Bolted Joint Design

There is no one fastener material that is right for every environment. Selecting the right fastener material from the vast array of those available can be a daunting task. Careful consideration must be given to strength, temperature, corrosion, vibration, fatigue, and many other variables. However, with some basic knowledge and understanding, a well thought out evaluation can be made.

Mechanical Properties of Steel Fasteners in Service

Most fastener applications are designed to support or transmit some form of externally applied load. If the strength of the fastener is the only concern, there is usually no need to look beyond carbon steel. Considering the cost of raw materials, non-ferrous metals should be considered only when a special application is required.

Tensile strength is the mechanical property most widely associated with standard threaded fasteners. Tensile strength is the maximum tension-applied load the fastener can support prior to fracture.

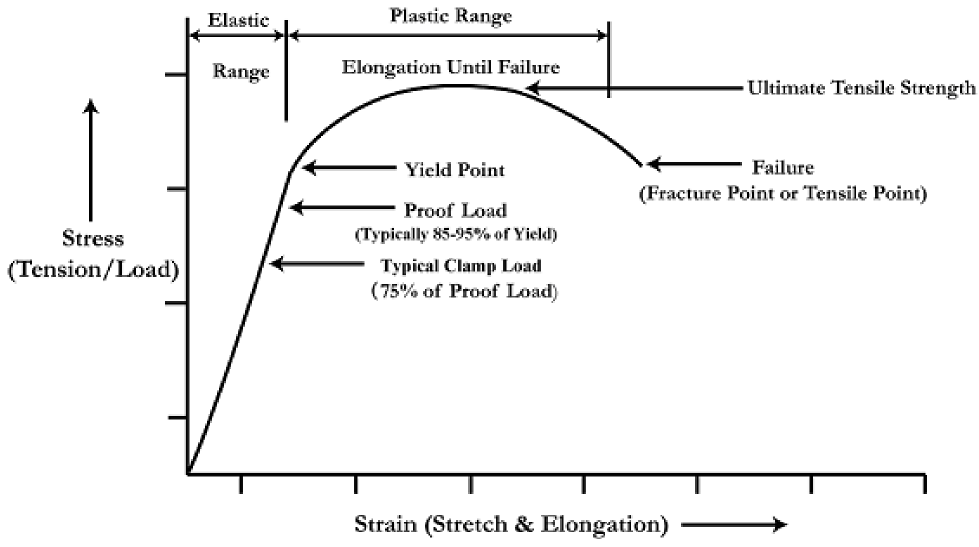
The tensile load a fastener can withstand is determined by the formula $P = St \times As$.

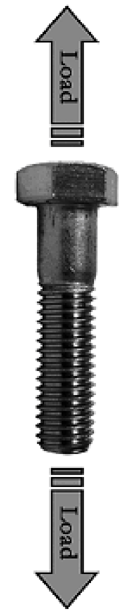
- P= Tensile load– a direct measurement of **clamp load** (lbs., N)
- St= Tensile strength– a generic measurement of the material's strength (psi, MPa).
- As= Tensile stress area for fastener or area of material (in^2, mm^2)

P* = *St* x *As		
P = tensile load (lbs., N)	***St*** = tensile strength (psi, MPa)	***As*** = tensile stress area (sq. in, sq. mm)
Applied to a 3/4-10 x 7" SAE J429 Grade 5 HCS		
P = ?	***St*** = 120,000 psi	***As*** = 0.3340 sq. in
P = 120,000 psi **x** 0.3340 sq. in		
P = 40,080 lbs.		

To find the tensile strength of a particular bolt, you will need to refer to Mechanical Properties of Externally Threaded Fasteners chart in the Fastenal Technical Reference Guide. To find the tensile stress area, refer to the Thread Stress Areas chart also in the Guide.

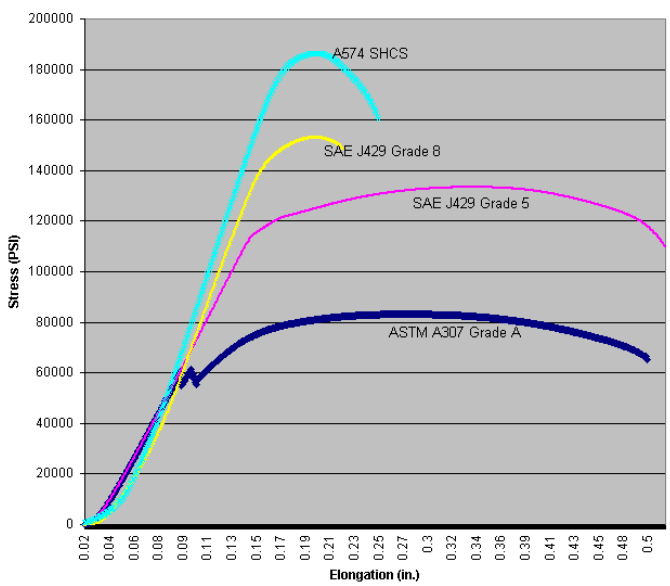
For this relationship, significant consideration must be given to the definition of the tensile stress area, *As*. When a standard threaded fastener fails in pure tension, it typically fractures through the threaded portion (as this is characteristically its smallest and therefore weakest area). For this reason, the tensile stress area is calculated through an empirical formula involving the nominal diameter of the fastener and the **thread pitch**.





