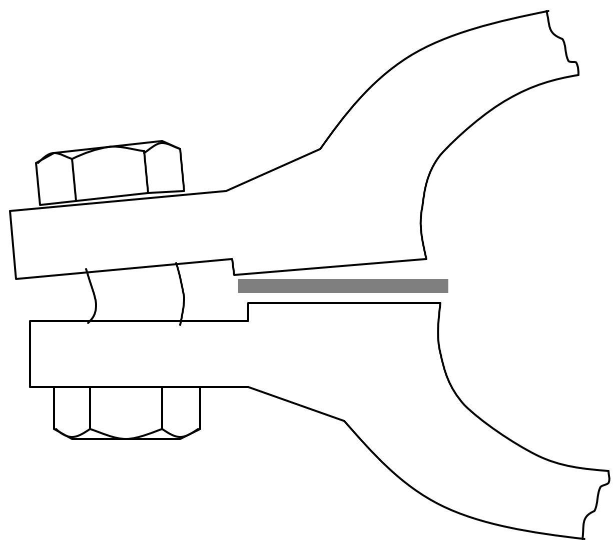
Part Failure Due to Excessive Load



Tensile Failure



Bolts failure in long joints – outermost bolts which carry the most load fail first.



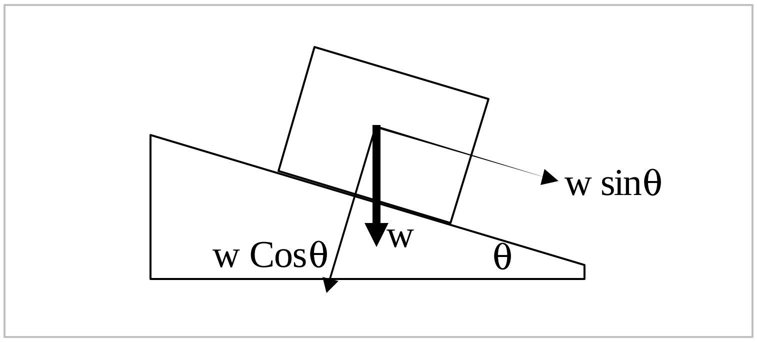
Flange rotation

As bolts are tightened, the outer edges Of flange are pulled together. The inner Diameter of gasket could be fully unloaded by this process.

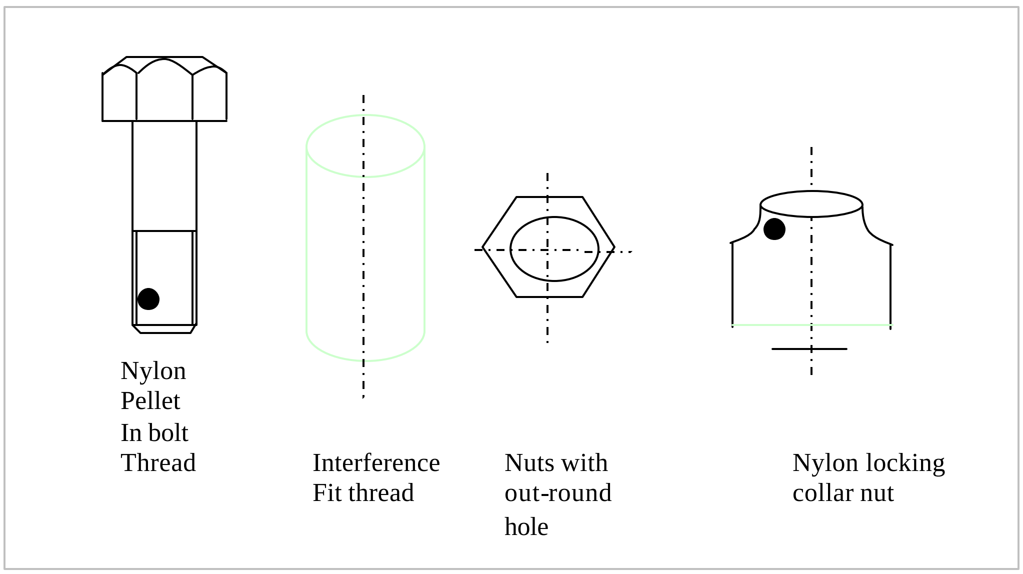
Prevention of Vibration Loosening

Common actions

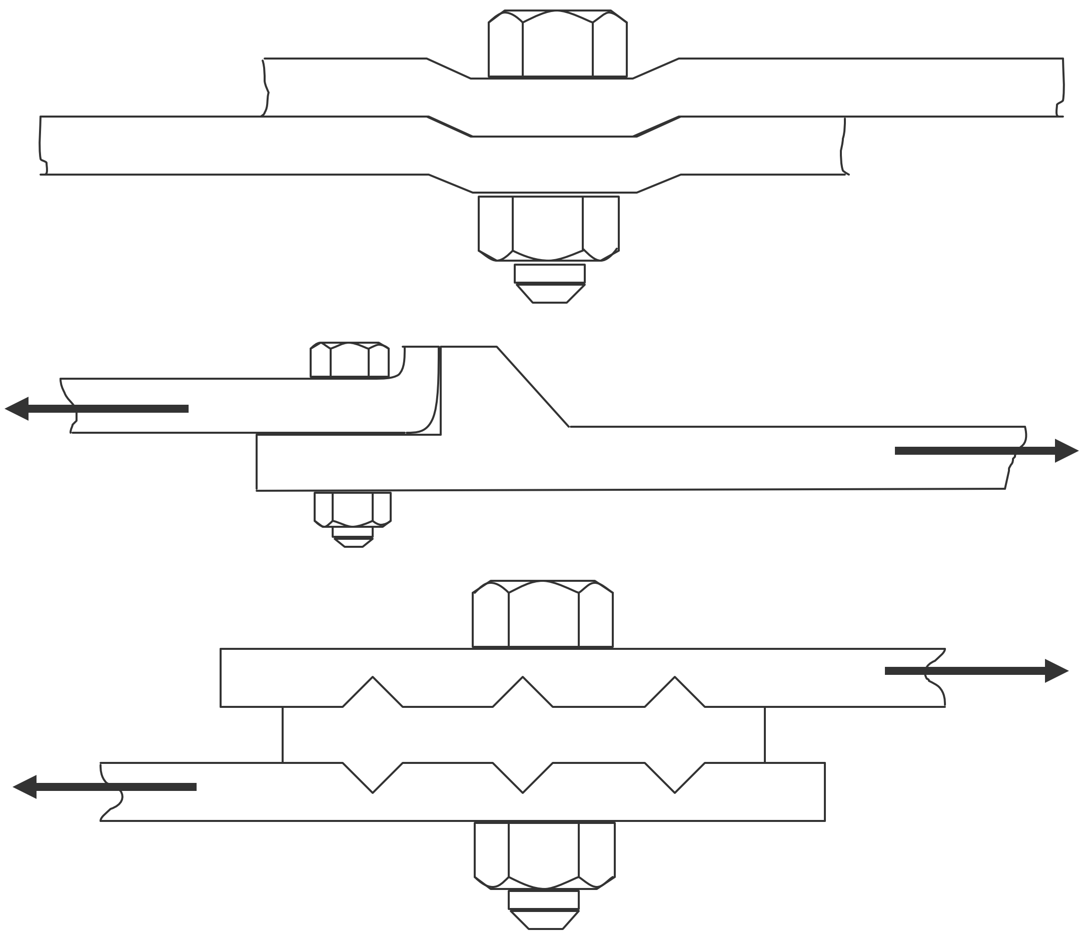
- Keep friction forces above the forces trying to loosen joint.
- Mechanically prevent slip between bolt/nut and the joint surfaces.
- Reduce the helix angle of the thread to reduce back-off torque component
- (w sinθ).



