# Installation and Maintenance of Threaded Fasteners (Nuts and Bolts)

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The information contained in this booklet represents a significant collection of information about installation and maintenance of threaded fasteners (nuts and bolts) such as tightening of threaded fasteners, locking of threaded fasteners, helical coil inserts and screw thread tapping. This information will help to achieve increased reliability at a decreased cost. Assemblage of this information will provide a single point of reference that might otherwise be time consuming to obtain. Most of the information given in this booklet is mainly derived from literature on the subject from various sources as per the reference list given at the end of this booklet. For more information, please refer them. All information contained in this booklet has been assembled with great care. However, the information is given for guidance purposes only. The ultimate responsibility for its use and any subsequent liability rests with the end user. Please view the disclaimer uploaded at http://www.practicalmaintenance.net.

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# Content



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# Introduction

A bolt/screw is a threaded fastener. A bolted joint is a separable joint between two or more components using one or more bolts. The joint is designed considering that when the bolts (bolts of designed size and material) are preloaded (tensioned/tightened/assembled) as designed, the joint will fulfil its designed function (withstand the working loads). Hence, equally important to use of the designed bolts in a bolted joint in determining joint performance is the assembly procedure. If the bolts are not adequately preloaded, the joint will not perform as intended. All threaded fasteners also self-loosen under vibration, shock, thermal cycles, or the like. Hence it is also important to design a bolted joint with appropriate locking arrangement to prevent it from self-loosening.

In view of above, information about tightening of threaded fasteners and locking methods for threaded fasteners is given in this booklet. During maintenance of threaded fasteners, sometimes it is required to use helical coil inserts and tap threads. In view of this, information about helical coil inserts, tapping of screw thread and other useful information for maintenance (screw extraction and drill & counter bore sizes) is given in this booklet.

# Tightening of Threaded Fasteners

Threaded fasteners are tightened to clamp parts together. In gasketed joints, the purpose is to prevent leakage. In other joints, the clamping load/force is developed to transmit loads and prevent the parts from separating or shaking loose. In normal (non-gasketed) joints, the clamping force should be equal the working load. In gasketed joints, it should be sufficient to create a seal. The proper amount of tightening (or preloading/tensioning) the fasteners is very important. If the fasteners are too tight they may break - either during the tightening itself or when the working load is added to the preload in applications such as gasketed joints. If inadequately tightened, the fastener will shake loose in vibration. There is a tendency in fasteners subjected to cyclic loading to fail from fatigue if they are not tightened sufficiently. In view if this, information about various methods of tightening, sequence of tightening and recommended tightening torque values to preload fasteners (**for engineering applications**) is given in this chapter. The clamp load is also known as preload or initial tension load in the bolt. In this chapter the word “bolt” is used in a generic sense to cover all types of threaded fasteners (screws, studs, etc.).

## Tightening Methods

All fastener materials are slightly elastic and must be stretched a small amount to develop clamping load. However, if they are stretched beyond their elastic limit (yield point), they will deform permanently. In view of this, any tightening method must ensure that the stress in the bolt never exceeds the elastic limit, both during the tightening operation and when the assembly is later exposed to efforts during operation.

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. For bolting specifications that do not have a published proof load, it is usually calculated at 92% of yield strength.

Usually clamp load is calculated by arbitrarily assuming usable bolt strength to be 75% of the proof load for reusable connections. Higher or lower values of clamp load can be used depending on the application requirements and the judgment of the designer. For permanent connections, clamp load is calculated by arbitrarily assuming usable bolt strength to be 90% of the proof load.

Within elastic limit, stress is proportional to strain. From this information, one can easily calculate fastener to be stretched for desired preload. This method of preloading is very accurate but it requires that the ends of the bolts be properly prepared and also that all measurements be very carefully made. In addition, direct measurements are only possible where both ends of the fastener are available for measurement after installation. Since in most cases it is not possible to measure fastener elongation easily, other indirect methods are used for preloading. Following methods are used to preload a threaded fastener in order of increasing accuracy.

| Method | Accuracy |
|---|---|
| Feel | ± 35% |
| Torque wrench | ± 25% |
| Turn-of-nut | ± 15% |
| Direct tension indicating (DTI) washers | ± 10% |
| Bolt elongation | ± 3 to 5% |
| Strain gages | ± 1% |

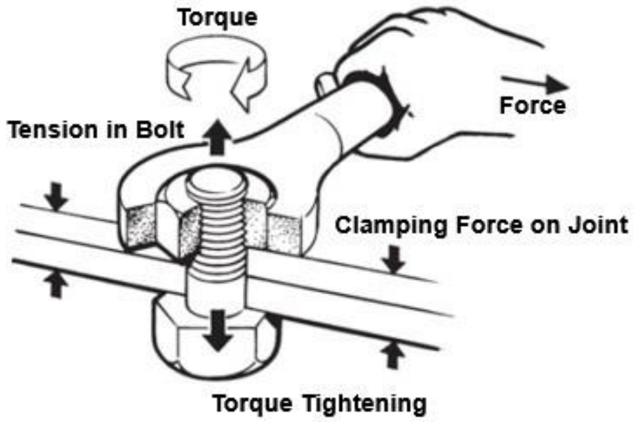
The decision as to which tightening method to use depends primarily on the criticality of the joint as cost increases with higher accuracy method. Some applications will allow the high inaccuracy of the "feel" method, while the high cost, highly accurate strain gages are used almost entirely in the laboratory.

## Feel Method

In this method, fasteners are tightened as per feel of a person based on his work experience. As there is no measurement, quality of work depends on person's expertise. This method remains the most common form of bolt tightening for non-engineered type applications. This method is generally satisfactory in noncritical joints where loads are static and not subject to vibration. The advantages of this method are speed and ease of use.

## Torque Wrench Method

Torque wrench (often called torque tightening) method is by far the most common tensioning method for engineered joints because of low cost and simplicity. In this method, fasteners are tightened by a calibrated torque wrench to a desired torque value. A torque wrench is a manual wrench which incorporates a gauge or other method to indicate the amount of torque transferred to the fastener. Torque wrenches are a means to improve control over bolt preloading in comparison with feel method.




In this method much attention is given to the level of torque that should be applied to a specific application. However, it is not the torque that is important but the end result of the torque, bolt preload (tension in the bolt which creates a compressive force in the bolted joint known as clamp force as shown in above figure).

Torque is the measure of the torsion required to turn a nut up the inclined plane of a thread. The efficiency of the nut's turn along the bolt thread to generate preload is dependent upon many factors, including thread pitch, friction between the threads and friction between the nut face and the matting face.

In general, only about 10 percent of the applied torque goes toward providing bolt preload. The rest is lost in overcoming friction: 50 percent in overcoming the friction between the nut (or bolt head) and matting/bearing face, and 40 percent in overcoming friction between the threads of the nut and the bolt. Hence it may be noted that change in the coefficient of friction for different conditions can have a very significant effect on fastener loading.

Fastener manufacturers usually recommend torque values for each size and fastener material based on test carried out by them. In absence of information about torque values, the torque tightening equation may be used to decide torque value.

Within the elastic range (before permanent stretch is induced), the relationship between torque and tension is essentially linear. The torque tightening equation (formula) developed based on empirical test results is as under.

$$T = KDP$$

Where,
T = torque, Nm (in-pounds).
D = fastener nominal diameter, m (inch)
P = preload, N (pounds)
K = “nut factor,” “tightening factor,” or “K-value”

Based on condition of lubrication, the nut factor may be assumed as give in the following table.

| Material | Condition of Lubrication | Nut Factor (k-value) |
|---|---|---|
| Steel | Graphite in petrolatum or oil | 0.1 |
| | Molybdenum disulfide grease | 0.11 |
| | Light machine oil as shipped | 0.15 |
| | Copper-based anti-seize compound | 0.13 |
| Steel | Without lubrication | 0.2 |
| Steel, hot dip galvanized | Without lubrication | 0.25 |
| Plated fasteners | Without lubrication | 0.15 |
| Stainless steel | Without lubrication | 0.30 |

However, the lubricant is not the only variable affecting the nut factor. The nut factor also depends on bolt diameter and material, tightening speed, thread fit, and operator skill. Thus the K-value is not the coefficient of the friction (μ); it is an empirically derived correlation factor. Hence, when accuracy is very important, the actual nut factor for a given application should be determined experimentally.

**Checking Joint Tightness**

Joint tightness can be checked by checking the applied torque by static measurement or dynamic measurement.

In static measurement, the torque is checked after the tightening process has been completed. The measurement is usually done by hand with a torque wrench which has either a spring loaded torque scale or a strain gauge transducer activated instrument. To measure the static torque, the torque value must be read instantly as the screw starts to turn while checking it with a torque wrench. An electronic torque wrench can be used for a more sophisticated static measurement of the joint. The tool has strain gauge torque transducer which gives a high level of accuracy.

A very common method for checking the tightening torque is to use a click wrench, which is a torque wrench equipped with a clutch that can be pre-adjusted to a specific torque. If the torque is greater than the preset torque value, the clutch will release with a click. If the torque is less, final torque-up is possible until the wrench clicks. However, over-tightening cannot be detected with the click wrench.

In dynamic measurement, the torque is continuously measured during the complete tightening cycle. This is usually the preferred method in production where power tools are used for tightening. Dynamic measurement also eliminates the necessity for subsequent checking. Dynamic measurement is done either directly by measuring with a built-in or a separate in-line torque transducer, or indirectly by current measurement in some sophisticated electric powered screwdrivers and nut-runners.

## Turn-of-nut Method

The turn-of-nut method utilizes the change in bolt length on its tightening. In theory, one nut/bolt revolution (360° rotation) should increase the bolt length by the thread pitch.

In this method, the nut or bolt is turned a predetermined number of degrees after all play has been removed from the joint by snug tightening. Snug tightening (snugging) is the process of pulling parts of a joint together and the joint have been tightened sufficiently to prevent the removal of the nuts without the use of a wrench. In snug tightening, most of the applied torque (in turning nut/bolt) is absorbed in the joint with little tension being given to the bolt.

This method eliminates the friction factor. However, its accuracy is affected by the care of the workman in measuring the angle the nut or bolt is turned.

This method is commonly used in structural bolting because if the joint is not stiff (for example - gasketed joint), the bolts will get tensioned/preloaded during snugging due to joint (gasket) compression.

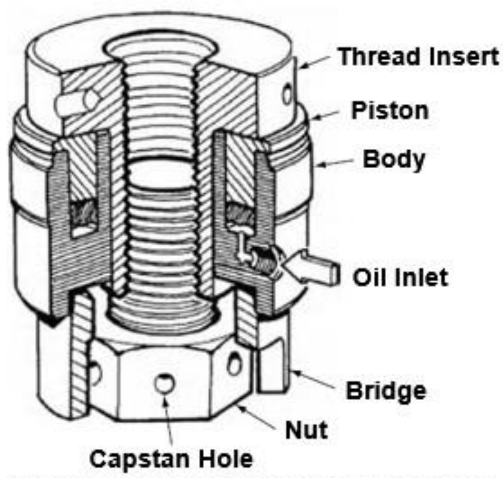
## Direct Tension Indicating (DTI) Washers

Direct tension indicating (DTI) washers utilize compression of plastically deformable material of washer under load. The use of load indicating washers is widespread in structural engineering. One type of such washers has small raised pips on their surface which plastically deform as the bolt is tensioned. The correct preload is achieved when a predetermined gap is present between the washer and the under head of the bolt. This is measured using feeler gauges. The smaller the gap, greater is the tension in the bolt.

## Bolt Elongation

Since for any given material within its elastic limit, stress is proportional to strain, in a bolt elongation method, the relationship is used to preload bolt/stud by staining (stretching) it for desired stress (loading). Various methods used to stretch bolt or measure stretch as the bolt is being loaded are as under.