Tensile Stress-Strain Diagram

As the fastener approaches the maximum strength of the threaded portion, it will permanently deform. To avoid this risk, most carbon or alloy steel bolts have a defined **proof load**, which represents the usable strength range for that particular fastener. By definition, the proof load is an applied tensile load that the fastener must support without permanent deformation. In other words, the bolt returns to its original shape once the load is removed.

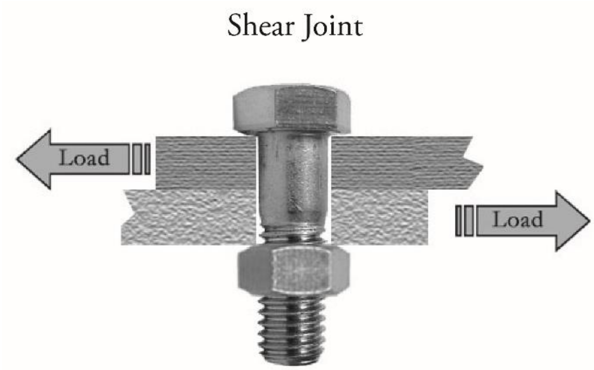


The relationship between tension and bolt stretch can be observed on a Tensile Stress-Strain Diagram. To the left is the stress-elongation curve. Steel possesses a certain amount of elasticity as it is stretched. Thus, a bolt that is properly tensioned should be functioning in the **elastic range** (as viewed on the Diagram). If the load is removed and the fastener is still within the elastic range, the fastener will always return to its original shape.

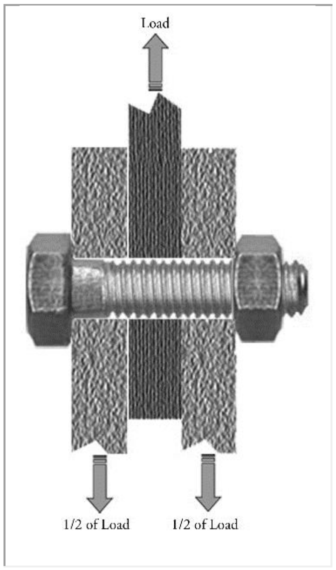
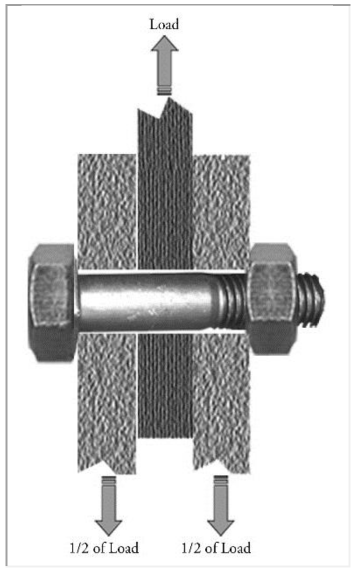
However, if the load applied causes the fastener to exceed its **yield point**, it enters the **plastic range**. At this point, the steel is no longer able to return to its original shape if the load is removed. The **yield strength** is the point at which a specified amount of permanent deformation occurs. If we would continue to apply a load, we would reach a point of maximum stress known as the ultimate tensile strength. Past this point, the fastener continues to **neck down** and elongate further with a reduction in stress. Additional stretching will ultimately cause the fastener to break at the tensile point.

Harder, higher tensile strength fasteners, such as the A574 tend to be less ductile than the softer lower strength fasteners. Although they have higher tensile strength, the overall length of the strain curve is often decreased.

Shear strength is defined as the maximum load that can be supported prior to fracture, when applied at a right angle to the fastener's axis. A load occurring in one transverse plane is known as single shear. **Double shear** is a load applied in two planes where the fastener could be cut into three pieces.



For most standard threaded fasteners, shear strength is not specified even though the fastener may be commonly used in shear applications. While shear testing of blind rivets is a well-standardized procedure that calls for a single shear test fixture, the shear testing technique for threaded fasteners is not as well designed. Most procedures use a double shear fixture, but variations in the test fixture designs cause a wide scatter in measured shear strengths (i.e., the variations in test procedures produce non-standard results).

Double Shear Through Threads (½-13 SAE J429 Grade 8)	**Double Shear Through Body** (½-13 SAE J429 Grade 8)
	
½-13 Thread Root Area: 0.126 sq-in	Minimum Body Area 0.191 sq-in
60% of Tensile Strength: 90,000 PSI	60 % of Tensile Strength: 90,000 PSI
Double Shear = 2 x 0.126 sq-in x 90,000 PSI	Double Shear = 2 x 0.191 sq-in x 90,000 PSI
Double Shear = **22,680 lbs.**	Double Shear = **34,380 lbs.**

To determine the shear strength of the fastener, the total cross-sectional area of the **shear plane** is important. For shear planes through the threads, we could use the thread root area. There are two possibilities for applied shear load (as illustrated below). One possibility is that the shear plane occurs in the threaded portion of the bolt. Since shear strength is directly related to the net sectional area (i.e.: the amount solid bolt material in the diameter), a narrower area will result in lower bolt shear strength. To take full advantage of strength properties the shank of the bolt body should be within the shear planes. To illustrate, consider the difference in shear strength between the two Grade 8 bolts on the previous page; one with the threads in the shear plane, the other with the shank in the shear plane.