- Provide a "prevailing torque" or locking action of some sort which counters the back-off torque created by the incline plane of the thread.



JOINTS THAT RESIST TRANSVERSE SLIP

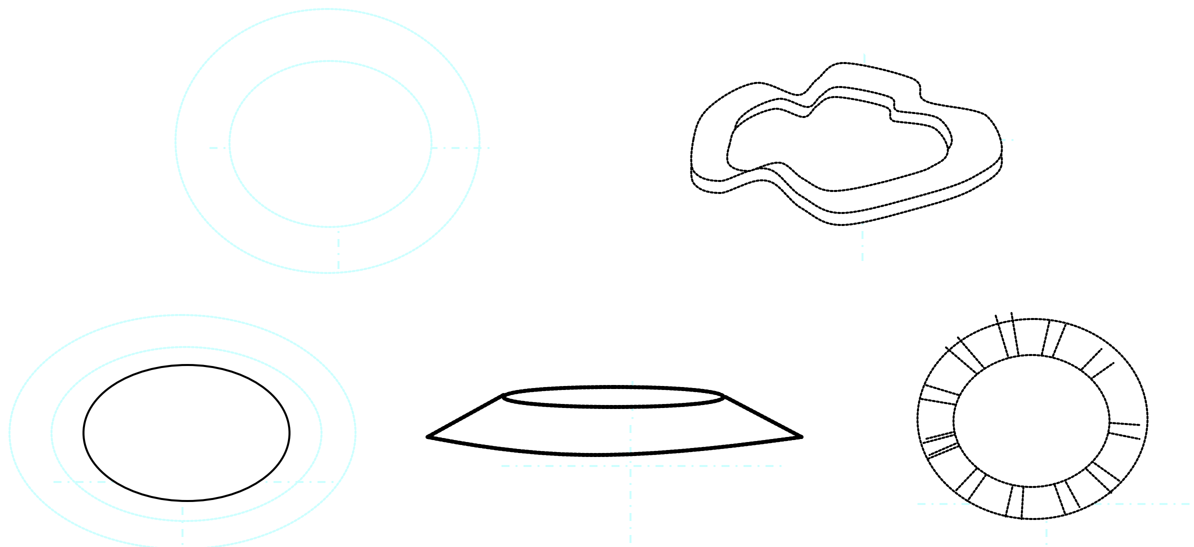


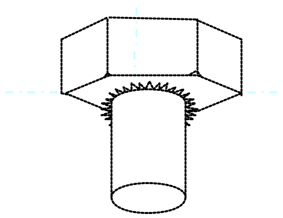
Toothed Shear Washer

Other devices

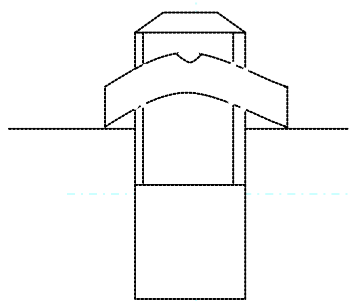
- dowel pin
- welding
- glue/adhesive

Lock or Spring Washers, Free-Spinning Lock nuts or Bolts, and Lock Bolts.

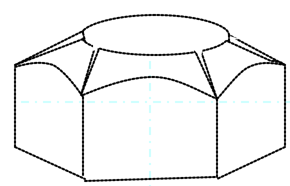




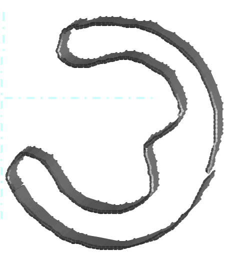
Serrated Head Bolt



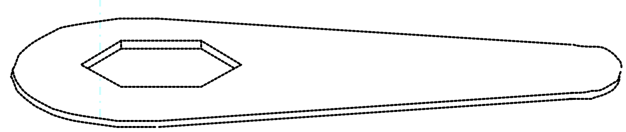
Spring Head Nut



Spring Arm on top of nut



Spring Clip



Retainer Plate

Mechanical Properties of Typical Medium/Low Carbon and Case Hardened Steels

s_u = Tensile Strength (ultimate Stress) s_y = Yield Strength (psi)

Steel	Condition	s_u	s_y	Steel	Condition	s_u	s_y
1010	Hot-rolled Cold-drawn	51,000 56,000	29,000 33,000	4320	Hot-rolled Cold-drawn	87,000 99,000	59,000 65,000
1020	Hot-rolled Cold-drawn	67,000 69,000	45,000 48,000	4340	Hot-rolled, annealed	115,000	95,000
1035	Hot-rolled Cold-drawn	88,000 92,000	55,000 59,000	4620	Hot-rolled Cold-drawn	82,000 98,000	55,000 70,000
1045	Hot-rolled Cold-drawn	99,000 110,00	60,000 69,000	4640	Hot-rolled, annealed	100,0076, 000	87,000 57,000
1112	Hot-rolled Cold-drawn	67,000 80,000	40,000 62,500	8620	Hot-rolled Annealed	91,000 76,000	64,000 57,000
2317	Hot-rolled Cold-drawn	85,000 95,000	56,000 75,000	8640	Hot-rolled Annealed	126,000 95,000	89,000 63,000
3140	Hot-rolled-annealed Cold drawn-annealed	96,000 115,000	64,000 98,000	9260	Hot-rolled, annealed	142,000	92,000
	1010 – 1020 Steels are used for common fasteners						

Numbering Systems for Carbon and Alloy Steels

The numbering system of the Society of Automotive Engineers (SAE) and American Iron and Steel Institute (AISI) is based on the chemical composition, and provides a simple means of identifying steel.

Indicate type of alloy (alloy classification)

4 (or 5) digit designation **X X X X X**

Indicate carbon content (2 or 3 digits)