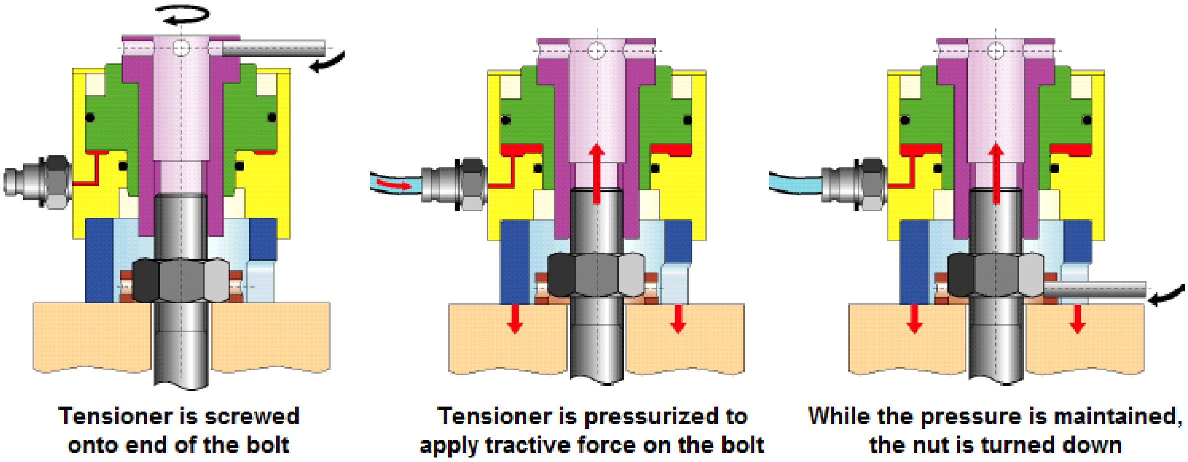
**Use of Hydraulic Tensioner**




Cutaway View of a Hydraulic Tensioner

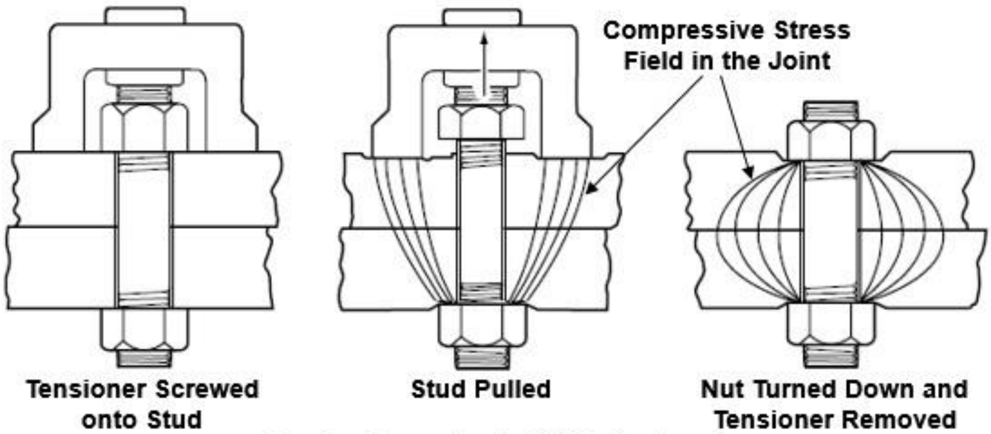
Above figure shows cutaway view of a typical hydraulic tensioner.

Hydraulic tensioner is used to tighten a fastener by stretching it rather than applying a large torque to the nut. As shown in the following figure, to start the tensioning process the tensioner is screwed onto end of the stud/bolt by turning (running) down the thread insert of the tensioner by hand over exposed threads of the stud to be tightened. It may be noted that thread insert is screwed onto the stud threads beyond the threads engaged by the stud's own nut. Now the tensioner is pressurized to apply tractive force on the bolt to stretch it. After the stud is stretched, while the pressure is maintained, the nut of the stud is turned down to snug it against the top surface of the joint. The hydraulically applied load is then removed resulting in tension/preload being induced into the stud.




Operating Principle of Hydraulic Bolt Tensioner

At first glance this method sounds like a perfect answer to the torque and friction uncertainties in the torque wrench method but there is one problem. As shown in the following figure, the amount of tension retained by the fastener's nut is never the same as the tension introduced by the tensioner because the nut must embed itself in the joint to pick up the load originally supported by the much larger feet of the tensioner. This elastic recovery loss, percent of the initial tension, depends on whether the fastener is relatively long (smaller loss) or short, and on how much torque was applied to the nut when it was turned down (more torque, less loss). In view of this, typically the initial tension should be 25% to 30% higher than the tension desired in the stud/bolt. The loss of tension is often referred to as "tensioner efficiency".




Tension Loss due to Nut Embedment

Ensuring that the nut is turned down firmly before removing the hydraulically applied load is the most important consideration in the tensioning process. If the nut is not turned down firmly, zero preload can result. Nut turning down is adversely affected by the following:

- A poorly constructed or damaged nut turning down mechanism.
- A tensioner base that does not fit squarely on the joint surface. Check the base for signs of yielding or distortion. An ill-fitting base can create interference with the nut and thus prevent nut turning down.
- Studs/bolts that are not perpendicular to the joint surface. Non-perpendicularity results in stud bending and binding of the nut during turning down. Shimming the tensioner can correct for perpendicularity problem.
- Avoid using fine threads. Fine threads can cause the nut to bind during turning down. Coarse threads are preferable.

Hydraulic tensioners, however, are very good tools when it comes to preloading large fasteners. They can be gang-driven from a single hydraulic pump and so can tighten several fasteners simultaneously with the same initial tension in each. This can be a very important feature when you are tightening large joints.

Tensioners also eliminate the galling problems often encountered when we attempt to torque large fasteners (75 mm in diameter or more) because in this method the male and female threads are not turned relative to each other under heavy contact pressure.

**Heat Tightening**

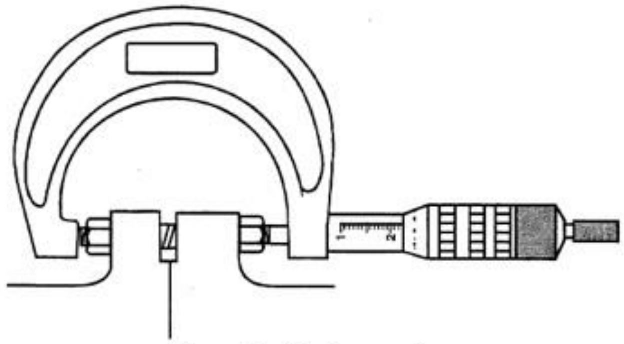
In heat tightening a bolt heater is used to accomplish the same thing as a hydraulic tensioner, although the equipment and procedures used are vastly different.

Heat tightening utilizes the thermal expansion characteristics of the stud/bolt. In this method, the stud is heated to expand. To heat the stud, a heating rod is inserted in a hole drilled down the axis of the stud. The stud expands (increases in both length and diameter) as it heats up. After it has increased in length by a desired amount, the nut is placed on the stud and is run down hard against the top surface of the joint. The nut is supposed to retain the change in length of the stud. Preload builds up in the fastener as the stud cools. Dial gauges or micrometers are usually used to measure the net change in length of the studs after they have cooled. If they have been stretched too much or too little, the studs must be reheated and the nuts run down again. Methods of stud heating include direct flame, heating rod or any other suitable method. The process is widely used on very large fasteners.

One big advantage of the bolt heater is that it is very inexpensive. The larger the stud (in diameter), the greater the cost advantage of the heater over the wrench or hydraulic tensioner large enough to do the job. A disadvantage is that this is a relatively slow procedure, requiring a fair amount of skill on the part of the operators. Another possible disadvantage is the fact that decarbonization of thread surfaces can sometimes occur when the stud is heated.

**Stretch Measurements**

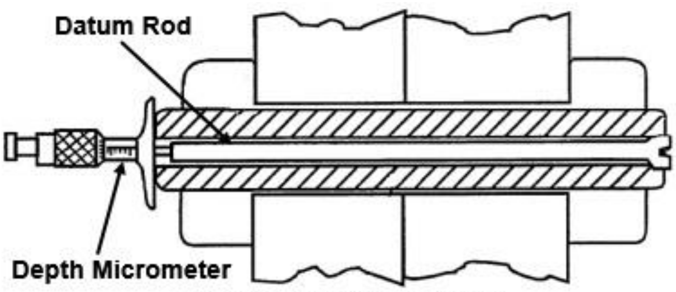
"C" type micrometers, depth micrometers and ultrasonic extensometers are used to measure fastener stretch. As they are not practical for mass production operations, they are used for critical joints in heavy equipment, process systems, and the like.



Use of a C-micrometer

As shown in above figure a "C" type micrometer requires access to both ends of the fastener and a reasonably short fastener length. However, there are number of problems associated with a "C" type micrometer measurements; for example, the ends of a bolt are not flat and parallel to each other as received.

A datum rod and depth micrometer are used to make stretch measurements if the fastener is drilled for the datum rod.




Use of a Depth Micrometer

In this method, a datum rod (loose rod) having ground and parallel ends is placed inside a hole which has been drilled through the center of the stud. The lower end of the hole is capped by a threaded plug. Instead of a threaded plug, sometimes a rod is used having an enlarged end which is threaded (as shown in above figure). A depth micrometer is now used to measure the distance between the end of the stud and the datum rod in the center. As the stud is stretched, this distance increases because the datum rod, being loose, is not stretched. Generally, depth micrometers are used to control preload in very large studs, especially those tightened by heater rods or hydraulic tensioners. This method also has the advantage that the operator only needs access to one end of the bolt.

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

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

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

## Strain Gages

One way to determine stress is to use strain (not stress) gages. Strain gages may be applied directly to the outside surface of the bolt or by installing them in the bolt by drilling a hole in the center of the bolt. With the proper procedures and instruments, one can determine the stress with far more precision. However, this method is expensive and not always practical.

Another way to measure preload is to use a force/sensor washer - a compressible ring that has been provided with strain gages. Force washer can be used to measure preload continuously while the fastener is being tightened. An obvious disadvantage is their cost - they have to be left in place after use to be meaningful. As a result, force washers are useful only for experimental measurements and for very special applications.

## Recommended Tightening Torque

Generally, it is recommended that preload (clamp load) should be 75% of the proof load for reusable connections.

In case torque value for tightening a fastener is not specified, it can be calculated using torque tightening equation (formula): T = KDP.

Where,
T = torque, Nm (in-pounds).
D = fastener nominal diameter, m (inch)
P = preload, N (pounds)
K = “nut factor,” “tightening factor,” or “K-value”

Torque values given in the following table may be used for tightening metric coarse pitch thread fasteners. The torque values are calculated using the torque tightening equation: T = KDP where; K = 0.15 for steel fasteners lubricated with light machine oil in as shipped condition. The torque values will give preload (clamp load) equal to 75% of the proof load.

| **Torque for Metric Coarse Pitch Thread Fasteners** | | | | | | |
|---|---|---|---|---|---|---|
| **Size** | **Property Class 8.8** | | **Property Class 10.9** | | **Property Class 12.9** | |
| | kg m | ft lb | kg m | ft lb | kg m | ft lb |
| M3 | 0.10 | 0.73 | 0.14 | 1.04 | 0.17 | 1.21 |
| M3.5 | 0.16 | 1.14 | 0.23 | 1.63 | 0.26 | 1.91 |
| M4 | 0.23 | 1.69 | 0.33 | 2.42 | 0.39 | 2.83 |
| M5 | 0.47 | 3.41 | 0.68 | 4.89 | 0.79 | 5.72 |
| M6 | 0.80 | 5.77 | 1.15 | 8.31 | 1.34 | 9.70 |
| M7 | 1.35 | 9.75 | 1.93 | 13.93 | 2.25 | 16.26 |
| M8 | 1.95 | 14.07 | 2.79 | 20.17 | 3.26 | 23.56 |
| M10 | 3.87 | 27.95 | 5.52 | 39.90 | 6.46 | 46.70 |
| M12 | 6.73 | 48.67 | 9.64 | 69.67 | 11.26 | 81.42 |
| M14 | 10.71 | 77.45 | 15.34 | 110.90 | 17.99 | 130.06 |
| M16 | 16.70 | 120.77 | 23.86 | 172.53 | 27.90 | 201.72 |
| M18 | 23.75 | 171.70 | 32.83 | 237.39 | 38.41 | 277.70 |
| M20 | 33.73 | 243.86 | 46.57 | 336.76 | 54.60 | 394.82 |
| M22 | 45.93 | 332.11 | 63.60 | 459.85 | 74.20 | 536.49 |
| M24 | 58.37 | 422.03 | 80.67 | 583.27 | 94.16 | 680.82 |
| M27 | 85.18 | 615.87 | 118.01 | 853.26 | 137.83 | 996.59 |
| M30 | 115.98 | 838.58 | 160.37 | 1159.58 | 187.21 | 1353.68 |
| M33 | 157.48 | 1138.68 | 218.05 | 1576.63 | 254.77 | 1842.14 |
| M36 | 202.36 | 1463.16 | 279.99 | 2024.54 | 327.07 | 2364.95 |
| M39 | 262.17 | 1895.64 | 362.38 | 2620.26 | 423.67 | 3063.44 |

If you're having a condition of lubrication different than fasteners lubricated with light machine oil in as shipped condition (K = 0.15), you can compute the new torque value from the equation: $T = (T_T \times K) / 0.15$

Where,
T = corrected torque
K = nut factor for your application
$T_T$ = torque value given in above table

Similarly, you can also compute the new torque value for a desired preload (clamp load) different than 75% of the proof load.