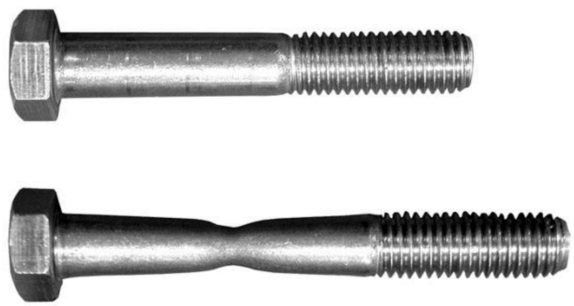
When no shear strength is given for common carbon steels with hardness up to 40 HRC, 60 % of the ultimate tensile strength of the bolt is typically used as acceptable shear strength. Note: the shear strength must fall within the constraints of a suitable safety factor. This formula should only be used as an estimation.

A fastener subjected to repeated cyclic loads can break suddenly and unexpectedly, even if the loads are well below the strength of the material. The fastener fails in fatigue. **Fatigue strength** is the maximum stress a fastener can withstand for a specified number of repeated cycles prior to its failure.

Torsional strength is a load usually expressed in terms of torque, at which the fastener fails by being twisted off about its axis. Self-tapping screws and socket set screws require a torsional test to ensure that the screw head can withstand the required tightening torque.

Other Mechanical Properties

Hardness is a measure of a material's ability to resist abrasion and indentation. For carbon steels, **Brinell** and **Rockwell** hardness testing can be used to estimate tensile strength properties of the fastener (occasionally **Vickers**). For more information about these hardness tests and their corresponding scales (e.g.: HRC, HRB, etc.) see the glossary.



Stainless steel is an example of a very ductile metal. When placed under enough stress, it will elongate significantly before it fractures.

Ductility is the ability of a material to deform before it fractures. A material that experiences very little or no plastic deformation upon fracture is considered brittle (e.g.: SHCS). A reasonable indication of a fastener's ductility is the ratio of its specified minimum yield strength to the minimum tensile strength. The lower this ratio the more ductile the fastener will be.

Toughness is a materials ability to absorb impact or shock loading. Impact strength toughness is rarely a specification requirement. Besides various aerospace industry fasteners, ASTM A320 *Specification for Alloy Steel Bolting Materials for Low-Temperature Service* is one of the few specifications that require impact testing on certain grades.

Joint Design

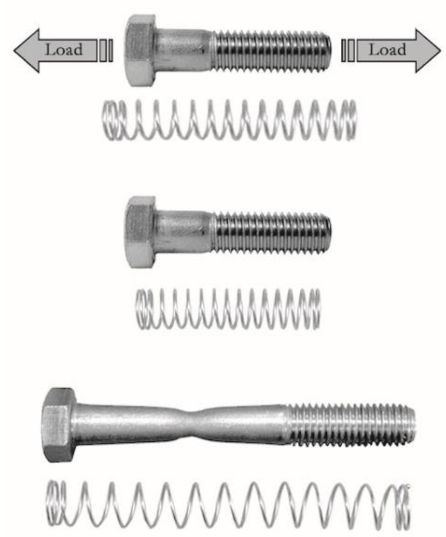
Loads can be applied to bolted joints in a number of different ways, each of which produces unique effects on the joint. These effects result from the way the joint is loaded, as well as how the joint responds to the load. Some of the various load types include tensile, shear and bending. The type of bolted joint derives its name from the external load applied to the joint.

A **tension joint**, as illustrated in the photo, is affected by loads that try to pull the joint apart. The forces on the joint and those on the bolts are roughly parallel to the axes of the bolts. All tensile forces try to stretch and/or separate the joint. The tension load, no matter how small, will add to the stress in the bolt and/or partially relieve the joint.

The bolts in a tension joint must act like clamps. The tightening of the bolt and nut produces a tensile pre-stress, which is approximately equal to the compressive stress introduced in the joint material. The behavior and life of the joint depends on how tightly the bolts clamp and how long they can maintain their preload.