Example: Steel 1020 → plain Carbon Steel with .20% carbon

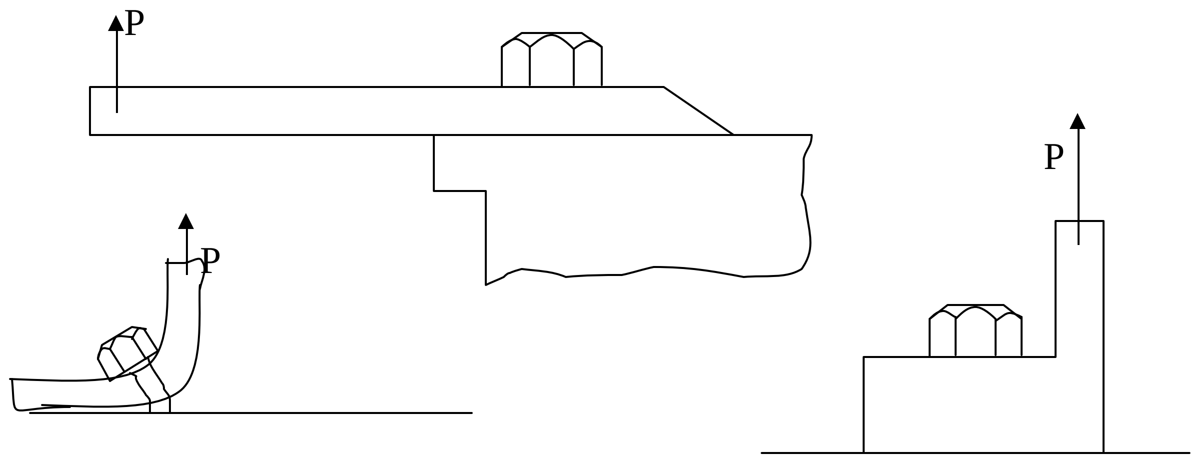
Basic Numerals for SAE and AISI Steels

Carbon Steels Plain Carbon Free Cutting Screw Stock	1xxx 10xx 11xx	Molybdenum Steel Carbon-Molybdenum Chr.-Molybdenum Ni.-Molybdenum	4xxx 40xx 43xx 46xx
Nickel Steel 3.5% Ni 5% Ni	2xxx 23xx 25xx	Chromium Steels Low chromium Medium Chromium Corrosion and heat resist.	5xxx 51xx 52xx 51xxx
Nickel Chromium Steel 1.25% Ni 1.75% Ni 3.5% Ni	3xxx 31xx 32xx 33xx	Chromium-Vanadium Steels (1% Cr)	6xxx 61xx
		Silicon-Manganese Steels 2% Si	9xxx 92xx

Load Magnification on Joints
(Change in clamping force)

Effect of Prying:

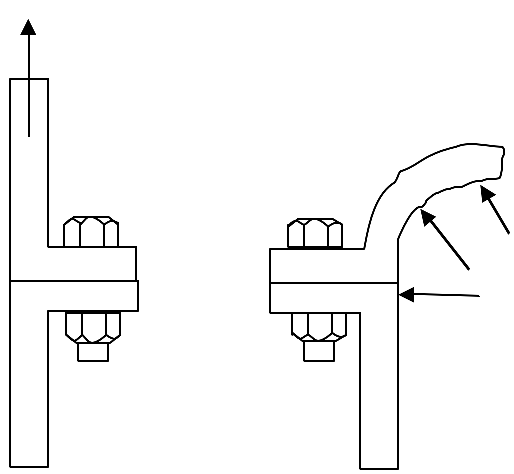
- Magnifies load (exceeds yield point)
- Applies bending load

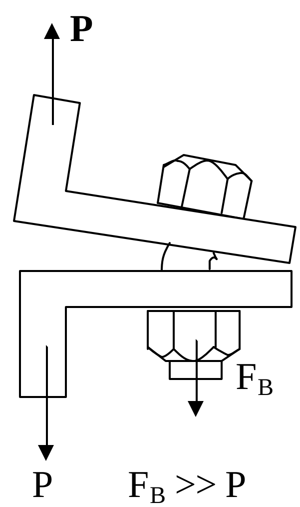


More prying when joint member is flexible.

Less prying when joint member is rigid

Prying Caused by Offset Loads





Fatigue Failure

A fastener under repeated cyclic tension loads can unexpectedly and suddenly fail – even though the loads applied are below the yield strength of material.

Failure under **repeated compression load is** rare and is ignored.

Necessary conditions for fatigue failure:

1. Cyclic tensile load
2. Stress level above a threshold value (Endurance limit)
3. A susceptible to fatigue material
4. An initial flaw in that material

The process and sequence of fatigue failure:

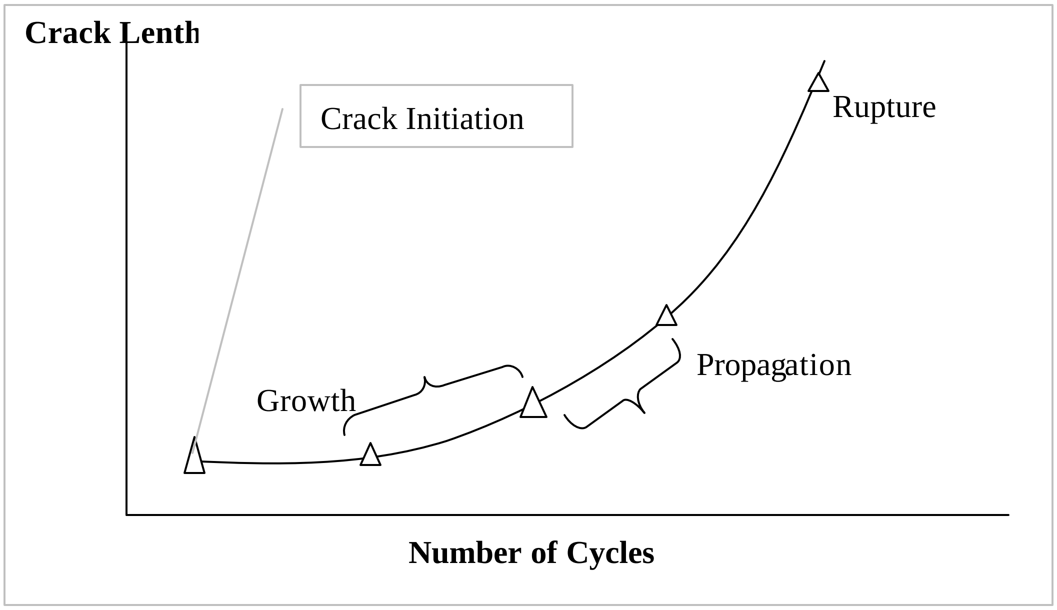
Crack initiation – generally initiated by a flaw.

- tool mark
- scratch produced by mishandling of part
- improper heat treatment
- corrosion
- inclusions in material

Crack growth – Small cracks create stress concentrations. Subsequently when the part is subjected to cyclic load, the stress concentration areas yield and tear the material.

Crack propagation – As the crack grows, the stress level increases, since there is less area to support the load. As stress level increases, the crack grows more rapidly.

Final rupture – The part fails at a point when crack has damaged the capability for the part to withstand additional load cycle. The failure occurs rapidly, which appear to be sudden to the user, as there are no visible signs of failure before this point in time.



Types of Fatigue Failure

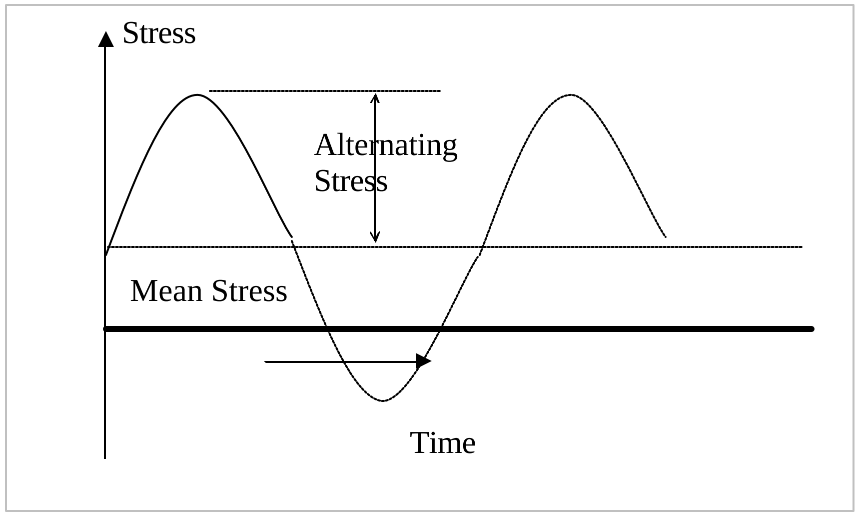
Low-cycle failure – failure occurring in any number from 1 – a few ten thousands cycle (Large load)

High-cycle failure – parts failing in cycles over hundreds of thousands or millions cycles (Smaller loads)

The number of cycles required to break a bolt depends on the magnitude of mean and alternating stress applied.