Following tables (as per ISO 898-1) may be used to calculate torque for other property classes of metric coarse pitch threaded fasteners and metric fine pitch threaded fasteners. In these tables, full stop (.) is used as a decimal marker.

| **Proof Load, $F_p$ ($A_{s,nom} \times S_{p,nom}$), N - ISO Metric Coarse Pitch Threaded Fasteners** | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Thread[a] *d* | Nominal Stress Area $A_{s,nom}$ mm$^2$ | Property Class | | | | | | | | |
| | | 4.6 | 4.8 | 5.6 | 5.8 | 6.8 | 8.8 | 9.8 | 10.9 | 12.9 |
| M3.5 | 6.78 | 1530 | 2100 | 1900 | 2580 | 2980 | 3940 | 4410 | 5630 | 6580 |
| M4 | 8.78 | 1980 | 2720 | 2460 | 3340 | 3860 | 5100 | 5710 | 7290 | 8520 |
| M5 | 14.2 | 3200 | 4400 | 3980 | 5400 | 6250 | 8230 | 9230 | 11800 | 13800 |
| M6 | 20.1 | 4520 | 6230 | 5630 | 7640 | 8840 | 11600 | 13100 | 16700 | 19500 |
| M7 | 28.9 | 6500 | 8960 | 8090 | 11000 | 12700 | 16800 | 18800 | 24000 | 28000 |
| M8 | 36.6 | 8240[b] | 11400 | 10200[b] | 13900 | 16100 | 21200[b] | 23800 | 30400[b] | 35500 |
| M10 | 58 | 13000[b] | 18000 | 16200[b] | 22000 | 25500 | 33700[b] | 37700 | 48100[b] | 56300 |
| M12 | 84.3 | 19000 | 26100 | 23600 | 32000 | 37100 | 48900[c] | 54800 | 70000 | 81800 |
| M14 | 115 | 25900 | 35600 | 32200 | 43700 | 50600 | 66700[c] | 74800 | 95500 | 112000 |
| M16 | 157 | 35300 | 48700 | 44000 | 59700 | 69100 | 91000[c] | 102000 | 130000 | 152000 |
| M18 | 192 | 43200 | 59500 | 53800 | 73000 | 84500 | 115000 | - | 159000 | 186000 |
| M20 | 245 | 55100 | 76000 | 68600 | 93100 | 108000 | 147000 | - | 203000 | 238000 |
| M22 | 303 | 68200 | 93900 | 84800 | 115000 | 133000 | 182000 | - | 252000 | 294000 |
| M24 | 353 | 79400 | 109000 | 98800 | 134000 | 155000 | 212000 | - | 293000 | 342000 |
| M27 | 459 | 103000 | 142000 | 128000 | 174000 | 202000 | 275000 | - | 381000 | 445000 |
| M30 | 561 | 126000 | 174000 | 157000 | 213000 | 247000 | 337000 | - | 466000 | 544000 |
| M33 | 694 | 156000 | 215000 | 194000 | 264000 | 305000 | 416000 | - | 576000 | 673000 |
| M36 | 817 | 184000 | 253000 | 229000 | 310000 | 359000 | 490000 | - | 678000 | 792000 |
| M39 | 976 | 220000 | 303000 | 273000 | 371000 | 429000 | 586000 | - | 810000 | 947000 |

[a] - Where no thread pitch is indicated in a thread designation, coarse pitch is specified.
[b] - For fasteners with thread tolerance 6az in accordance with ISO 965-4 subject to hot dip galvanizing, reduced values in accordance with ISO 10684, Annex A, apply.
[c] - For structural bolting 50700 N (for M12), 68800 N (for M14) and 94500 N (for M16).

| **Proof Load, $F_p$ ($A_{s,nom} \times S_{p,nom}$), N - ISO Metric Fine Pitch Threaded Fasteners** | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Thread *d × P* | Nominal Stress Area $A_{s,nom}$ mm$^2$ | Property Class | | | | | | | | |
| | | 4.6 | 4.8 | 5.6 | 5.8 | 6.8 | 8.8 | 9.8 | 10.9 | 12.9 |
| M8x1 | 39.2 | 8820 | 12200 | 11000 | 14900 | 17200 | 22700 | 25500 | 32500 | 38000 |
| M10x1.25 | 61.2 | 13800 | 19000 | 17100 | 23300 | 26900 | 35500 | 39800 | 50800 | 59400 |
| M10x1 | 64.5 | 14500 | 20000 | 18100 | 24500 | 28400 | 37400 | 41900 | 53500 | 62700 |
| M12x1.5 | 88.1 | 19800 | 27300 | 24700 | 33500 | 38800 | 51100 | 57300 | 73100 | 85500 |
| M12x1.25 | 92.1 | 20700 | 28600 | 25800 | 35000 | 40500 | 53400 | 59900 | 76400 | 89300 |
| M14x1.5 | 125 | 28100 | 38800 | 35000 | 47500 | 55000 | 72500 | 81200 | 104000 | 121000 |
| M16x1.5 | 167 | 37600 | 51800 | 46800 | 63500 | 73500 | 96900 | 109000 | 139000 | 162000 |
| M18x1.5 | 216 | 48600 | 67000 | 60500 | 82100 | 95000 | 130000 | - | 179000 | 210000 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| M20x1.5 | 272 | 61200 | 84300 | 76200 | 103000 | 120000 | 163000 | - | 226000 | 264000 |
| M22x1.5 | 333 | 74900 | 103000 | 93200 | 126000 | 146000 | 200000 | - | 276000 | 323000 |
| M24x2 | 384 | 86400 | 119000 | 108000 | 146000 | 169000 | 230000 | - | 319000 | 372000 |
| M27x2 | 496 | 112000 | 154000 | 139000 | 188000 | 218000 | 298000 | - | 412000 | 481000 |
| M30x2 | 621 | 140000 | 192000 | 174000 | 236000 | 273000 | 373000 | - | 515000 | 602000 |
| M33x2 | 761 | 171000 | 236000 | 213000 | 289000 | 335000 | 457000 | - | 632000 | 738000 |
| M36x3 | 865 | 195000 | 268000 | 242000 | 329000 | 381000 | 519000 | - | 718000 | 839000 |
| M39x3 | 1030 | 232000 | 319000 | 288000 | 391000 | 453000 | 618000 | - | 855000 | 999000 |

In above tables, $S_{p,nom}$ = nominal stress under proof load.

For conversion of units, use: 1 kgf m = 9.807 N m = 7.231 lbf ft

Tensile stress areas for UNC, UNF, and 8 UN thread series are given in the following table. The table can be used to calculate preload (clamp load) for imperial (inch) fasteners.

| **Tensile Stress Areas** | | | | | |
|---|---|---|---|---|---|
| **Nominal Size in. - Threads per Inch** | **Tensile Stress Area, $A_s$ in.$^2$ for UNC** | **Nominal Size in. - Threads per Inch** | **Tensile Stress Area, $A_s$ in.$^2$ for UNF** | **Nominal Size in. - Threads per Inch** | **Tensile Stress Area, $A_s$ in.$^2$ for 8 UN** |
| 0.073 - 64 | 0.00263 | 0.060 - 80 | 0.00180 | - | - |
| 0.086 - 56 | 0.00370 | 0.073 - 72 | 0.00278 | - | - |
| 0.099 - 48 | 0.00487 | 0.086 - 64 | 0.00394 | - | - |
| 0.112 - 40 | 0.00604 | 0.099 - 56 | 0.00523 | - | - |
| 0.125 - 40 | 0.00796 | 0.112 - 48 | 0.00661 | - | - |
| 0.138 - 32 | 0.00909 | 0.125 - 44 | 0.00830 | - | - |
| 0.164 - 32 | 0.0140 | 0.138 - 40 | 0.01015 | - | - |
| 0.190 - 24 | 0.0175 | 0.164 - 36 | 0.01474 | - | - |
| 0.216 - 24 | 0.0242 | 0.190 - 32 | 0.0200 | - | - |
| ¼ - 20 | 0.0318 | ¼ - 28 | 0.0364 | - | - |
| 5/16 - 18 | 0.0524 | 5/16 - 24 | 0.0580 | - | - |
| ⅜ - 16 | 0.0775 | ⅜ - 24 | 0.0878 | - | - |
| 7/16 - 14 | 0.1063 | 7/16 - 20 | 0.1187 | - | - |
| ½ - 13 | 0.1419 | ½ - 20 | 0.1599 | - | - |
| 9/16 - 12 | 0.182 | 9/16 - 18 | 0.203 | - | - |
| ⅝ - 11 | 0.226 | ⅝ - 18 | 0.256 | - | - |
| ¾ - 10 | 0.334 | ¾ - 16 | 0.373 | - | - |
| ⅞ - 9 | 0.462 | ⅞ - 14 | 0.509 | - | - |
| 1 - 8 | 0.606 | 1 - 12 | 0.663 | 1 - 8 | 0.606 |
| 1⅛ - 7 | 0.763 | 1⅛ - 12 | 0.856 | 1⅛ - 8 | 0.790 |
| 1¼ - 7 | 0.969 | 1¼ - 12 | 1.073 | 1¼ - 8 | 1.000 |
| 1⅜ - 6 | 1.155 | 1⅜ - 12 | 1.315 | 1⅜ - 8 | 1.233 |
| 1½ - 6 | 1.405 | 1½ - 12 | 1.581 | 1½ - 8 | 1.492 |
| 1¾ - 5 | 1.90 | - | - | 1¾ - 8 | 2.08 |
| 2 - 4½ | 2.50 | - | - | 2 - 8 | 2.77 |
| 2¼ - 4½ | 3.25 | - | - | 2¼ - 8 | 3.56 |
| 2½ - 4 | 4.00 | - | - | 2½ - 8 | 4.44 |
| 2¾ - 4 | 4.93 | - | - | 2¾ - 8 | 5.43 |
| 3 - 4 | 5.97 | - | - | 3 - 8 | 6.51 |
| 3¼ - 4 | 7.10 | - | - | 3¼ - 8 | 7.69 |
| 3½ - 4 | 8.33 | - | - | 3½ - 8 | 8.96 |
| 3¾ - 4 | 9.66 | - | - | 3¾ - 8 | 10.34 |
| 4 - 4 | 11.08 | - | - | 4 - 8 | 11.81 |

The grade designations and important mechanical properties for fasteners as per SAE J429 are given in the following table.

| Grade Designation | Products | Nominal Size, Diameter, in. | Full Size[1] Fasteners Proof Stress psi | Full Size Fasteners Tensile Strength (Stress) Min, psi | Machine Test Specimens Yield Strength[2] (Stress) Min, psi | Core Hardness Rockwell | |
|---|---|---|---|---|---|---|---|
| | | | | | | Min | Max |
| 1 | Bolts, Screws, Studs | ¼ thru 1½ | 33000 | 60000 | 36000 | B70 | B100 |
| 2 | Bolts, Screws, Studs | ¼ thru ¾ | 55000 | 74000 | 57000 | B80 | B100 |
| | | Over ¾ thru 1½ | 33000 | 60000 | 36000 | B70 | B100 |
| 4 | Studs | ¼ thru 1½ | 65000 | 115000 | 100000 | C22 | C32 |
| 5 | Bolts, Screws, Studs | ¼ thru 1 | 85000 | 120000 | 92000 | C25 | C34 |
| | | Over 1 thru 1½ | 74000 | 105000 | 81000 | C19 | C30 |
| 5.1 | Sems | No. 4 thru ⅝ | 85000 | 120000 | - | C25 | C40 |
| 5.2 | Bolts, Screws | ¼ thru 1 | 85000 | 120000 | 92000 | C26 | C36 |
| 8 | Bolts, Screws, Studs | ¼ thru 1½ | 120000 | 150000 | 130000 | C33 | C39 |
| 8.1 | Studs | ¼ thru 1½ | 120000 | 150000 | 130000 | C33 | C39 |
| 8.2 | Bolts, Screws | ¼ thru 1 | 120000 | 150000 | 130000 | C33 | C39 |

[1] - "Full Size" means a tension test specimen consisting of a completed fastener for testing in the ready to use condition without alteration.