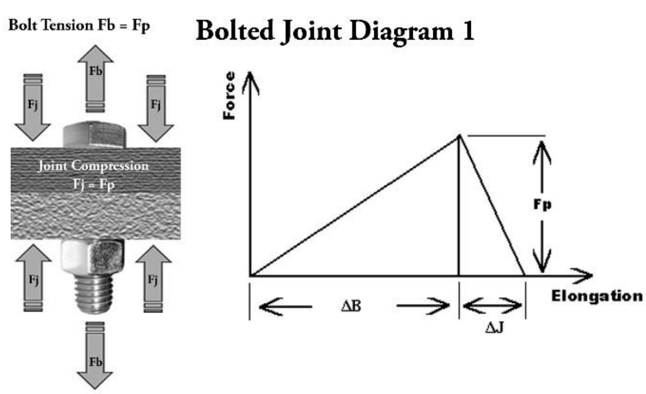
Proper amount of tensioning of the bolts is vital. With too little clamping force, the joint may loosen. If the joint is exposed to cyclical loads, too little clamping force can shorten the bolt's fatigue life. Too much clamping force can also cause severe problems. By over-tightening the bolt, one may exceed the proof load of the bolt. Even if the bolt does not fail during assembly, it may later break under the external tensile load. Over-tightening of the bolt can also encourage the advancement of hydrogen embrittlement or stress corrosion cracking. The joint members can also be damaged or warp from too much clamp force.

The clamping force created during tightening stretches the bolt similar to a spring. A similar analogy can be made for the joined materials, except they are compressed like a spring during assembly. These "springs" exert a clamping force that will remain only as long as the bolts are stretched. Any applied service load or condition, which relaxes the bolt or reduces the clamping force, will release some of the spring's energy (i.e.: clamp force within the joint). This will increase the chances that the joint may loosen or that the bolts may fail.



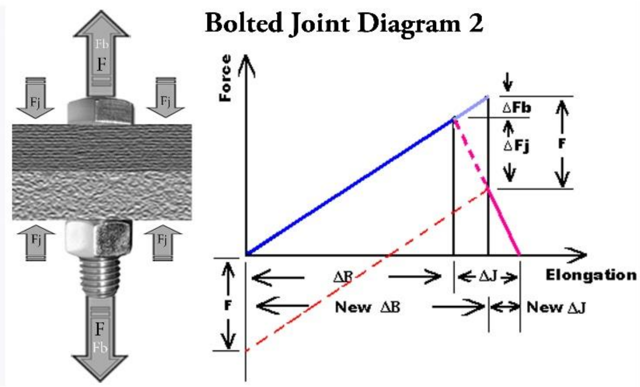
A bolt under tension functions similarly to a spring. When a spring is pulled by its ends, it stretches, and when tension is released it returns to its origional shape. However, if the spring is stretched too far, it will remain elongated.

A joint diagram may help illustrate what happens as we apply our preload and the effects of external loads. In "Bolted Joint Diagram 1", as the bolt is tightened, the bolt elongates (ΔB). Due to the internal forces resisting the elongation, a tension force or preload is produced (Fp). Notice the constant slope or straight-line relationship between the force and elongation. Remember that the stress-strain curve (which is basically applied force-elongation) will be constant or straight until the fastener begins to yield (elastic region).



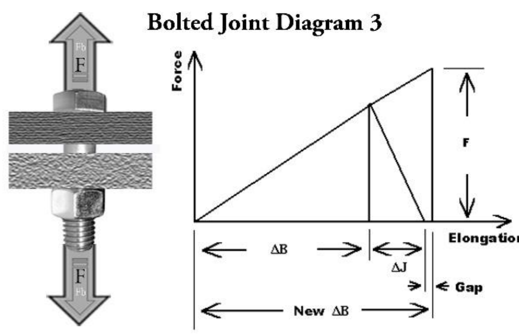
The reaction force (the right hand side of the graph) is the **clamp load** of the joint being compressed. ΔJ represents the amount that the joint has compressed. As is illustrated, ΔJ is smaller than ΔB. These values represent the stiffness of each component (joint and bolt). Often a bolt will only be about 1/3 to 1/5 as stiff as the joint that it is being used in. In this instance, our bolt tension is equivalent to the preload, which is equivalent to the joint compression. Or, to phrase it another way, the tension force on the bolt (Fb) is equal to the compression force on the joint (Fj), which is equivalent to the preload (Fp). This will change with the application of an external load. Bolted Joint Diagram 1 is illustrating elastic bolt elongation and elastic joint compression in the axial direction.

In "Bolted Joint Diagram 2", an external tensile force (F) has been applied to the joint under the bolt head and nut. The addition of this force has reduced some of the clamp force on the joint (ΔFj) and applied an additional force on the bolt (ΔFb). Since the bolt and joint have a different stiffness, ΔFb will not be the same as ΔFj. Also, the bolt will further elongate (new ΔB), and the joint compression will be reduced (new ΔJ). The increase in length is equal to the increase in thickness of the joint.



Bolted Joint Diagram 2

If the applied load (F) is allowed to increase, the clamp force acting on the joint will continue to decrease until the joint is fully unloaded, as can be seen in "Bolted Joint Diagram 3" (ΔJ = 0). Any further increase in the applied force will result in a gap between the plates and the bolt sustaining all of the additional force. In this case, the bolt or bolts are almost always subjected to non-linear loadings from bending and shear forces. This quickly leads to bolt failure.