Low-cycle failure

- Very high alternating stress
- Low or none Mean stress

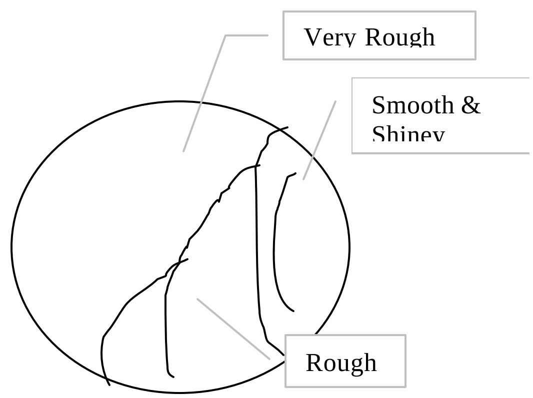


High-cycle failure

Smaller magnitude Stress

Most well designed bolts suffer high-cycle failure.

Appearance of Failure Surface



Failure during:

1. Crack initiation, growth,
   Slow failure Smooth & shiny
2. Crack propagation – rough surface
3. Final rupture – very rough

A close examination of a broken bolt can reveal the nature of failure as above. If the entire fastener fails suddenly during tightening, the entire broken surface will be rough.

Common Places of Fatigue Cracks in Bolts

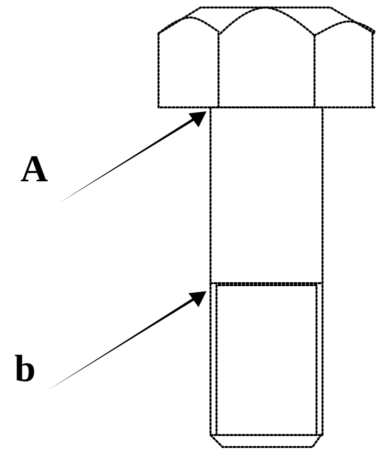
These are regions of high stress concentrations and locations of failure.

a. Head joins the shank

b. Thread run-out point

c. Location of change in diameter

d. First thread or two of engagement



What Determines Fatigue Life

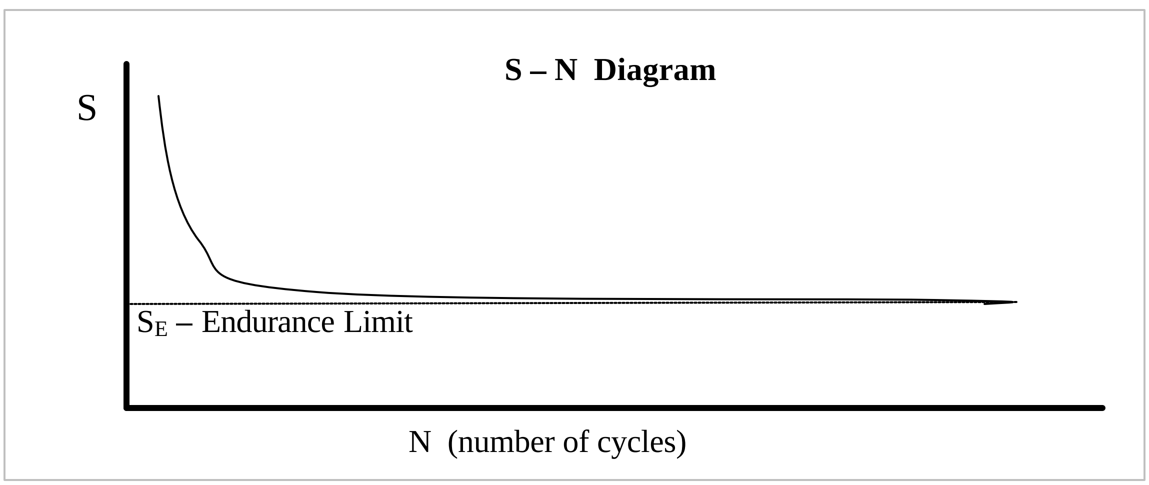
Magnitudes of (Mean stress + alternating stress).

1. Only mean stress (static loading, not a fatigue situation)
   - failure is determined by σ_u

2. Only alternating stress (fully reversed load, tension = compression)
   - failure can be predicted by S-N diagram for the material

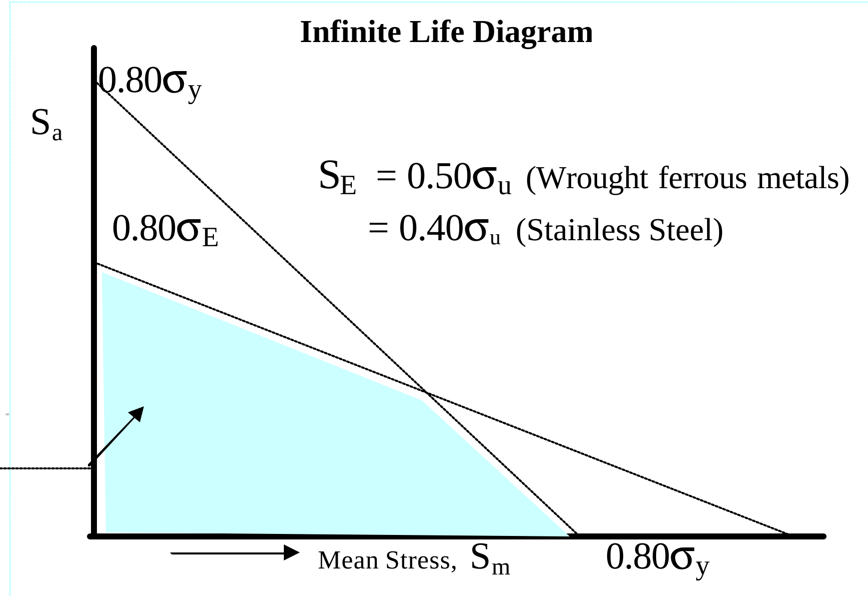
     S – Stress level. N – Number of cycles

     S_E – Endurance Limit (completely reversed stress limit Below which fatigue life is infinite).



3. Both mean stress and alternating stress present.

   The safe operating stress levels are determined by the shaded area in the Infinite Life Diagram.