[2] - Yield strength is stress at which a permanent set of 0.2% of gage length occurs.

It may be noted that above table gives proof stress in psi. Hence to find out proof load on a fastener, multiply proof stress by tensile stress area of the fastener (proof load = proof stress × tensile stress area of the fastener).

Following tables may be used to torque tighten SAE fasteners (SAE J429) to preload (clamp load) them to 75% of the proof load. For the torque values given in the tables, value of nut factor (K) is assume to be 0.20 for dry bolts (bolts without lubrication) and 0.15 for lubricated bolts (as shipped light machine oil lubricated bolts).

| Tightening Torque for SAE Grade 2 Coarse Thread Fasteners | | | |
|---|---|---|---|
| **Size** | **Clamp Load, lbs.** | **Dry Bolts** | **Lubricated Bolts** |
| $^1/_4$-20 (.250) | 1,313 | 66 in. lbs | 49 in. lbs. |
| $^5/_{16}$-18 (.3125) | 2,175 | 11 ft. lbs | 8 ft. lbs. |
| $^3/_8$-16 (.375) | 3,188 | 20 ft. lbs | 15 ft. lbs. |
| $^7/_{16}$-14 (.4375) | 4,388 | 32 ft. lbs. | 24 ft. lbs. |
| $^1/_2$-13 (.500) | 5,850 | 49 ft. lbs. | 37 ft. lbs. |
| $^9/_{16}$-12 (.5625) | 7508 | 70 ft. lbs | 53 ft. lbs |
| $^5/_8$-11 (.625) | 9,300 | 97 ft. lbs. | 73 ft. lbs. |
| $^3/_4$-10 (.750) | 11,400 | 166 ft. lbs. | 125 ft. lbs. |
| $^7/_8$-9 (.875) | 13,800 | 173 ft. lbs. | 129 ft. lbs. |
| 1-8 (1.000) | 15,000 | 250 ft. lbs. | 188 ft. lbs. |
| $1^1/_8$-7 (1.125) | 18,900 | 354 ft. lbs. | 286 ft. lbs. |
| $1^1/_4$-7 (1.250) | 24,000 | 500 ft. lbs. | 375 ft. lbs. |
| $1^3/_8$-6 (1.375) | 28,575 | 655 ft. lbs. | 491 ft. lbs. |
| $1^1/_2$-6 (1.500) | 34,800 | 870 ft. lbs. | 652 ft. lbs. |

| Tightening Torque for SAE Grade 5 Coarse Thread Fasteners | | | |
|---|---|---|---|
| **Size** | **Clamp Load, lbs.** | **Dry Bolts** | **Lubricated Bolts** |
| $^1/_4$-20 (.250) | 2,025 | 8 ft. lbs | 76 in. lbs. |
| $^5/_{16}$-18 (.3125) | 3,338 | 17 ft. lbs | 13 ft. lbs. |
| $^3/_8$-16 (.375) | 4,950 | 31 ft. lbs | 23 ft. lbs. |
| $^7/_{16}$-14 (.4375) | 6,788 | 50 ft. lbs. | 37 ft. lbs. |
| $^1/_2$-13 (.500) | 9,075 | 76 ft. lbs. | 57 ft. lbs. |
| $^9/_{16}$-12 (.5625) | 11,625 | 109 ft. lbs. | 82 ft. lbs. |
| $^5/_8$-11 (.625) | 14,400 | 150 ft. lbs. | 112 ft. lbs. |
| $^3/_4$-10 (.750) | 21,300 | 266 ft. lbs. | 200 ft. lbs. |
| $^7/_8$-9 (.875) | 29,475 | 430 ft. lbs. | 322 ft. lbs. |

| | | | |
|---|---|---|---|
| 1-8 (1.000) | 38,625 | 644 ft. lbs. | 483 ft. lbs. |
| $1^{1}/_{8}$-7 (1.125) | 42,375 | 794 ft. lbs. | 596 ft. lbs. |
| $1^{1}/_{4}$-7 (1.250) | 53,775 | 1120 ft. lbs. | 840 ft. lbs. |
| $1^{3}/_{8}$-6 (1.375) | 64,125 | 1470 ft. lbs. | 1102 ft. lbs. |
| $1^{1}/_{2}$-6 (1.500) | 78,000 | 1950 ft. lbs. | 1462 ft. lbs. |

| **Tightening Torque for SAE Grade 8 Coarse Thread Fasteners** | | | |
|---|---|---|---|
| **Size** | **Clamp Load, lbs.** | **Dry Bolts** | **Lubricated Bolts** |
| $^{1}/_{4}$-20 (.250) | 2,850 | 12 ft. lbs | 9 ft. lbs. |
| $^{5}/_{16}$-18 (.3125) | 4,725 | 25 ft. lbs | 18 ft. lbs. |
| $^{3}/_{8}$-16 (.375) | 6,975 | 44 ft. lbs | 33 ft. lbs. |
| $^{7}/_{16}$-14 (.4375) | 9,600 | 70 ft. lbs. | 52 ft. lbs. |
| $^{1}/_{2}$-13 (.500) | 12,750 | 106 ft. lbs. | 80 ft. lbs. |
| $^{9}/_{16}$-12 (.5625) | 16,350 | 153 ft. lbs. | 115 ft. lbs. |
| $^{5}/_{8}$-11 (.625) | 20,325 | 212 ft. lbs. | 159 ft. lbs. |
| $^{3}/_{4}$-10 (.750) | 30,075 | 376 ft. lbs. | 282 ft. lbs. |
| $^{7}/_{8}$-9 (.875) | 41,550 | 606 ft. lbs. | 454 ft. lbs. |
| 1-8 (1.000) | 54,525 | 909 ft. lbs. | 682 ft. lbs. |
| $1^{1}/_{8}$-7 (1.125) | 68,700 | 1288 ft. lbs. | 966 ft. lbs. |
| $1^{1}/_{4}$-7 (1.250) | 87,225 | 1817 ft. lbs. | 1363 ft. lbs. |
| $1^{3}/_{8}$-6 (1.375) | 103,950 | 2682 ft. lbs. | 1787 ft. lbs. |
| $1^{1}/_{2}$-6 (1.500) | 126,450 | 3161 ft. lbs. | 2371 ft. lbs. |

| **Tightening Torque for SAE Grade 8 Fine Thread Fasteners** | | | |
|---|---|---|---|
| **Size** | **Clamp Load, lbs.** | **Dry Bolts** | **Lubricated Bolts** |
| $^{1}/_{4}$-28 (.250) | 3,263 | 14 ft. lbs | 10 ft. lbs |
| $^{5}/_{16}$-24 (.3125) | 5,113 | 27 ft. lbs | 20 ft. lbs |
| $^{3}/_{8}$-24 (.375) | 7,875 | 49 ft. lbs | 37 ft. lbs |
| $^{7}/_{16}$-20 (.4375) | 10,650 | 78 ft. lbs. | 58 ft. lbs |
| $^{1}/_{2}$-20 (.500) | 14,400 | 120 ft. lbs. | 90 ft. lbs |
| $^{9}/_{16}$-18 (.5625) | 18,300 | 172 ft. lbs. | 129 ft. lbs |
| $^{5}/_{8}$-18 (.625) | 23,025 | 240 ft. lbs. | 180 ft. lbs |
| $^{3}/_{4}$-16 (.750) | 33,600 | 420 ft. lbs. | 315 ft. lbs |
| $^{7}/_{8}$-14 (.875) | 45,825 | 668 ft. lbs. | 501 ft. lbs |
| 1-12 (1.000) | 59,700 | 995 ft. lbs. | 746 ft. lbs |
| $1^{1}/_{8}$-12 (1.125) | 77,025 | 1444 ft. lbs. | 1083 ft. lbs |
| $1^{1}/_{4}$-12 (1.250) | 96,600 | 2012 ft. lbs. | 1509 ft. lbs |
| $1^{3}/_{8}$-12 (1.375) | 118,350 | 2712 ft. lbs. | 2034 ft. lbs |
| $1^{1}/_{2}$-12 (1.500) | 142,275 | 3557 ft. lbs. | 2668 ft. lbs |

If torque tightening values are not specified, metric coarse pitch **austenitic stainless steel fasteners** (Grades A1, A2, A3, A4 and A5), may be preloaded by tightening them to torque values given in the following table.

For the following table, torque values are calculated using formula T = KDP. Value of K = 0.2 has been used which assumes that the threads are burr free and a good quality lubricant (molybdenum disulfide $MoS_2$) is used. The clamping/preload is calculated at 65% yield stress (0.2% permanent strain $R_{p\,0.2}$). In the table, full stop (.) is used as a decimal marker.

| **Torque, Nm for Austenitic Stainless Steel Metric Coarse Pitch Threaded Fasteners** | | | | |
|---|---|---|---|---|
| Size | Stress Area, mm$^2$ | Class 50 | Class 70 | Class 80 |
| M3x0.5 | 5.03 | 0.41 | 0.88 | 1.18 |
| M4x0.7 | 8.78 | 0.96 | 2.05 | 2.74 |
| M5x0.8 | 14.2 | 1.94 | 4.15 | 5.54 |
| M6x1 | 20.1 | 3.29 | 7.06 | 9.41 |
| M8x1.25 | 36.6 | 7.99 | 17.13 | 22.84 |
| M10x1.5 | 58 | 15.83 | 33.93 | 45.24 |

| | | | | |
|---|---|---|---|---|
| M12x1.75 | 84.3 | 27.62 | 59.18 | 78.90 |
| M14x2 | 115 | 43.95 | 94.19 | 125.58 |
| M16x2 | 157 | 68.58 | 146.95 | 195.94 |
| M18x2.5 | 192 | 94.35 | 202.18 | 269.57 |
| M20x2.5 | 245 | 133.77 | 286.65 | 382.20 |
| M22x2.5 | 303 | 181.98 | 389.96 | 519.95 |
| M24x3 | 353 | 231.29 | 495.61 | 660.82 |
| M27x3 | 459 | 338.33 | 724.99 | 966.65 |
| M30x3.5 | 561 | 459.46 | 984.56 | 1312.74 |
| M33x3.5 | 694 | 625.22 | 1339.77 | 1786.36 |
| M36x4 | 817 | 802.95 | 1720.60 | 2294.14 |
| M39x4 | 976 | 1039.15 | 2226.74 | 2968.99 |

**Tightening of Prevailing Torque Type Nuts**

Usually nuts are free-running, but prevailing torque type nuts have a self-contained feature that causes resistance to nut turning. This resistance is called "prevailing torque". Prevailing torque is the torque required to turn the nut. None of the prevailing torque goes toward tightening/tensioning the bolt.

The rule of thumb for tightening of prevailing torque type nuts is to add the prevailing torque to the recommended torque value when applying torque to them. This is because the prevailing torque doesn't contribute to bolt tightening. It is just friction that needs to be overcome.

ISO 2320 specifies the prevailing torque for metric fasteners whereas ASME B18.16.6 specifies the prevailing torque for imperial (inch series) fasteners.

Following table shows prevailing torque as per ISO 2320 for metric property class 8, 10 and 12 fasteners. The values are applicable to coarse as well as fine threaded fasteners.

The prevailing torques shown in the following table apply for all metal type nuts only. The prevailing torque for non-metallic insert type nuts shall be 50 % of the values shown in the table. In the table, full stop (.) is used as a decimal marker.

| **Prevailing Torque, N m for Metric Coarse and Fine Threaded Prevailing Torque Type Nuts** | | | |
|---|---|---|---|
| Thread *d* | Property Class 8 | Property Class 10 | Property Class 12 |
| M3 | 0.43 | 0.6 | 0.6 |
| M4 | 0.9 | 1.2 | 1.2 |
| M5 | 1.6 | 2.1 | 2.1 |
| M6 | 3 | 4 | 4 |
| M7 | 4.5 | 6 | 6 |
| M8 | 6 | 8 | 8 |
| M10 | 10.5 | 14 | 14 |
| M12 | 15.5 | 21 | 21 |
| M14 | 24 | 31 | 31 |
| M16 | 32 | 42 | 42 |
| M18 | 42 | 56 | 56 |
| M20 | 54 | 72 | 72 |
| M22 | 68 | 90 | 90 |
| M24 | 80 | 106 | 106 |
| M27 | 94 | 123 | 123 |
| M30 | 108 | 140 | 140 |
| M33 | 122 | 160 | 160 |
| M36 | 136 | 180 | 180 |
| M39 | 150 | 200 | 200 |

For other grades of fasteners, please see ISO 2320. You may also see https://amesweb.info.