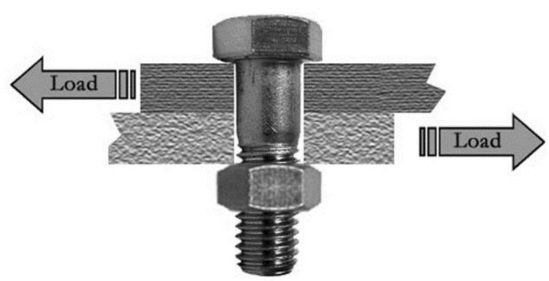


Bolted Joint Diagram 3

This is only a fraction of the possible scenarios that can be examined through joint diagrams. There are a number of other "real-life" factors, which may be impossible to predict, that allow the spring energy to be lost in the assembled joint. These factors include (but are not limited to) different points of loading, creep, alternating external loads, stress relaxation (in some instances a relaxation of 10% to 20% is common), temperature, differential thermal expansion, and vibration.

The overriding concern with the tension joint is its reliance on bolt tension or **preload**. If the clamping force is not correct, the joint can fail in several ways; either by bolt fatigue, vibration loosening, stress corrosion cracking or hydrogen embrittlement.

A **shear joint** is one in which the applied loading is at right angles to the fastener axis; that is, across the bolt shank. Shear joint failure occurs when the joint members are slipped sideways past each other, and eventually cut the fastener.



With some shear joints; the ultimate joint strength depends only upon the shear strength of the bolts. This type of joint is referred to as a "bearing type" joint. The amount of tension created in the bolts during assembly is relatively unimportant as long as the fastener is retained in the assembly. The joint member is allowed to slip until the fasteners come into "bearing" and prevents further slip. The fastener in this assembly is basically used as a pin.

Other types of shear joints rely on initial clamp load to resist slip. This type of joint requires a frictional force between the joint members. The shear forces have to overcome the friction developed by the clamp load, which in most cases will be far more than the actual "shear strength" of the fastener itself. This type of joint is common in the structural steel construction industry and may be referred to as a **friction-type** or **slip-critical joint**.

Unfortunately, many joints are rarely loaded in pure shear or tension. Some applications subject the joint to a **bending force**, which results in a combination of tension, and shear load acting simultaneously on the fastener. Extreme caution must be taken when working with a joint subjected to bending.

The combination of tension and shear loads can substantially impact the strength and behavior of a joint.

Usually designers would like to employ the highest clamping force the parts can withstand in order to compensate for some of the anticipated losses in preload. However, there are several limitations to the assembly preload. Too much force on the joint may damage joint members and gaskets or encourage stress cracking.

If more preload is not a possibility, there are other options such as altering the stiffness ratios between the bolt and the joint, or using similar materials for bolt and joint members.

High Temperature Effects

Most fastener materials are temperature sensitive; that is to say, their properties are influenced by a change in temperature. The strength of a metal fastener declines as temperature increases. At significant temperatures coatings may breakdown, high temperature corrosion, **creep**, or **stress relaxation** may occur, and differential thermal expansion rates between the fastener and the joint may cause failure.


The thermal expansion of materials and their ability to maintain suitable strengths is a crucial factor in the design of joints that will be exposed to extreme temperature environments.

While the strength of most fasteners will decrease as temperatures rise, any type of plating or coating will also alter working temperatures. For example, zinc plated fasteners are usually not recommended to be used above a temperature of 250°F.

A good example of temperature effects on bolts is Grade 8.2 bolts. At room temperatures, Grade 8.2 and Grade 8 bolts have similar properties. But, Grade 8.2 bolts are made of low carbon boron steel, whereas the Grade 8 fastener is a medium carbon alloy steel. Boron steel has a lower tempering temperature (minimum 650°F compared to the 800°F for the Grade 8). Thus, in a high temperature environment Grade 8.2 cannot be substituted for Grade 8. If the Grade 8.2 is used in a high temperature environment, it will cause relaxation and a decrease in strength as the higher temperature causes the fastener to anneal.

Every bolting material has a temperature above which it is unsafe to use. Often times this is referred to as the high temperature service limit. Although the fastener loses strength as the temperature increases, the service limit is usually determined by an occurrence known as **stress relaxation**.

When a significant amount of stress is placed on a bolt and it is then exposed to a high temperature the bolt begins to relieve itself of some of the stress and ultimately, reduces the clamping force (i.e., **preload**). Since the stress and the preload are related, this implies that the clamping force with which the bolt holds the joint together will be significantly reduced.

Thermal Expansion is one of the most problematic temperature effects. As the temperature rises, heat causes all bolt and joint materials expand, but not all at the same rate.

As an example consider the use of aluminum in conjunction with carbon steel fasteners. The aluminum will expand about twice as much as some carbon steel fasteners. If using a SAE J429 Grade 8 fastener to clamp an aluminum joint, we would expect to see a significant increase in the tension on the bolts as the temperature increase, which would also increase the clamping force. This reaction could damage the joint or gasket material, or even break the bolt. In the event that the bolt material would expand more than the joint, clamping force would be lost.

Differential expansion problems could occur even if the fastener and the joint are made of the same material. If the bolt and the joint heat up at different rates, the corresponding thermal expansion will also cause the bolt and the joint to expand at different rates.

Among the various other temperature related effects, two of the more common problems are creep and stress relaxation.

If a constant load is applied to a fastener and the service temperature is increased, the temperature places the bolt in its **creep range**; the bolt will begin to stretch even if the load is well within the fastener's mechanical limits. Eventually, the bolt may stretch to a point where it cannot support the load and will fail.