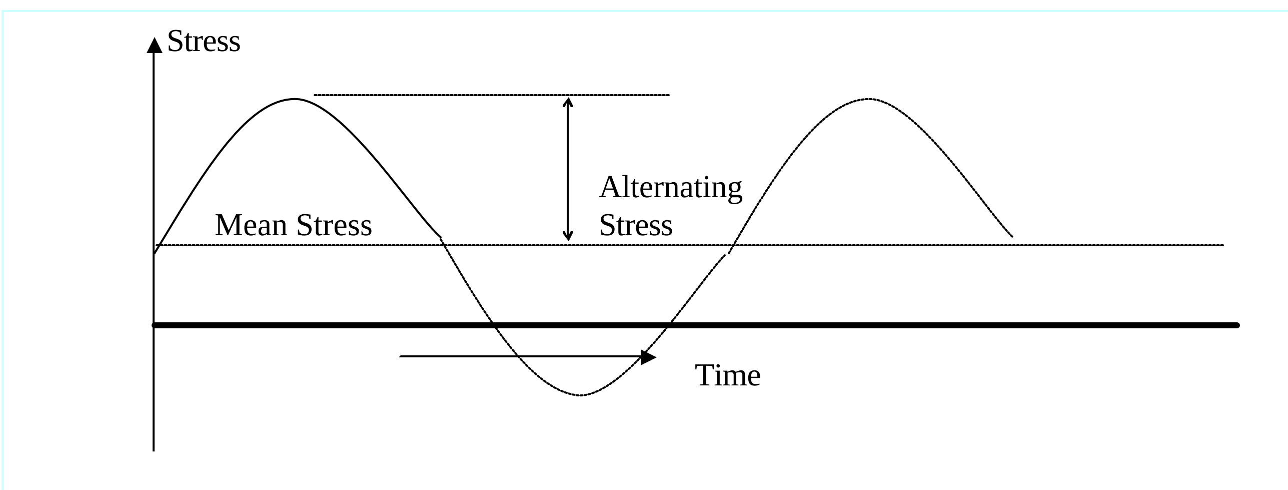
Safe operating range
Mean Stress → anywhere
Between $S_m = O \leftrightarrow 0.80\sigma_y$

Alternating Stress →

S_a $0 \leftrightarrow .80\sigma_E$ or S_E

S_E – Endurance limit of material
S_y or σ_y – Yield strength of material
S_u or σ_u - Ultimate strength of material

Any combination of S_a and S_m which is outside the shaded area is unsafe.



Bolt Load = Safe Stress x Area (bolt section)
↓
$(S_a + S_m)$

Refer to Design Handbook for more info.

Reducing Fatigue Problems

Among the common remedies is to minimize stress level or prevent it from occurring.

Rolled threads – instead of cutting them provides an unbroken flow of grains of the material reducing stress concentration due to notch effect.

Fillets – between head and shank of the bolt reduces stress concentration.

Perpendicularity – of the bottom surface of the bolt and the joint surface to the axis of bolt is important to avoid fatigue life being affected (2° error reduces fatigue life by 79%).

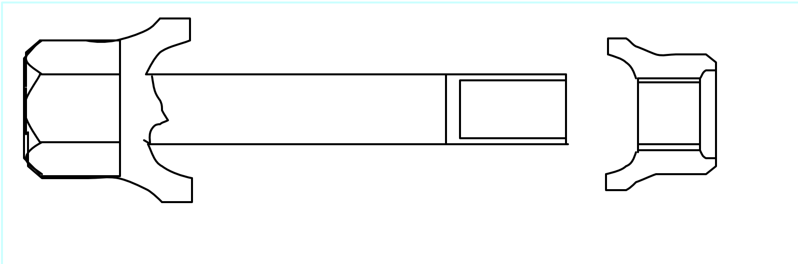
Overlapping Stress Concentration – due to multiple geometry can enhance fatigue failure. At bolt head to body connection, fillet radius and perpendicularity, described above, can severely affect fatigue.

Thread Stress Distribution – Can be reduced by making sure active thread engagement. The first three engaged threads share most of the bolt tension.

Bending – increases the stress level on one side of the fasteners. Use of spherical washer reduces this effect.

Corrosion – when reduced, diminishes the possibility of crack initiation or its growth and thus, increases fatigue life.

Flanged head and Nut – produces more uniform distribution of stress significantly affecting fatigue life.



Surface Condition – which reduces cracks can greatly improve fatigue life. Polished Surface, surfaces undergoing shot peening operation, etc. have fewer cracks.

<u>Prediction of Endurance Limit</u> (Fatigue Life)

$S_E^1 = S_E\left(C_1 x C_2 x C_3\right)$ $\qquad$ $S_E \cong .50\sigma_u$

C_1 = Load factor (.85 - .58)
C_2 = Stress bending/torsional factor (.85 - .5)
C_3 = Stress concentration factor (.3 for rolled threads)

DESIGN STRATEGY

1. Find load: - Steady - Alternating

2. Find Load/Bolt (fastener)

3. Select bolt material and obtain σ_y, σ_u etc.

   Say 1020 Hot-rolled, σ_u= 67,000, σ_y = 45,000

4. Find $S_E = ½\ \sigma_u$ = 33,500 psi

5. Calculate Working: $S'_E = C_1 \text{ x } C_2 \text{ x } C_3\ S_E$

   Conservative estimate = .75 x .75 x .5 x 33,500
   $\cong$ 9,400 psi

6. Plot Infinite Life Diagram using:
   σ_u, σ_y and $S_E{}'$ (don't forget .80σ_y etc. for plot) safety factor

7. Select working stress level and calculate BOLT DIAMETER
   [using standard Bolt diameter, recalculate item 6]

etc.

APPENDIX

Sources of Bolting Specifications

AISC	American Institute of Steel Construction, Inc. 400 North Michigan Avenue Chicago, Illinois 60611
AISI	American Iron and Steel Institute 1000 16th Street N.W. Washington, D.C. 20036
AMS	Obtain Aeronautical Material Specifications from: Society of Automotive Engineers 400 Commonwealth Drive Warrendale, Pennsylvania 15096
ASME	American Society of Mechanical Engineers United Engineering Center 345 E. 47th Street New York, N.Y. 10017
ANSI	American National Standards Institute 1430 Broadway New York, N.Y. 10018 (or from ASME)
ASTM	American Society for Testing and Materials 1916 Race Street Philadelphia, Pennsylvania 19103
BS	British Standards Institution Sales Branch Newton House 101/113 Pentonville Rd. London N1, England

The Fastener Quality Act: Insignia Recordals
(The Fastener Quality: http://www1.uspto.gov/web/offices/tac/fqa/fqa.htm)

The Fastener Quality Act (15 U.S.C. §§ 5401 et seq.) requires that certain fasteners bear an identifying insignia for traceability purposes, and requires the Secretary of Commerce to provide for the recordation of these insignia. The implementing regulations (15 C.F.R. §§280.700 et seq.) direct the Commissioner of Patents and Trademarks to maintain a Fastener Insignia Register.

Background
The Fastener Quality Act requires that certain fasteners sold in commerce conform to the specifications to which they are represented to be manufactured. Product inspection, testing, and certification are required, as well as record keeping and fastener insignia recordal. The National Institute of Standards and Technology (NIST) have responsibility for all aspects of the law except its enforcement and the mandatory insignia recordal requirement. The Patent and Trademark Office (PTO) has been tasked with establishing and maintaining a Fastener Insignia Register (FIR); the Bureau of Export Administration (BXA) will be the enforcement arm of FQA. All three agencies mentioned are sub-agencies within the purview of the U.S. Department of Commerce.