Following table shows the prevailing torques as per ASME B18.16.6 for imperial (inch series) coarse and fine threaded prevailing torque nylon insert type (grades N2, N5, and N8) and all-metal type (grades A, B, C, F and G) nuts.

| **Prevailing Torque, in.-lb for Imperial Coarse and Fine Threaded Prevailing Torque Type Nuts** | |
|---|---|
| Nut Size | Prevailing Torque, in.-lb |
| No. 4 | 4 |
| No. 6 | 8 |
| No. 8 | 12 |
| No. 10 | 17 |
| No. 12 | 27 |
| ¼ | 40 |
| 5⁄16 | 80 |
| ⅜ | 110 |
| 7⁄16 | 135 |
| ½ | 204 |
| 9⁄16 | 300 |
| ⅝ | 420 |
| ¾ | 540 |
| ⅞ | 840 |
| 1 | 1080 |
| 1⅛ | 1200 |
| 1¼ | 1320 |
| 1⅜ | 1620 |
| 1½ | 1800 |

You can also use your torque wrench to measure prevailing torque for prevailing torque type nut and then add this value to the bolt's recommended torque value.

It may be noted that in a reused nut, prevailing torque may be less than shown in above tables.

Usually you wouldn't add prevailing torque to the torque value published by the equipment manufacturer. The equipment manufacturer has already done this for you. For example, propeller bolts use prevailing torque nuts. The propeller manufacturer is aware of this and takes care of this in the torque values recommended by them. So just go ahead and use the published torque values.

**Manual Torque Tools and Microprocessor-Controlled Torque Tools**

Hydraulic tensioners and bolt heaters make it possible to tighten fasteners without suffering the uncertainties of the torque-preload relationship, but they can be used only on large diameter fasteners. Hence for smaller ones, manual torque tools (torque screwdrivers and torque wrenches) and microprocessor controlled torque wrenches are used.

Mass-production assembly lines (automatic or semiautomatic) require high-speed bolting tools that must be able to detect and respond to torque, turn, and other variables more sensitively and rapidly than is possible for a human operator. Hence mass-production tools are similar to the manual torque tools, but they include torque transducers. If turn is also to be controlled, they include angle transducer also. In view of this, computer or microprocessor control is a must for mass-production tools. (Some suppliers use ‘‘current limit control’’ to control torque indirectly and to save the cost of the torque transducer.)

Microprocessor controlled torque wrenches can control preload to ± 2 to 5 percent if reasonable control is maintained over fastener lubricity.

For information on torque tools and microprocessor controlled torque wrenches please see website of Gedore: https://us.gedore.com.

## Typical Elongation for Common Bolting Materials

Following table shows the amount of stretch (in thousandths of an inch) you might see in various bolts, per inch grip length, if they were loaded to various percentages of their yield strength (y.s.). The modulus of elasticity ($E$) is assumed to be $30 \times 10^6$ psi unless otherwise noted. The modulus of elasticity is the measurement of a material's elasticity, stress ⁄ strain.

| Bolting Material | Loading - % of Yield Strength | | | | |
|---|---|---|---|---|---|
| | **20%** | **40%** | **60%** | **80%** | **100%** |
| ASTM A193 Class 1: B8, B8M, B8C - 30 ksi y.s., $E = 28.5 \times 10^6$ psi | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Monel - 40 ksi y.s. | 0.3 | 0.5 | 0.8 | 1.1 | 1.3 |
| SAE Grade 2 - 55 ksi y.s. | 0.4 | 0.7 | 1.1 | 1.5 | 1.8 |
| SAE Grade 5; A325; ASTM A193 B7 and B16 over 2½ up to 4 in. diameter - 96 ksi y.s. | 0.6 | 1.3 | 1.9 | 2.6 | 3.2 |
| ASTM A193 B7 and B16 2½ in. and under diameter, 105 ksi y.s. | 0.7 | 1.4 | 2.1 | 2.8 | 3.5 |
| SAE Grade 8; A490 - 120 ksi y.s. | 0.8 | 1.6 | 2.4 | 3.2 | 4.0 |
| Inconel 718 - 180 ksi y.s. | 1.2 | 2.4 | 3.6 | 4.9 | 6.1 |
| 4340 steel, RC 47 - 200 ksi y.s. | 1.3 | 2.7 | 4.0 | 5.3 | 6.6 |
| High strength bolt material - 240 ksi y.s. | 1.6 | 3.2 | 4.8 | 6.4 | 8.0 |
| Titanium (6AL4V) - 134 ksi y.s., $E = 17 \times 10^6$ psi | 1.6 | 3.2 | 4.8 | 6.4 | 8.0 |

To obtain desired elongation for a particular metal, read the elongation figure under the appropriate percentage of yield strength and multiply by the grip length in inches. For example, to obtain the expected elongation for an SAE Grade 5 bolt stretched to 80% of yield strength, with a 5 in. grip length, select the appropriate figure, which in this case is 2.6, and multiply by 5. The answer is 0.013 in. (13 thou).

**Rule of Thumb**

The stretch of a fastener can be estimated using the following rule of thumb.

A fastener will stretch 0.001 inch per inch of effective length (grip length plus one nut height) per 30,000 psi stress in steel and 15,000 psi stress in titanium.

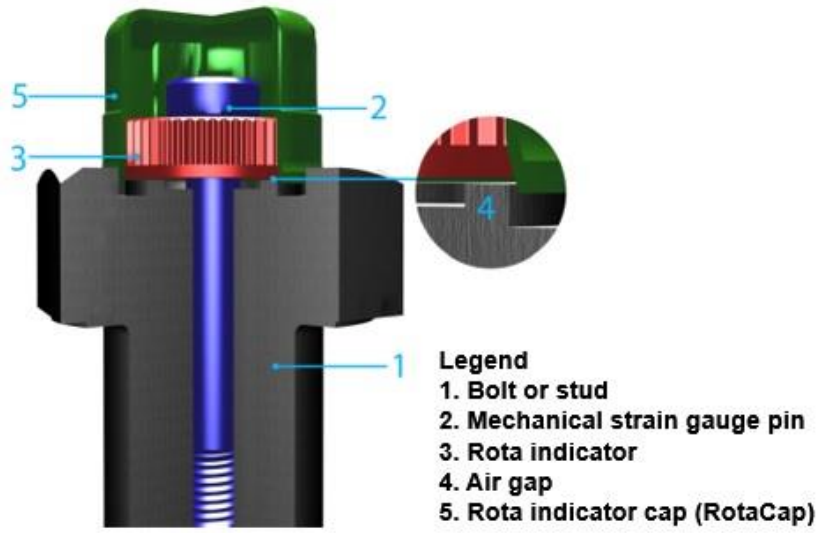
Example: Applying this rule of thumb to SAE Grade 5 (⅝ in. diameter) bolt loaded to 80% yield strength with a grip of 5 inches, gives 0.0138 in. stretch.

Stretch = $(0.001) \times (5 + 0.625) \times [(0.8 \times 92000) / 30000] = 0.0138$ in.

## Commercially Available Gage Bolts

Gage bolts have provision to measure stretch as they are tightened.

One such fastener is the RotaBolt® which measures bolt stretch (extension) by the use of a central gauge pin which passes down a centrally drilled hole in the bolt. RotaBolts are a “state of the art” system where a modified bolt or stud has an indicator cap fitted at the end which locks when the required tensile load is attained.




RotaBolt Construction

Above figure shows construction of a RotaBolt. Conversion of a bolt/stud into a RotaBolt tension measuring fastener involves the drilling and tapping of a small diameter hole along the neutral axis of the fastener. A datum face is also machined on the head of the fastener. A headed steel mechanical strain gauge pin is secured in the drilled and tapped hole along the centre of the bolt/stud. Rota indicator is a small disc, retained on the stem of the mechanical strain gauge pin by a hole through its centre. A datum face is machined on the face of the indicator disc and the circumference of the disc is serrated to engage with the RotaCap. An air gap is created between the datum faces machined on the bolt/stud head and the Rota indicator during load calibration. This gap is set so that when the air gap is closed and the two datum faces are in contact, the RotaCap is no longer free to rotate by finger-and-thumb action. This signifies that the correct tension has been achieved in the bolt/stud. The “thimble-shaped” RotaCap is press fitted onto and over the serrated outer diameter of the Rota indicator enclosing and sealing all the working components of the RotaBolt system. Indentations on the outer surface of the RotaCap enable it to be gripped by finger-and-thumb to test for correct tension in the system.

When a RotaBolt is installed and tightened, the bolt/stud extends fractionally under the applied load but the mechanical strain gauge pin anchored within the body of the bolt/stud does not extend. As a result, as load is applied to the bolt, the datum face of the indicator disc, positioned under the head of the strain gauge pin, is drawn closer to the machined datum face on the head of the bolt/stud, closing the air gap.

Initially the RotaCap, attached to the Rota indicator, can be rotated readily between finger and thumb. However, when the bolt/stud is tightened the strain gauge pin is drawn down into the body of the bolt/stud to a point where the air gap between the datum faces on both the indicator disc and the bolt/stud head close and the two faces are forced into contact, preventing the RotaCap from being rotated by finger-and-thumb action.

When the RotaCap ‘locks’ in this way it indicates the calibrated pre-set tension of the RotaBolt has been reached and the joint is now clamped to within ±5% of the required design load. Should tension be lost across the bolted joint - for example, due to deterioration of a gasket or after thermal cycling - then the two datum faces will move apart re-creating the air gap and making it possible again to rotate the RotaCap by finger-and-thumb. This indicates that remedial action is required and the bolt/stud in question needs re-tightening until the RotaCap locks off and the required tension is once again achieved.

James Walker sales three types of RotaBolts: RB1 Touch, RB2 Touch and RotaBolt® Vision. For more information on RotaBolt® please see the relevant section of the James Walker website: www.jameswalker.biz.


SmartBolt

Direct Tension Indicating (DTI) SmartBolts®, manufactured by Stress Indicators, Inc., are specialty fasteners with a built-in visual indicator embedded in an ordinary bolt that shows the developed tension as the bolt is installed. As shown in the following figure, color of the built-in visual indicator gradually darkens from bright red to black as the fastener is tightened and is completely reversible for the life of the fastener.


Change in Color of Built-in Visual Tension Indicator

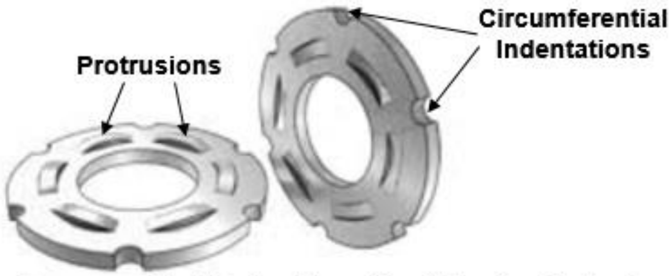
This technology uses color to indicate tension. As shown in the following figure, the changes in color are proportional to bolt stretch, which ensures accurate and reliable measurements.


Color Changes Showing Bolt Stretch

For more information on Smart-Bolts®, please see website of Stress Indicators, Inc.: www.stressindicators.com and website of Industrial Indicators: www.smartbolts.com.

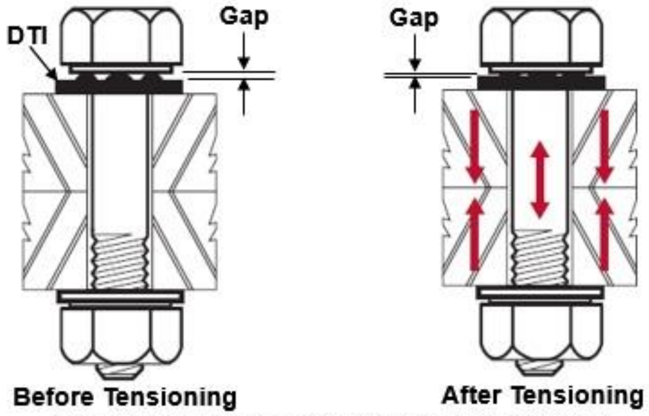
## Compressible-Washer-Type Direct Tension Indicator (DTI)

The compressible-washer-type direct tension indicators, called DTIs have been used in the structural bolting industry for years and is now starting to carry over into engineering fields as well. DTIs are the ideal solution whenever achieving and maintaining bolt tension is critical. As shown in the following figure, a DTI is having protrusions/bumps on one side and circumferential indentations spaced equally around the outside circumference. The circumferential indentations indicate where feeler gauge, if used, is to be inserted during installation inspections, and further make it visually obvious that a direct tension indicator (rather than a flat washer) has been used in the assembly.