Stress relaxation is very similar, but specifically refers to the steady loss of stress in a loaded part with fixed dimensions.

Tensioning Methods

Threaded fasteners can clamp materials together only when they are holding with the proper amount of tension. For this to happen they must be properly tightened. To this day a simple, inexpensive, and effective way to consistently and accurately tighten a fastener does not exist. There are a number of tensioning methods that function well enough but they are both complicated and expensive. In most situations, less-than-perfect traditional methods are sufficient.

Engineers compensate for the inability to consistently and accurately determine bolt tension by massively over-designing joints. This accommodates inaccurate tightening and avoids catastrophic failure. Designers will specify more or larger bolts than needed in order to ensure that the joints are sufficiently clamped together. Fewer or smaller fasteners can be used when bolt preload is accurately and consistently controlled. Historically, the over-design of the joint has been far cheaper than controlling the assembly process.

However, current trends are moving away from the use of over-design. Increasing demands on cost, strength-to-weight ratios, product safety, product performance, and environmental safety have put pressure on designers, manufacturers, and assemblers to increase design efficiency. This trend has lead to the invention of more options in controlling bolt preload.

The tightening of a bolt follows a defined sequence of events and causes predictable results within the fastener. If the nut and head of the bolt are firmly seated against non-compressible materials, the torsional action of tightening the assembly stretches the bolt, thereby creating tension in the bolt. In most cases this tension or preload is required to make a fastening. By controlling torque, turn, or stretch, one can control the buildup of tension. The closer a method is to direct control of tension, the more expensive it will be.

Some options for tension controls during assembly are: Torque Control, Torque and Turn Control, Stretch Control, and Direct Tension Control.

Indirect Methods of Controlling Tension

Torque Control

Torque is one of the most common methods of installing fasteners. Often a torque value is specified for a given application; and with the use of a calibrated torque wrench, this torque value can be obtained. However, one must realize that this reading does not indicate the bolt tension directly. Rather the torque reading is only an *indirect* indication of the *desired* tension.

People frequently ask how much torque should be used to properly tension a fastener. Though the question appears simple, the answer depends on a variety of factors. During tightening roughly 90% of input energy is lost overcoming the mating friction under the head, nut, and mating threads. Only 10% of input energy is converted into bolt stretch. Bear in mind that this is an oversimplification of what may actually happen, since no consideration is given to the various forms of heat and strain energy introduced into the system.

In most situations there is a relatively simple relationship between the torque applied to the bolt or nut and the tension created in it. Usually this relationship is linear. For such cases, the equation: Torque = K x d x F is applicable; where d is the nominal diameter, F is the clamp load, and K is a torque variable (Nut Factor).

This equation implies a linear elastic zone of torque verses angle-tightening curve. For most common fastener materials the engineering values of the first two variables (d & F) are well defined. The problem with this equation lies in the Nut Factor (K).

The **K**, or **nut factor**, can be thought of as anything that increases or decreases the friction within the threads of the nut. This is a combination of three sub-factors:

- K1, a geometric factor- the shape of the threads. Variation in the shape of the thread may cause friction in increase or decrease.
- K2, a thread friction related factor- the friction between the threads of the bolt and the threads of the nut.
- K3, an underhead friction related factor- the friction of the nut against the surface it rotates on.

While there are published tables for K, these will usually vary from publication to publication.

No two bolts respond exactly the same to a given torque. There are numerous real-world complications. Dirt in a tapped hole, damaged threads, hole misalignment, and many other factors can absorb a large amount of the input torque and thereby decrease the amount of energy that actually becomes preload. Some of the other common variables affecting the K factor may include, but not limited to:

- Hardness of all parts
- Types of materials
- Class of fit
- Plating, thickness and type
- The number of times the fastener was used
- Surface finishes on all parts
- Washers, present or not
- Manufacturing processes, such as cut or rolled thread
- Type of tool used for tightening
- Speed of tightening
- Element being torqued, nut vs. bolt
- Lubricant: the type, amount, condition, method of application, contamination, and temperature

This is by no means an exhaustive list, and the extent to which these factors are controlled is directly proportionate to the cost incurred by the end user. Regardless of the time, effort, and money invested to monitor the process, complete control is impossible.

Hence, each fastener (even from the same lot) will require different torque values to achieve the same preload. The K value can be thought of as anything and everything that affects the relationship between torque and preload. The following is a brief examination of some estimated K factors that we have found in our laboratory. The list is intended to be used only as a suggestion and will actually vary per application.

K Factors	
Bolt Condition	**K**
Non-plated, black finish (dry)	0.20 - 0.30
Zinc-plated	0.17 - 0.22
Lubricated	0.12 - 0.16
Cadmium-plated	0.11 - 0.15

Due to variations in the K factor, even perfect input torque can give a variation of preload by as much as 25 - 30 %. The following example will demonstrate some complications that can be involved in calculating torque.