When Will the FQA be in Full Force?
On August 14, 1998, an amendment to the Fastener Quality Act was signed into law. (P.L 105-234). This amendment provides, among other things, that regulations issued under the Act will not take effect until after the later of June 1, 1999, or the expiration of a 120 day period after submission of a report by the Secretary of Commerce to various congressional committees. The amendment has been interpreted to provide that the implementation date of the FQA will be the date on which regulations issued under the Act take effect.

Who Must Record Their Insignia?
Fastener manufacturers and private label distributors, as defined in the Fastener Quality Act. 15
U.S.C. 5402.

What Kind of Fasteners Must Bear Insignias and Record these Insignias Under FQA?
The FQA insignia recordal requirements in Section 8 apply to a specific sub-set of fasteners, i.e., to those fasteners that are required by the manufacturing standards and specifications to bear a raised or depressed insignia that identifies its maker or distributor. 15 U.S.C. §5407. Certainly there are fasteners in which the standards and specifications do not require that they bear a source-identifying insignia, and there are cases in which it is impracticable for the fastener to bear an insignia. Please note that quality-identifying insignia are not recorded under FQA. If there are any questions regarding whether recordal is required, please contact NIST for verification.

Glossary of Fastener and Bolted Joint Terms

Angularity The underfaces of the nut and bolt head should be exactly perpendicular to the thread or shank axes. If the angle between the face and the axis is, for example, 86° or 94°, the fastener is said to have an angularity of 4° (sometimes called Perpendicularity).

Anode That electrode in a battery or corrosion cell which produces electrons. It is the electrode which is destroyed (corrodes).

Brittle A bolt is said to be brittle if it will break when stretched only a small amount past is yield point. (Compare Ductile.)

Cathode That electrode in a battery or corrosion cell which attracts electrons.

Clamping force The equal and opposite forces which exist at the interface between two joint members. The clamping force is created by tightening the bolts, but is not always equal to the combined tension in the bolts. Hole interference problems, for example, can create a difference between clamping force and bolt loads.

Constant life diagram A plot of experimentally derived fatigue –life data; perhaps the most complex and complete of the popular charts used to represent such data.

Corrosion cell A natural "battery" formed when two metals having different electrical potentials (an Anode and a Cathode) are connected together in the presence of a liquid (the Electrolyte).

Creep The slow, plastic deformation of a body under heavy loads. Time-dependent plasticity.

Ductile If a bolt can be stretched well past its yield point before breaking, it is said to be ductile. (See also Brittle.)

Eccentric load The external load on a fastener or groups of fasteners is said to be eccentric if the resultant of that load does not pass through the center of the group of fasteners (eccentric shear load) or does not coincide with the bolt axis (eccentric tensile load).

Effective length of a bolt The grip length plus some portion of the bolt (often one-half of the thickness of the nuts) which lies within the nut(s) plus some portion (often one-half thickness) of the head. Used in stiffness and stretch calculations.

Effective radius of not, or bolt head, or threads Distance between the geometric center of the part and the circle of points through which the resultant contact forces between mating parts passes. Must be determined by integration.

Electrode The two metallic bodies in a battery or Corrosion cell which give up electrons (the Anode) or which attract them (the Cathode).

Embedment Localized plastic deformation in heavily loaded fasteners allows one part to sink into, or smooth the surface of, a softer or more heavily loaded second part. Nuts embed themselves in joint faces. Bolt threads embed themselves in nut threads, etc.

Endurance limit That completely reversing stress limit below which bolt or joint member will have an essentially infinite life under cycle fatigue loads. Note that the mean stress on the bolts here is zero.

External load Forces exerted on fastener and/or joint members by such external factors as weight, wind, inertia, vibration, temperature expansion, pressure, etc. Does not equal the Working load in the fastener.

Failure of the bolt Failure of a bolted joint to behave as intended by the designer. Failure can be caused or accompanied by broken or lost bolts, but can also mean joint slip or leakage from a gasketed joint even if all bolts still remain whole and in place. Common reasons for joint failure include vibration loosening, poor assembly practices, improper design, unexpected service loads or conditions, etc.

Fillet Transition region between bolt head and shank, or between other changes in diameter. (See Bolt, parts of.)

Galling An extreme form of adhesive wear, in which large chunks of one part stick to the mating part (during sliding contact).

Galvanic protection The coating on a fastener is said to provide galvanic protection if it is more anodic than the fastener and will, therefore, be destroyed instead of the fastener. Zinc plate (galvanizing) provides galvanic protection to steel fasteners, for example.

Grip length Combined thickness of all the things clamped together by the bolt and nut, including washers, gaskets and joint members.

Hydrogen embrittlement A common and troublesome form of Stress cracking. Several theories have been proposed to explain hydrogen embrittlement, but, at present, the exact mechanism is still unknown. What is known, however, is the fact that if hydrogen is trapped in a bolt by poor electroplating practices, it can encourage stress cracking. Bolts can fail, suddenly and unexpectedly, under normal load.

Impact wrench An air- or electric-powered wrench in which multiple blows from tiny hammers are used to produce output torque to tighten fasteners.

Inclusions Small pieces of nonmetallic impurities trapped within the base metal or, for example, a bolt.

Infinite life diagram A simple plot experimentally derived fatigue life data, showing the conditions required for infinite life.

Initial preload The tension created in a single bolt as it is tightened. Will usually be modified by the subsequent assembly operations and/or by in-service loads and conditions.

Joint diagrams Mathematical diagrams which illustrate the forces on and deflections of fasteners and joint members.

Junker machine A test machine, first proposed by Gerhard Junker, for testing the vibration resistance of fasteners.

Lockbolt A fastener which bears a superficial resemblance to a bolt, but which engages a collar (instead of a nut) with annular grooves (instead of threads). The collar is swaged over the grooves on the male fastener to develop preload.

Lock nut A nut which provides extra resistance to vibration loosening (beyond that produced by proper Preload), either by providing some form of Prevailing torque, or, in free-spinning lock nuts, by deforming, cramping, and/or biting into mating parts when fully tightened.

Material velocity The velocity of sound in a body (e.g., a bolt). A term used in the ultrasonic measurement of bolt stress or strain.

Nominal diameter The "catalog diameter" of a fastener. Usually roughly equal to the diameter of the body, or the outer diameter of the threads.

Nut factor An experimental constant used to evaluate or describe the ratio between the torque applied to a fastener and the Preload achieved as a result.

Pitch The nominal distance between two adjacent thread roots or crests.

Preload The tension created in a threaded fastener when the nut is first tightened. Often used interchangeably, but incorrectly, the Working load or bolt force or bolt tension.

Prevailing torque Torque required to run a nut down against the joint when some obstruction, such as a plastic insert in the thread or a noncircular thread, or other, has been introduced to help the fastener resist vibration loosening. Prevailing torque, unlike "normal" torque on a nut or bolt, is not proportional to the Preload in the fastener.

Proof load The maximum, safe, static, tensile load which can be placed on a fastener without yielding it. Sometimes given as a force (in lb or N) sometimes as a stress (in psi or Mpa).
Relaxation The loss of tension, and therefore Clamping force, in a bolt and joint as a result of Embedment, vibration loosening, gasket creep, differential thermal expansion, etc.