Compressible-Washer-Type Direct Tension Indicator

Direct tension indicators are intended for installation under a bolt or cap screw head, a hex nut, or against a hardened washer or other flat hardened surface. As shown in the following figure, the protrusions are flattened as the fastener is tensioned (tightened). A feeler gauge is used to measure the gap developed by the protrusions. When the fastener has developed the appropriate tension, the feeler gauge will no longer fit in the gap. The DTIs are covered under the ASTM designation ASTM F2437 / F2437M.




Reduction in Gap due to Fastener Tensioning

**ASTM F2437 / F2437M**

ASTM F2437 / F2437M: Standard Specification for Carbon and Alloy Steel Compressible-Washer-Type Direct Tension Indicators for Use with Cap Screws, Bolts, Anchors, and Studs covers the requirements for carbon and alloy steel compressible-washer-type direct tension indicators (DTIs) capable of indicating a specified bolt tension in cap screws, bolts, anchors, and studs. Following is covered by the standard specification.

Direct tension indicators in inch sizes ¼ through 2½ in. and metric sizes M6 and M72 are covered.

Direct tension indicators have two styles and four grades for inch fasteners, Grades 5, 8, 55, and 105, and two property classes for metric fasteners, property classes 8.8 and 10.9 each of which differ in their compressive load requirements at a given gap.

Style 1 DTIs are suitable for comparatively smaller bearing surfaces. Style 1 DTIs are available in Grades 5 and 8, which differ in the amount of tension they indicate at a prescribed gap (test gap being 0.010 in. for inch series and 0.250 mm for metric series).

Style 2 DTIs are suitable for comparatively large bearing surfaces. Style 2 DTIs are available in Grades 55 and 105, which differ in the amount of tension they indicate at a prescribed gap (test gap being 0.010 in. for inch series and 0.250 mm for metric series).

Compression load requirements establish the capability of the direct tension indicators to satisfy typical tension requirements for these grades. The user is not obliged to install fasteners and DTIs to these tensions, and is free to specify installation to lower tension values. When so specified, the DTI supplier shall provide a load-gap curve to enable the user to select the appropriate gap criteria for the intended target tension of the application.

The Mean compression load values for Grades 5 and 8 in nominal sizes up through 1½ in are based upon 75% of the proof load for SAE J429 cap screws. For Grade 5 in nominal sizes over 1½ in and up to 2½ in inclusive, the mean compression load values are based on 75% of the proof load for ASTM A449. For grade 8 in nominal sizes over 1½ in and up to 2½ in inclusive, mean compression load values are based upon 75 % of the proof load for ASTM A354BD.

Mean compression load values for Grades 55 and 105 are based upon 60 % of the yield strength for the matching fasteners on which they are used.

The Mean compression load values for property classes 8.8 and 10.9 are based upon 75% of the proof load for ISO 898-1 Bolts/Stud/Screws.

Recommended Fasteners: Fasteners meeting the requirements of the standards referenced in the following table are considered compatible with the DTI grade or class listed.

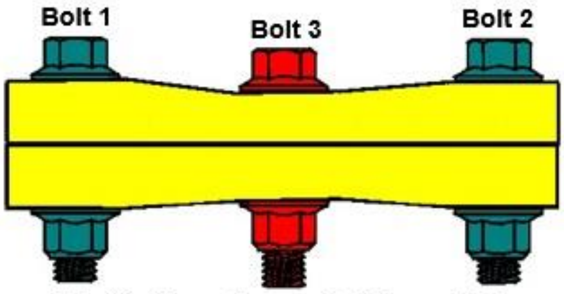
| **Recommended Fasteners** | | | |
|---|---|---|---|
| Series/Grade or Property Class | Recommended Cap Screws, Bolts, Anchors, or Studs[A] | Recommended Nuts[B] | Recommended Flat Washers |
| **Inch Fasteners** | | | |
| Style 2 Grade 55 | Specification F1554 Grade 55<br>Specification A307 | Specification A194/A194M 2H<br>Specification A563 A, C, DH | Specification F436/F436M |
| Style 2 Grade 105 | Specification A193/A193M B7<br>Specification F1554 Grade 105 | Specification A194/A194M 2H<br>Specification A563 DH | Specification F436/F436M |
| Style 1 Grade 5 | Specification A449,<br>Specification A354 BC<br>SAE J429 Grade 5 | SAE J995 Grade 5<br>Specification A563 B, C, D, DH | Specification F436/F436M |
| Style 1 Grade 8 | Specification A354 BD<br>SAE J429 Grade 8 | SAE J995 Grade 8<br>Specification A563 D, DH | Specification F436/F436M |
| **Metric Fasteners** | | | |
| Property Class 8.8 | Specification ISO 898-1 Class 8.8 | Specification ISO 898-2 Class 10 | Specification<br>ISO 887 (300 HV)<br>DIN 125 Part 2 |
| Property Class 10.9 | Specification ISO 898-1 Class 10.9 | Specification ISO 898-2 Class 10 | Specification<br>ISO 887 (300 HV)<br>ISO 7089 |

[A] - Inch bolt and cap screw dimensions are designated in ASME B18.2.1 and metric bolt dimensions are designated in ISO 4014.
[B] - Inch nuts dimensions are designated in ASME B18.2.2 and metric nut dimensions are designated in ISO 4032.

It may be noted that DTIs for engineering applications are designed for the bolts to be tensioned to 75% of the proof load. DTIs for structural applications are designed for the bolts to be tensioned well beyond yield strength. Information about DTIs for structural applications is given in the next chapter.

TurnaSure LLC (www.TurnaSure.com) and TorqBolt Inc. (www.torqbolt.com) are manufacturers / suppliers of DTIs as per ASTM F2437 / F2437M.

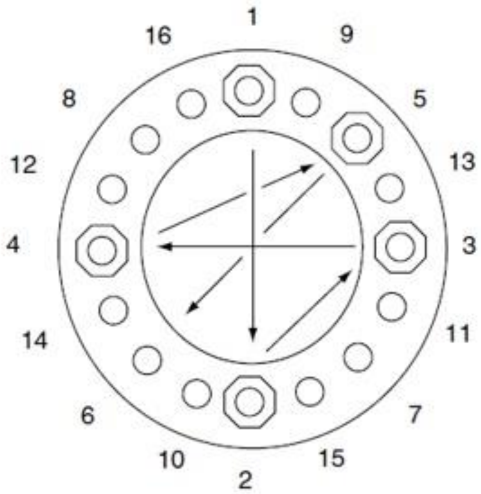
## Bolt Tightening Sequence




Elastic Interaction or Bolt Cross Talk

As shown in above figure, if the middle bolt (Bolt 3) is tightened compressing the joint directly under it after tightening the outer two bolts (Bolt 1 and Bolt 2), the bolt also compresses the joint slightly under the outer two bolts. Due to this, the outer two bolts partially loosen leading to a loss of preload in them.

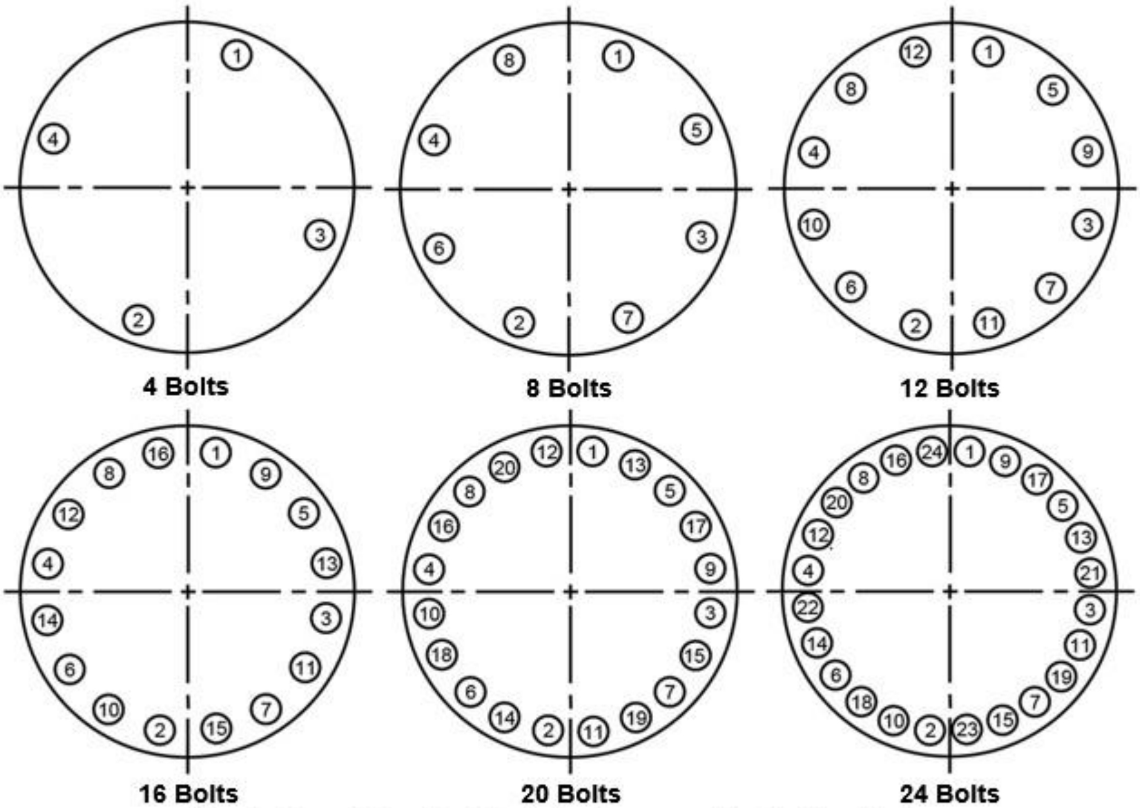
Thus, tightening one fastener will often partially loosen previously tightened fasteners near it. This "cross talk" between fasteners, during assembly or disassembly, is called elastic interaction.




Cross or Star Bolting Pattern for Circular Joints

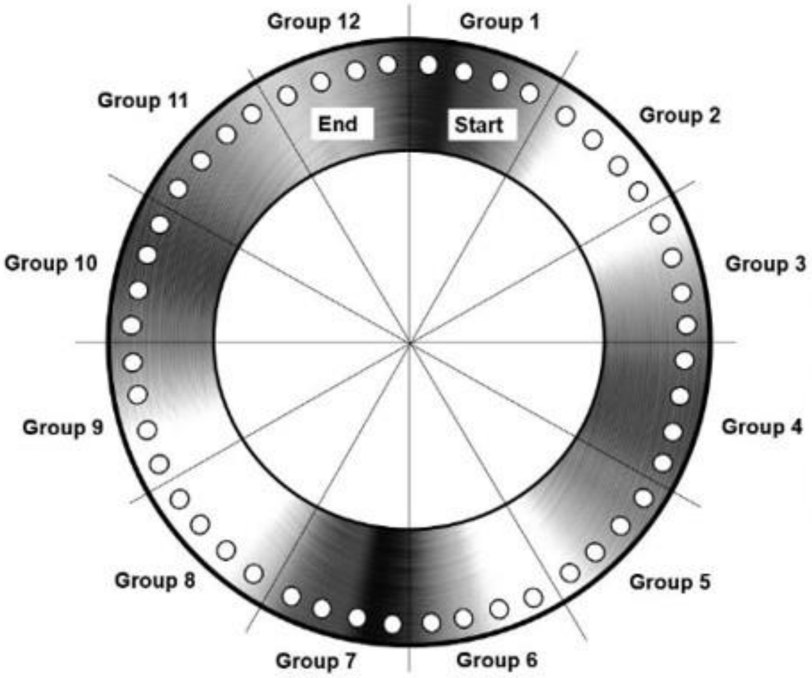
Since in most cases, all bolts of a joint are not tightened simultaneously, tightening of a bolt effects preload in other previously tightened bolts in the group. To minimize this, bolts are tightened as per a sequence based on geometry of the joint. A cross/star pattern is used for circular joints. Above figure shows cross or star bolting pattern for circular (16 Bolt) joints. In the cross or star bolt tightening pattern, after tightening the first bolt one move directly across or 180° for the second bolt, then moves 90° around the circle for the third bolt and then directly across or 180° for the fourth bolt. This pattern is repeated for all other bolts. The numbering should be done by a technician with the relevant experience. The crew members who do the tightening then simply follow the right numerical sequence (bolts 1,2,3,4 etc.).