K = 0.20 (Non-lubricated; Dry)		
Hex Cap Screw 1/2 - 13 zinc plated Grade 5		
Using Torque = K x d x F		
K = 0.20	d = 0.50 in.	F = 9045 lbs.
T = **(0.20)** (0.5 in.) (9045 lbs.)		
T= 905 in.-lbs./12 = **75 ft.-lbs.**		

K = 0.10 (Highly lubricated)		
Hex Cap Screw 1/2 - 13 zinc plated Grade 5		
Using Torque = K x d x F		
K = 0.10	d = 0.50 in.	F = 9045 lbs.
T = **(0.10)** (0.5 in.) (9045 lbs.)		
T= 452 in.-lbs./12 = **38 ft.-lbs.**		

Note: F = (75% of the proof load) = 0.75 x 85,000 psi x 0.1419 sq.in = 9045 lbs.

Dry Torque Value used on Lubricated Bolts (K = 0.10)		
Hex Cap Screw 1/2 - 13 zinc plated Grade 5		
Torque = K x d x F to find F (preload): **F = T/(K x d)**		
K = 0.10	d = 0.50 in.	**T = 905 in.-lbs.**
F = **905 in.-lbs.**/ (**0.10** x 0.5 in.)		
F = **18,100 lbs.**		

The above example examines K = 0.20. This is a common K factor for zinc-plated fasteners. However, what might happen if the person installing the fastener without realizing the extent or type of lubrication with the same torque? In such conditions, K may drop to a value of 0.10. If the K = 0.20 (dry) torque calculation was used on a heavily lubricated bolt it would result in a preload of 18,100 lbs. This would exceed the tensile strength of a ½ - 13 Grade 5 bolt. More than likely the bolt in question would fail. The exact opposite can also happen, resulting in a lower clamp load than required and field failure.

As a group of fasteners is tightened, those tightened first will tend to relax slightly as subsequent fasteners are tightened. This is due to the creep and flow of heavily loaded contact surfaces and the relaxation between groups of bolts as they are individually tightened. The first bolts tightened partially pull the joint together. As we tighten the rest of the bolts the joint is further compressed, and the previously tightened bolts relax and lose some of their preload. In some cases, this can virtually eliminate bolt tension. If bolt tension is desired with a high degree of accuracy, the torque wrench is not the answer.

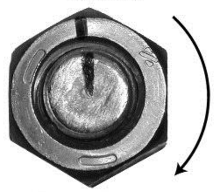
Torque and Turn Control (Torque-Degree)

This method involves tightening the fastener to a low initial "snug tight" condition and then applying a prescribed amount of turn to develop the required preload. The actual preload will depend on how far the nut is turned as well as how much preload was established prior to the turning.

The most general model of torque-turn analysis consists of four distinct zones.

1. The run-down prevailing torque zone is the initial application of the nut before the fastener head or nut contacts the bearing surface.

Use torque to reach "snug tight"



Turn an additional 1/4 to 1/3 after "snugged"

Desired clamp is achieved under proper conditions

2. The alignment or snugging zone is the area in which the fastener and the joint mating surfaces are drawn into alignment or a "snug" condition.

3. The elastic clamping range zone beings with the application of the angle-controlled tightening. Through this region the slope of the torque-angle curve is constant.

4. The post-yield zone is where an inflection point (permanent deformation) begins.

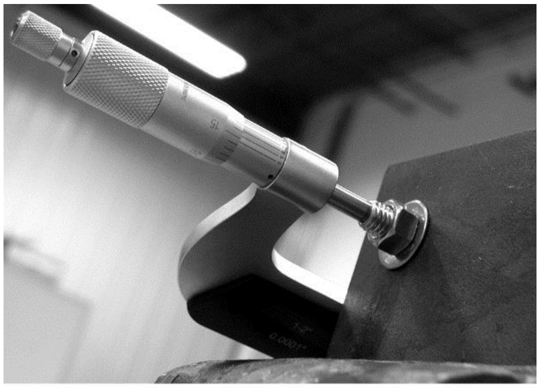
The amount of turn required for run-down, the amount of preload created during snugging, and the bolt-to-joint stiffness ratios are all extremely difficult to predict or control. As a result, pure turn control has a high potential for inaccuracy. But the fact remains that the preload applied to a previously snugged bolt indicates bolt preload regardless of friction. Thus, the combination of torque and turn is more accurate than either system alone.

Torque is used to determine the initial preload developed in a fastener during the snugging operation. Regardless of the external factors that can influence torque, this serves as a relatively accurate way of ensuring that slack is removed from the assembly before tightening. The turn of the nut method is a more reliable measure of the further increase of tension of a previously "snugged" bolt. Once the slack has been taken out of the assembly through the use of torque, it is easy to determine the amount of turn needed to reach the proper preload for the bolt.

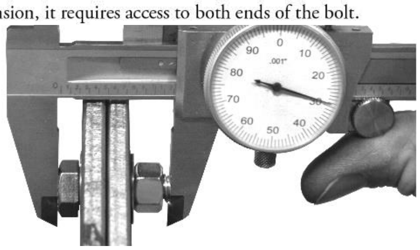
However, this method can only be used on joints with a predetermined response to assembly. Experiments must first be done on a sample joint to determine the affect of turn on the joint and the fastener. This increases the cost of the method. The method is also similar to the torque method in that it has no way to account for phenomena like creep relaxation and tightening sequence. Accuracy is still affected by variables such as friction and stiffness ratios.