Residual preload The tension which remains in an unloaded bolted joint after Relaxation.

Rolled thread A thread formed by plastically deforming the surface of the blank rather than by cutting operations. Increases fatigue life and thread strength, but is not possible (or perhaps economical) on larger sizes

Scatter Data points or calculations are said to be scattered when they are not all the same. A "lot of scatter in preload" means wide variation in the preloads found in individual bolts.

Screw Threaded fastener designed to be used in a tapped or untapped (e.g., wood screw) hole, but not with a nut.

Self-loosening The process by which a supposedly tightened fastener becomes loose, as a result of vibration, thermal cycles, shock, or anything else which cause transverse slip between joint members and/or male and female threads. Vibration loosening is a common, but special, case of self-loosening.

Shank That portion of a bolt which lies under the head.

Spring constant The ratio between the force exerted on a spring (or a bolt) and the deflection thereof. Has the dimensions of force per unit change in length (e.g., lb/in.). Also called Stiffness.

Standard deviation A statistical term used to quantify the Scatter in a set of data points. If the standard deviation is small, most of the data points are "nearly equal". A large deviation means less agreement.

Strength of bolt An ambiguous term which can mean Ultimate strength or Proof load or Endurance limit or Yield strength.

Stress area The effective cross-sectional area of the threaded section of a fastener. Used to compute average stress levels in that section. Based on the mean of pitch and minor diameters.

Stress corrosion cracking (SCC) A common form of Stress cracking in which an Electrolyte encourages the growth of a crack in a high stressed bolt. Only a tiny quantity of electrolyte need be present at the tip or face of the crack.

Stress cracking A family of failure modes, each of which involves high stress and chemical action. The family includes Hydrogen embrittlement, Stress Corrosion Cracking, stress embrittlement, and hydrogen-assisted stress corrosion.

Stress factor A calibration constant used in ultrasonic measurement of bolt stress or strain. It is the ratio between the change in ultrasonic transit time caused by the change in length of the fastener under load, to the total change in transit time (which is also affected by change in the stress level).

Stress relaxation The slow decrease in stress level within a part (e.g., a bolt) which is heavily loaded under constant deflection conditions. A "cousin" to creep, which is a slow change in geometry under constant stress conditions.

Temperature factor A calibration constant used in ultrasonic measurement of bolt stress or strain. Accounts for the effects of thermal expansion and the temperature-induced change in the velocity of sound.

Tension, bolt Tension (tensile stress) created in the bolt by assembly preloads and/or such things as thermal expansion, service loads, etc.

Tensioner A hydraulic tool used to tighten a fastener by stretching it rather than by applying a substantial torque to the nut. After tensioner has stretched the bolt or stud, the nut is run down against the joint with a modest torque, ant the tensioner is disengaged from the fastener. The nut holds the stretch produced by the tensioner.

Thread form The cross-sectional shape of the threads, defining thread angle, root, and crest profiles, etc.

Thread length Length of that portion of the fastener which contains threads cut or rolled to full depth. (See <u>Dimensions of bolt</u>.)

Thread run-out That portion of the threads which are not cut or rolled full depth, but which provide the transition between full-depth threads and the body or head. (See <u>Bolts, parts of</u>.) Officially called thread washout or vanish, although the term run-out is more popular. (Run-out is officially reserved for rotational eccentricity, as defined by total indicator readings or the like.)

Threaded fastener Studs, bolts, and screws of all orts, with associated nuts. One of the most interesting, complex, useful – and frustrating—components yet devised.

Tightness A measure of the mass leak rate from a gasketed joint.

Tightness parameter A dimensionless parameter which defines the mass leakage of a gasket as a function of contained pressure and a contained fluid constant.

Torque The twisting moment, product of force and wrench length applied to a nut or bolt (for example).

Torque monitor A torque tool control system which monitors the amount of torque being developed by the tool during use, but does not control the tool or the torque produced.

Torque multiplier A gearbox used to multiply the torque produced by a small hand wrench (usually a <u>Torque wrench</u>). The output of the multiplier drives the nut or bolt with a torque that is higher, and a speed that is lower, than input torque and speed. There is no torque gauge or read-out on the multiplier.

Torque wrench A manual wrench which incorporates a gauge or measuring apparatus of some sort to measure and display the amount of torque being delivered to the nut or bolt. All wrenches produce torque. Only a torque wrench tells how much torque.

Transducer A device which converts one form of energy into another. An ultrasonic transducer, for example, converts electrical energy into acoustic energy (at ultrasonic frequencies) and vice versa.

Turn-of-nut Sometimes used to describe the general rotation of the nut (or bolt head) as the fastener is tightened. More often used to define a particular tightening procedure in which a fastener is first tightened with a preselected torque, and is then tightened further by giving the nut an additional, measured, turn such as "three flange" (180°).

Ultimate strength The maximum tensile strength a bolt or material support prior to rupture. Always found in the plastic region of the stress-strain or force-elongation curve, and so is not a design strength. Also called Tensile strength and ultimate tensile strength.

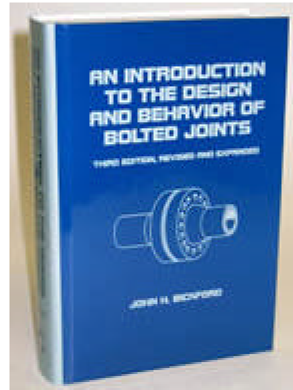
Ultrasonic extensometer An electronic instrument which measures the change in length of a fastener ultrasonically as, or before and after, the fastener is tightened.

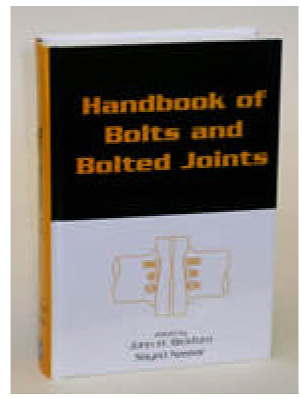
Work hardening The slight increase in hardness and strength produced when a body is loaded past its yield point. Also called strain hardening.

Working load The tension in a bolt in use; tension produced by a combination of Residual preload and a portion (usually) of any External load. The Joint diagram is usually used to predict the approximate working load a fastener will see in service.

Yield strength That stress level which will create a permanent deformation of 0.2% or 0.5% or some other small, preselected, amount in a body. Approximately equal to the elastic and proportional limits of the material; a little higher than the proof strength of a bolt. (See Proof load.)