In general, before commencement of torque tightening, all nuts should be brought up finger tight and then snug tightened evenly. If the joint is critical, it is recommended to consider a multiple pass tightening. Generally, tightening is carried out in three pass, each with approximately 1/3 of total torque value - say first pass with 30%, second pass with 30% and third pass with 40%. In multiple pass tightening, each bolt is tightened more than once so as to reduce the preload reduction caused by the tightening of the other bolts in the joint. After completing the required number of multiple pass tightening, it is recommended to continue torquing the nuts (with 100% of total torque) in a clockwise tightening sequence (rotational round) starting with bolt number 1 until no further rotation of the nut is observed.



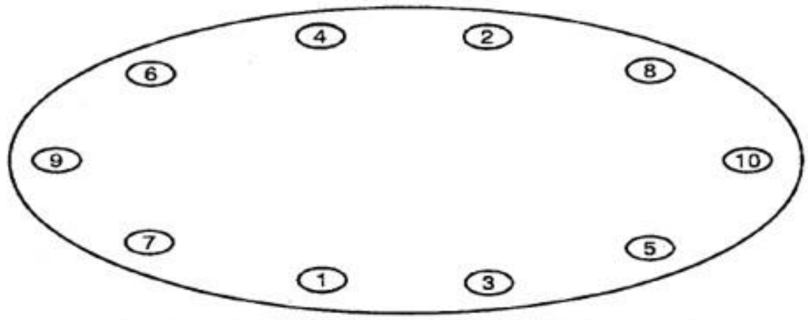
**Numbering of Circular Joints for Cross or Star Bolting Pattern**

For ready reference, above figure shows numbering of circular joints for tightening them as per cross or star bolting pattern.



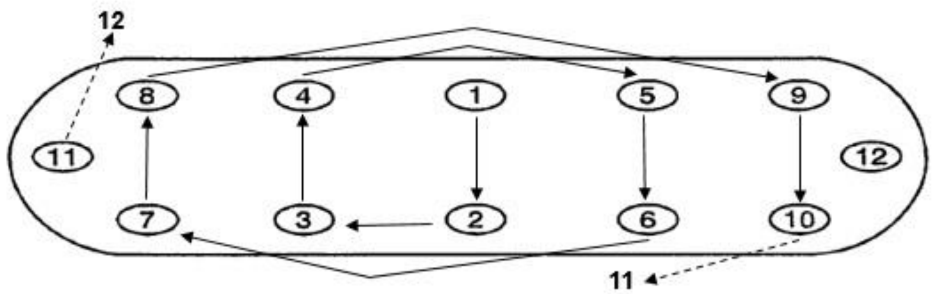
**Bolt Grouping of 48-Bolt Flange**

In the case of very large flanges such as on 48-bolt flanges (and larger), the process of bolts tightening can be different as shown in above figure. Here, groups of 4 adjacent bolts can be treated as ‘1 bolt’, resulting in 12 groups. For example, tightening 4 bolts of group 1 before moving onto group 7 - (Similar procedure to what you would do when tightening a 12-bolt flange).



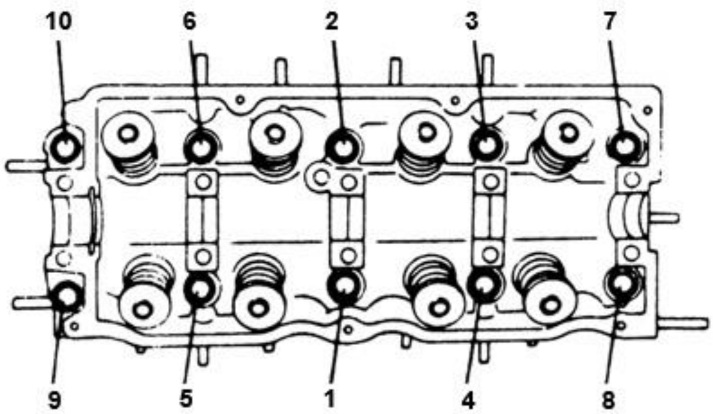

**Tightening Sequence for Non-Circular Multi-Bolt Joint**

Above figure shows bolts tightening sequence for non-circular multi-bolt joints.



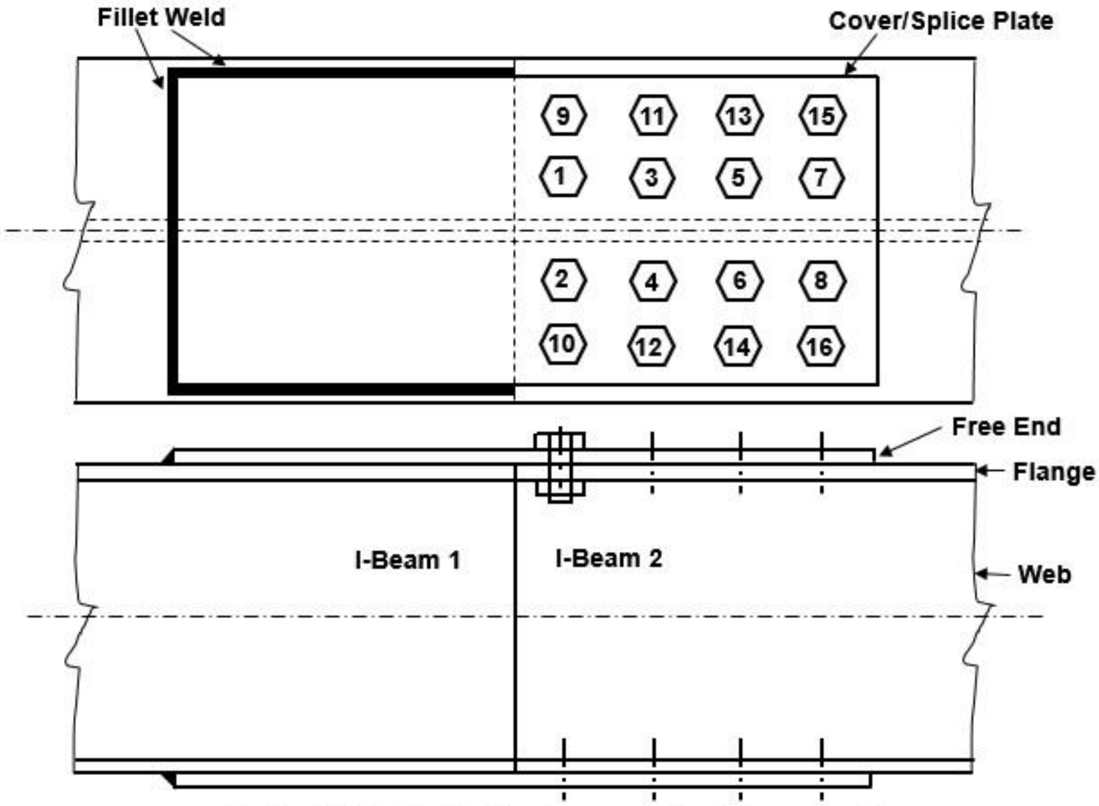

**Tightening Sequence for Obround Joint Using Spiral Pattern**

Above figure shows bolts tightening sequence for obround or rectangle joint (for example engine cylinder head as shown in the following figure) using spiral pattern.




**Engine Cylinder Head Bolt Tightening Sequence**

In a structural joint with a rectangular pattern, with several rows of bolts, the joint tightening shall progress systematically from the most rigid part of the joint, that is, to start tightening at the center of the bolt pattern and work the way out to the free edges.



Typical Bolts Tightening Sequence for Column Joints

Above figure shows typical bolts tightening sequence in a structural joint - erection joints in columns with structural bolts sometimes also called high strength friction grip (HSFG) bolts.

## Situations Affecting Preload

Please take care of the following situations while preloading fasteners as they will consume energy (tightening torque) but will not result in preloading them.

If there is interference between holes and fasteners, while tensioning the joint, some of the tightening torque will not end up in preloading fasteners (preload) because part of it will be lost in friction as the bolt fights its way past the walls of the hole.

Misalignment between the holes of upper and lower joint members could also create a situation similar to the interference between holes and fasteners.

Lifting up of a heavy cover against a flange on (say) a pressure vessel will consume torque to advance the cover up but there will not be any preload between the joint members. Eventually the two joint members will be brought into contact. Further torque, at this point, will be required to create a clamping force between joint members to load the gasket. In such cases, the torque required to lift the heavy cover should be added to the torque required to pull the joint members together to preload the fasteners.

Getting a pipe flange, for example, to mate with the flange of a pump or valve often requires a lot of motion in the flange members. The forces required to align such systems will have the same effect that the force created by the weight in lifting of a heavy cover against flange of a pressure vessel.

## Short-Term Relaxation of Individual Bolts

Often there will be some loss of tension in individual bolts after they are initially tightened. This loss of tension is referred here as ‘‘short term’’ relaxation, to distinguish it from other effects (for an example, due to gasket creep), which will cause further loss of tension over a long period of time. In general, short term relaxation occurs in a bolted joint because something has been loaded past its yield point and will deform to get out from under the excessive load. Following are the common sources which will cause short term relaxation.

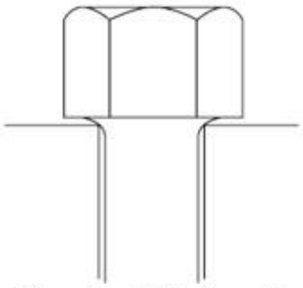
The surfaces of the threads in the nut, the bolt, and the faying surfaces of the joint, washers, etc. are never perfectly flat. Under a microscope one can see a series of hills and valleys. When such parts are first loaded, they contact each other only through high spots on the metal surfaces. Thread dimensions have been selected to support high loads, but only if a significant percentage of the total thread surface shares that load. Since initial total contact area of high spots is relatively small, the metal at the contact points cannot stand the pressures. Plastic deformation occurs at these high spots until enough of the total contact surface has been brought into play to stabilize the situation and support the load without further deformation. The same thing happens in the faying surfaces of the joint, though perhaps to a lesser extent because larger surfaces are involved and initial contact areas are larger.

If the bolt is undersized, or the nut oversized, thread contact areas will be less than those planned by the designer, and substantial plastic deformation may occur.

The length of thread engagement for steel fasteners should be at least 0.8 times the nominal diameter of the fastener. If the engagement length is too short (too few threads support the load), thread contact areas are again smaller than those intended by the fastener manufacturer and excessive relaxation can result.

If fasteners are softer than intended by the designer - perhaps because of improper heat treatment or incorrect material - they may creep and relax substantially even if the geometry is correct and loads are normal.

The contact faces of nuts and bolt heads are never exactly perpendicular to the axis of the threads or to the axis of the bolt hole. This means that only a portion of the contact surface of the nut or bolt head is loaded when we first tighten the fastener. These abnormally loaded surfaces will yield until enough additional contact area has been involved to reduce contact pressures and stabilize the joint.



**Assembly with Oversized Fillets or Undersized Holes**

If the head-to-body fillet contacts the edge of the bolt hole as shown in above figure, the edge of the hole will break down under initial contact pressures. This may result in a complete loss of preload, since such effects are usually large compared to the amount by which the bolt was stretched when it was initially tightened.

In case of oversized holes, there is too little contact between nut and joint surface or between bolt head and joint surface. Unless a washer or something is used to distribute contact pressures and limit contact stresses, the head, nut, or both will embed itself in the joint surfaces. The amount of relaxation will depend on the strength of the surface supporting the nut or washer.

## Gasket Creep

Gaskets have many mechanical characteristics which have an important effect on joint behavior. One important mechanical characteristic is gasket creep or creep relaxation. This occurs when the bolt preload is being developed because during assembly of a joint the gasket is compressed. The change in the thickness of the gasket will depend on its "compressibility" and on the amount of bolt load created during assembly. When the preload (compressive stress) is maintained for a long time, the gasket will slowly continue to get thinner and slightly wider, which allows the bolts' initial preload to relax.

There is no way to avoid gasket creep, because "a good gasket material must have some plasticity to allow it to mate intimately with all the imperfections of the flange surfaces." The amount of creep that will occur depends on the materials it is made of, and its construction. Other factors that can affect gasket creep are initial thickness of the gasket, time and temperature.