Alternative Design Bolts
Alternative design bolts use design features that indirectly indicate tension. The most common alternative design bolt is the twist-off bolt or tension control (TC) bolt. A further explanation of TC bolts is given at the end of this chapter under Structural Bolting.

Stretch Control
The stretch created in the bolt during tightening allows the fastener to clamp. This tension creates the clamping force, which holds our joint together. In critical joints, it is vital to ensure that the proper amount of preload is applied to the fastener and that the preload remains over the service life of the joint.



While stretch control is a very accurate method of determining bolt tension, it requires access to both ends of the bolt.



Stretch control is one method which offers a very accurate indication of preload. In this method a tool is used to measure bolt stretch (a micrometer is the simplest method). The tool can be used only if there is access to both ends of the bolt before and after installation. Also, since both ends will probably not be parallel, several measurements must be taken at different points around the circumference.

Again, the spring-to-bolt analogy is a useful way of understanding the stretch control method. When a spring stretches, an equation can be used to determine the change in length; the same applies to a bolt. Bolt lubricity or bolt-to-joint stiffness ratios have no affect on the calculations of stretch. Also, the stretch control method can be used to measure the bolt tension well after the tightening process is complete.

However, several factors can complicate this method.

In the spring analogy, a bolt would be thought of as a complex spring. That is, the unthreaded body of the bolt is a relatively stiff spring and the threaded portion of the bolt would be a less stiff spring. Thus, when the bolt is loaded the threaded portion of the bolt will tend to stretch more than the unthreaded **shank** of the bolt.

Also, the threads within the nut or tapped hole will stretch less. The amount of engaged and unengaged thread stretch is directly affected by the fastener diameter. Calculations must use exact lengths of both the unthreaded portion of the bolt and the remaining length of the unengaged threaded portion of the bolt and also the diameter of the bolt.

To further complicate the idea, most authorities say that the head of a bolt also stretches slightly as we tighten a bolt. The head is thought to stretch about half as much as the same volume of body would stretch.

In order to use stretch control one must initially determine the amount of stretch each separate portion of the bolt contributes to the total stretch. One must also take into account that the tension and stretch in the bolt head and the nut are not uniform throughout, but rather fall off from a maximum value at the joint surface to zero (no tension) at the outer ends of the fastener.

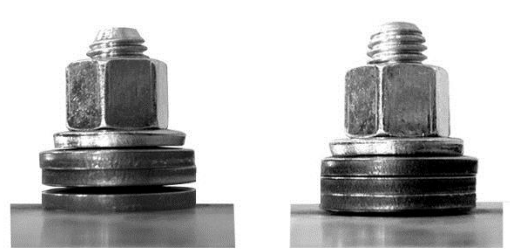
Other factors affect the relationship between stretch and tension. The basic elasticity of the bolt material may vary from lot to lot of bolts, the grip lengths and dimensional tolerances may vary, and the temperature of the bolt must be measured for precise tension determination. Overall, stretch control method is a very precise tool for evaluating bolt tension.

Direct Methods of Controlling Tension

The previously mentioned methods utilize indirect methods to indicate clamp load. Each method contains errors and uncertainties. Although some methods are more precise than others, none of them are able to control the tension developed in the fastener directly. However, the following methods will directly indicate the clamp load developed in our joint.

Washer Control
The least expensive and simplest tension control systems use direct tension indicating washers (DTI). A further explanation of DTI Washers is given at the end of this chapter under Structural Bolting.

DisDidiidididkdDisc Spring Washers (Belleville) involve a more spring-like action. A stack of these washers are placed under the head or the nut. As the fastener is tightened, the height of the Belleville is reduced. The washers deform elastically, so they will continue to push upwards on the head or nut. The washers can be used singularly or stacked in a means to modify the amount of deflection. Stacking in the same direction will add the spring constant in parallel, which will create a stiffer joint. Stacking in an alternating direction would be equivalent to adding springs in series, which would result in greater deflection.



When properly stacked and installed, Belleville Washers will indicate the amount o tension when tightening, as well as any loss of clamp during the life of the joint.

Hydraulic Tensioners
A hydraulic tensioner is sometimes used to tighten large diameter bolts. An upper collar is threaded down onto the exposed section of thread above the nut. Hydraulic pressure is then used to pull upward on the bolt. The nut is then run down, freely, against the upper surface of the joint. When the hydraulic pressure is relieved, the nut continues to hold the majority of the tension developed.

Ultrasonics
Ultrasonic instruments are another useful method of measuring bolt preload or tension. This instrument sends a brief burst of ultrasound through a bolt and measures the time required for the sound to echo off the end and return to the transducer. When a bolt is stretched, the time required for the signal to complete its circuit will increase. Elaborate computer equipment, grip length, bolt material, and thread run-out lengths are needed to measure the change in transmittal time to determine bolt tension. This equipment has been available for many years, but due to the cost and quantity of components needed, is used in only a few applications.