References





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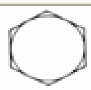
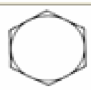
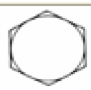
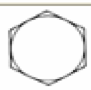
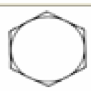
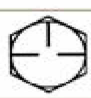
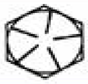
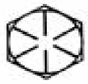
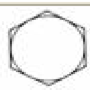
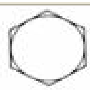
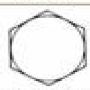
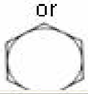
Fastener Material Property Comparison (Typical inch and metric fasteners)

Inch Grade	Marks on Head	Material	Tensile Strength		Yield Strength	
			N/mm^2	psi	N/mm^2	psi
2	none	Steel	510	74,000	393	57,000
5	3	Steel	827	120,000	634	92,000
8	6	Alloy Steel	1030	150,000	896	130,000
SHCS	none	Alloy Steel	1240	180,000	965	140,000
18-8	none	302 Stainless	690	100,000	448	65,000
316	none	316 Stainless	690	100,000	448	65,000
Metric Class	**Marks on Head**	**Material**	**Tensile Strength**		**Yield Strength**	
			N/mm^2	psi	N/mm^2	psi
8.8	8.8	Steel	800	116,000	640	93,000
10.9	10.9	Steel	1040	151,000	940	136,000
12.9	12.9	Alloy Steel	1220	177,000	1100	160,000
A2-70	A2-70	302 Stainless	700	102,000	450	65,000
A4-80	A4-80	316 Stainless	800	116,000	600	87,000

	Ultimate Strength			Yield Strength[3]						
	Density Kg / m^3	Tension, MPa	Compression[2], MPa	Shear, MPa	Tension, MPa	Shear, MPa	Modulus of Elasticity, GPa	Modulus of Rigidity, GPa	Coefficient of Thermal Expansion, 10^6/°c	Ductility, Percent Elongation 50mm
Material										
Steel:										
Structural	7860	400	...	...	260	145	200	79	11.7	23
High-strength-low-allow	7860	480	...	...	350	210	200	79	11.7	21
Quenched and tempered alloy	7860	825	...	...	700	380	200	79	11.7	18
Stainless										
Cold-rolled	7920	860	...	...	500	...	190	73	17.3	12
Annealed	7920	620	...	...	275	152	190	73	17.3	50

ASTM, SAE AND ISO GRADE MARKINGS AND MECHANICAL PROPERTIES FOR STEEL FASTENERS

Identification Grade Mark	Specification	Fastener Description	Material	Nominal Size Range (in.)	Mechanical Properties: Proof Load (psi)	Mechanical Properties: Yield Strength Min (psi)	Mechanical Properties: Tensile Strength Min (psi)
 No Grade Mark	SAE J429 Grade 1	Bolts, Screws, Studs	Low or Medium Carbon Steel	1/4 thru 1-1/2	33,000	36,000	60,000
	ASTM A307 Grades A&B		Low Carbon Steel	1/4 thru 4	--	--	
	SAE J429 Grade 2		Low or Medium Carbon Steel	1/4 thru 3/4 Over 3/4 to 1-1/2	55,000 33,000	57,000 36,000	74,000 60,000
 No Grade Mark	SAE J429 Grade 4	Studs	Medium Carbon Cold Drawn Steel	1/4 thru 1-1/2	--	100,000	115,000
 B5	ASTM A193 Grade B5		AISI 501	1/4 Thru 4	--	80,000	100,000
 B6	ASTM A193 Grade B6		AISI 410			85,000	110,000
 B7	ASTM A193 Grade B7		AISI 4140, 4142, OR 4105	1/4 thru 2-1/2 Over 2-1/2 thru 4 Over 4 thru 7	-- -- --	105,000 95,000 75,000	125,000 115,000 100,000
 B16	ASTM A193 Grade B16		CrMoVa Alloy Steel			105,000 95,000 85,000	125,000 115,000 100,000
	SAE J429 Grade 5	Bolts, Screws, Studs	Medium Carbon Steel, Quenched and Tempered	1/4 thru 1 Over 1 to 1-1/2	85,000 74,000	92,000 81,000	120,000 105,000
	ASTM A449			1/4 thru 1 Over 1 to 1-1/2 Over 1-1/2 thru 3	85,000 74,000 55,000	92,000 81,000 58,000	120,000 105,000 90,000
	SAE J429 Grade 5.1	Sems	Low or Medium Carbon Steel, Quenched and Tempered	No. 6 thru 3/8	85,000	--	120,000
	SAE J429 Grade 5.2	Bolts, Screws, Studs	Low Carbon Martensitic Steel, Quenched and Tempered	1/4 thru 1	85,000	92,000	120,000
 A325	ASTM A325 Type 1	High Strength Structural Bolts	Medium Carbon Steel, Quenched and Tempered	1/2 thru 1 1-1/8 thru 1-1/2	85,000 74,000	92,000 81,000	120,000 105,000
 A325	ASTM A325 Type 2		Low Carbon Martensitic Steel, Quenched and Tempered	1/2 thru 1	85,000	92,000	120,000
 A325	ASTM A325 Type 3		Atmospheric Corrosion Resisting Steel, Quenched and Tempered	1/2 thru 1 1-1/8 thru 1-1/2	85,000 74,000	92,000 81,000	120,000 105,000

	SAE J429 Grade 7	Bolts, Screws,	Medium Carbon Alloy Steel, Quenched and Tempered ⋆	1/4 thru 1-1/2	105,000	115,000	133,000
	SAE J429 Grade 8	Bolts, Screws, Studs	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 1-1/2	120,000	130,000	150,000
	ASTM A354 Grade BD		Alloy Steel, Quenched and Tempered ⋆				
 No Grade Mark	SAE J429 Grade 8.1	Studs	Medium Carbon Alloy or SAE 1041 Modified Elevated Temperature Drawn Steel	1/4 thru 1-1/2	120,000	130,000	150,000
 A490	ASTM A490	High Strength Structural Bolts	Alloy Steel, Quenched and Tempered	1/2 thru 1-1/2	120,000	130,000	150,000 min 170,000 max
 No Grade Mark	ISO R898 Class 4.6	Bolts, Screws, Studs	Medium Carbon Steel, Quenched and Tempered	All Sizes thru 1-1/2	33,000	36,000	60,000
 No Grade Mark	ISO R898 Class 5.8				55,000	57,000	74,000
8.8  or 	ISO R898 Class 8.8				85,000	92,000	120,000

Notes: (Blank Page)

Program Evaluations

Program Title_______________________________________ ____
Program completion date

Instructor Ranjit K. Roy & …. Training Location/host.______________________________________

We appreciate your comments and suggestions. Please take a moment to let us know how we can improve and serve you better. (Please use the following numbers for evaluation purposes).

[6] Excellent [5] Very Good [4] Satisfactory [3] Poor [2] Unsatisfactory [1] Needs Improvement

1. Overall Reaction to Program: 6 5 4 3 2 1 (Circle one number)

Comments:____

____

2. Reaction to Instructor: 6 5 4 3 2 1 (Circle one number)

Comments:____

____

3. What should be added and/or deleted to improve this program? (Please be specific)

____

4. What portions of this program do you feel will be most helpful at your work? How?

____

____

5. Do you feel the training provided you enough understanding of the technique for you to be able to start applying it to your own projects if opportunities were available?
[ ] Yes [ ] No [ ] May be

6. Would you consider/recommend us for training at your facility? [ ] yes [ ] No [ ] May be

Who should we contact for training at your facility ___________________________ Ph: ____________________

7. Would you recommend that others attend this seminar? [ ]YES [ ]NO

8. Portion of class sessions you attended [ ] 100% [ ] Over 90% [ ] over 75% [ ] Over 50%

9. Percent of class problems and assignment you completed and/or took active part: [ ] 100% [ ] Over 80% [ ] 50%

(Optional)
Your Name________________________________ Your Ph#/ & E-mail:___________________________________

THANK YOU for taking the time to complete this evaluation.