- Creep relaxation is relative to the thickness of the gasket. For example, a ¼ inch thick fiber gasket will relax almost 4 times as much as a $\frac{1}{16}$ inch thick gasket made from the same material.
- Most creep relaxation occurs within the first 15-20 minutes after preloading. However, it will also continue to creep slowly for several hours, and maybe even forever.
- High temperatures also increases the amount of creep. The higher the temperature, the higher the amount of creep produced.
- The amount of initial preload place on the gasket also affects its creep rate. The higher the load, the higher the amount of creep produced.

Bolt Stiffness, does not affect a gasket's creep-relaxation properties, it will however affect the relationship between the loss of thickness of the gasket and the loss of clamping force or compressive stress on the gasket.

## Thread Galling on Stainless Steel Fasteners

Galling is one of the most common problems when tightening fasteners. Also known as cold welding (atomic bonding due to high contact stress at room temperature), galling results in damaged threads, broken fasteners, weakened joints and seized bolts.

According to the Industrial Fastener Institute's (IFI) 6th Edition Standards Book, thread galling seems to be the most prevalent with fasteners made of stainless steel, aluminum, titanium, and other alloys which self-generate an oxide surface film for corrosion protection. During fastener tightening, as pressure builds between the contacting and sliding thread surfaces, protective oxides are broken, possibly wiped off, and interface metal high points shear or lock together. This cumulative clogging-shearing-locking action causes increasing adhesion. In the extreme, galling leads to seizing - the actual freezing together of the threads. If tightening is continued, the fastener can be twisted off or its threads ripped out.

The IFI gives the following three suggestions for dealing with the problem of thread galling in the use of stainless steel fasteners:

1. Slowing down the installation RPM speed will frequently reduce, or sometimes solve completely, the problem. As the installation RPM increases, the heat generated during tightening increases. As the heat increases, so does the tendency for the occurrence of thread galling.

2. Lubricating the internal and/or external threads frequently eliminates thread galling. The suggested lubricants should contain substantial amounts of molybdenum disulfide (moly), or graphite. Some proprietary, extreme pressure waxes may also be effective. You must be aware of the end use of the fasteners before settling on a lubricant. Stainless steel is frequently used in food related applications, which may make some lubricants unacceptable. Lubricants can be applied at the point of assembly or pre-applied as a batch process similar to plating.

3. Using different stainless alloy grades for the bolt and the nut reduces galling. The key here is the mating of materials having different hardnesses. If one of the components is 316 and the other is 304 they're less likely to gall than if they're both of the same alloy grade. This is because different alloys work-harden at different rates.

Another factor affecting thread galling in stainless steel fastener applications is thread roughness. The rougher the thread flanks, the greater the likelihood galling will occur. In an application where the bolt is galling with the internal thread, the bolt is usually presumed to be at fault, because it is the breaking component. Generally, it is the internal thread that is causing the problem instead of the bolt. This is because most bolt threads are smoother than most nut threads. Bolt threads are generally rolled, therefore, their thread flanks are relatively smooth. Internal threads are always cut, producing rougher thread flanks than those of the bolts they are mating with. The reason galling problems are inconsistent is probably due largely to the inconsistencies in the tapping operation. Rougher than normal internal threads may be the result of the use of dull taps or the tapping may have been done at an inappropriately high RPM. Use of premium fasteners can significantly reduce the risk of galling as they minimize friction (due to normal roughness of internal threads).

One should not use damaged fasteners. A bolt with dented or damaged threads has a significantly increased chance of galling. Dirty bolts with debris in the threads can also greatly increase the risk of galling - so make sure you only use clean bolts.

## Gasket Compression

In a gasketed joint, the gasket load or bolt/clamp force needed is composed of two major parts. First is the force needed to compress and hold the gasket in place. The load generated by the bolts has to compress the gasket to conform to the flange surfaces and “seat” it into the flange. Second is the force needed to maintain tightness when the vessel is pressurized. The flange designer calculates the total force required and suggests torque value to tighten the bolts. In case torque value is not suggested, a gasket may be compressed as per the gasket manufacturer’s recommendation.

Spiral wound gaskets may be compressed to the thicknesses shown in the following table.

| **Required Gasket Compression (as per Flexitallic, website: www.flexitallic.com)** | |
|---|---|
| Initial Gasket Thickness, mm | Recommended Compressed Thickness, mm |
| 1.6 | 1.3/1.4 |
| 2.5 | 1.9/2.0 |
| 3.2 | 2.3/2.5 |
| 4.5 | 3.2/3.4 |
| 6.4 | 4.6/5.1 |
| 7.2 | 5.1/5.6 |

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


Taper Gauge

During the incremental tightening steps, it is good practice to measure the flange gap at a minimum of four points at 90 degrees to each other around the flange OD at each step to assure even loading. When possible it is recommended to use more than four measurement points for flanges larger than 8” in diameter. This can be done easily by using a taper gauge (flange gap measurement tool) or Vernier calipers.

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

## General Assembly Procedure

Many surveys show that failures of bolted assemblies are mainly due to the fact that they were not properly designed (analysis, calculation, choice of components) or implemented (tightening method, tooling, checking).

The surveys also show that among the possible causes of assembly failure (overloading, improper design, manufacturing defects etc.) the most frequent is poor assembly. Under tightening, over tightening and irregular tightening alone cause 30% of all assembly failures.

Thus equally important to the design of a bolted joint in determining performance is the assembly procedure. If the joint is not properly assembled, it will not perform as intended.

The following are guidelines for bolting an assembly.

Train the bolting crews. Explain why good work practices are important. Warn the crews of problems that will be encountered if procedures are not followed. Training improves bolting results.

Supervise the work, especially on critical joints.

Clean the mating surfaces with a suitable solvent and wire bristle brush. Use stainless steel bristles on stainless steel components. Inspect the seating surface for defects such as burrs, damage, corrosion, etc. and take corrective action if required.

Wire brush fasteners if they are dirty and rusted. Use stainless steel bristles on stainless steel fasteners. Inspect the threads for burrs and damage. Chase threads with a tap or die if they’re damaged. Or replace them with new ones.

Coat the fasteners and the bearing surface of the turned element (the nut or the bolt head) with an approved clean lubricant.

If torque is used for assembly, it is recommended to use hardened washers between the turned element and the joint surface. Although hardened washers are not mandatory, they are helpful.

Run the nuts down by hand. If you can't, the threads may need to be cleaned or chased.

Align the joint members before tightening the bolts. Snug the joint first with a modest torque before tightening it.

It is recommended to use a multiple pass, cross bolting procedure for tightening the joint.

In case of torque tightening, if possible, develop your own nut factors, experimentally, rather than relying on a table.

In case of torque tightening, apply torque at a uniform rate.

Hold torque wrenches perpendicular to the axis of the bolt while torque is being applied.

Keep tools in good condition. Tool repairs waste time and are counterproductive. Calibrating the tools periodically ensures that they perform as required.

Keep good records of the tools, operators, procedures and lubricants used.

**Note**

The bolts may be reused provided you are absolutely certain that earlier they were not tightened beyond yield point.

# Tightening of Structural Bolts

Use of high strength bolts (for example, bolts as per ASTM F3125/F3125M grades A325, F1852, A490, and F2280) in structural connections is fast gaining popularity over the low carbon steel bolts (for example, bolts as per ASTM A307, Grade A. ASTM A307 bolts are manufactured with a hexagonal head and nut, either a regular or heavy head, depending on the bolt diameter. In application, A307 bolts are tightened so that some axial force is present that will prevent movement of the connected members in the axial direction of the bolt.). These structural connections which could be either bearing type or friction type have rigidity or continuity comparable with what is achievable in welded construction. The strength of the joint fabricated by means of these bolts is obtained by bearing or friction (grip) developed as a result of very high initial tension in the bolts produced by tightening the nuts to the specified bolt tension. Information about important design considerations and installation/tightening of these bolts is given in this chapter.

For simplicity, henceforth in this chapter ASTM F3125 grade A325 will be referred as grade A325 and ASTM F3125 grade A490 as grade A490.

## Structural Bolts vs. Standard Hex Bolts

As we have seen in the previous chapter, standard hex bolts (for example, SAE J429 grade 5 and grade 8 bolts) used in engineering applications are preloaded to 75% of their proof load (below yield point) because they are required to be reused after their disassembly. The structural bolts on other hand are preloaded to a minimum of 70% of the tensile strength and the bolts are likely be loaded above the yield point after assembly. The American Institute of Steel Construction (AISC) maintains that high bolt preloads are desirable for three reasons:

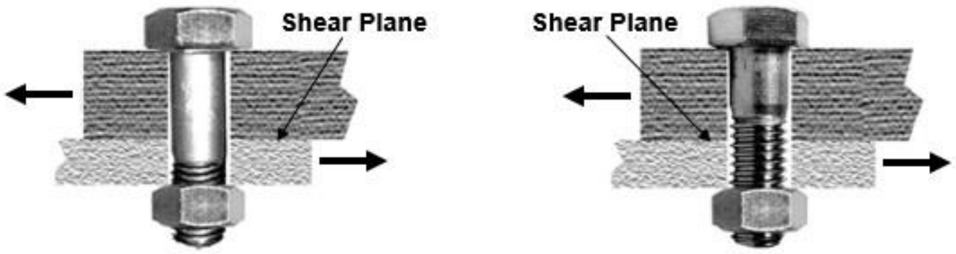
1. High bolt preloads increase joint rigidity.
2. High bolt preloads result in better stress patterns in the connected plies.
3. High bolt preloads provide security against loosening.

The AISC specification for structural joints using grade A325 or A490 bolts mandates minimum preload values of 70% of the ultimate tensile strength of the bolt. No maximum preload is mandated. If the bolt does not fail on installation, it will not fail in service. On the other hand, if a bolt breaks during tightening, it is simply replaced.

In view of above, a structural bolt which has been fully tensioned (loaded above the yield point) should not be reused. If a bolt has been tensioned and then has to be removed it must be marked accordingly and destroyed.

When looking at the mechanical requirements of bolts it appears that a grade A325 and SAE J429 grade 5 are identical as do the grade A490 and the SAE J429 grade 8. Similarly, metric property classes 8.8 and 10.9 correspond closely to grade A325 and grade A490 bolts respectively. However, we cannot substitute SAE J429 grade 5 or metric property class 8.8 when a grade A325 is specified and SAE J429 grade 8 or metric property class 10.9 when a grade A490 is specified for the following reasons.

Structural bolts (grade A325 and A490 bolts) are produced with a heavy hex head configuration which provides a wider bearing surface over which to distribute the load. For the same reason that the heads of the bolts are larger, these bolts are designed to be used with heavy hex nuts that conform to ASTM A563 or ASTM A194. Standard hex bolts (SAE J429 grade 5 and 8 bolts) are produced to standard hex cap screw configuration and therefore cannot distribute the load as much as is needed for structural applications.




Shear Through Shank in Structural Bolt    Shear Through Thread in Standard Hex Bolt

Structural Bolt and Standard Hex Bolt Subjected to Shear

Structural bolts have longer shank lengths (non-threaded portion of the body) and, thus, shorter thread length when compared with standard hex bolts of equivalent overall length. This can provide greater shear strength by allowing only the larger shank subjected to shear as shown in above figure.

## Bolting Components for Structural Connections

As per Specification for Structural Joints Using High-Strength Bolts by the Research Council on Structural Connections (RCSC) following bolting components and assemblies are used for structural connections.

### Group Designations as per RCSC

The following table shows three group designations for bolts and matched bolting assemblies by tensile strength levels of bolts.

A matched bolting assembly is a bolting assembly made of components that are supplied and tested by the manufacturer or supplier in controlled lots as an assembly.

| Group Designations for Bolts and Matched Bolting Assemblies | | | |
|---|---|---|---|
| Group | Tensile Strength | Bolts | Matched Bolting Assemblies |
| Group 120 | 120 ksi | ASTM F3125 Grade A325 | ASTM F3125 Grade F1852 |
| Group 144 | 144 ksi | - | ASTM F3148 Grade 144 |
| Group 150 | 150 ksi | ASTM F3125 Grade A490 | ASTM F3125 Grade F2280 |

### Heavy Hex Structural Bolts

Group 120 and 150 heavy hex structural bolts shall meet the requirements of ASTM F3125 Grades A325 and A490, respectively.

### Heavy Hex Nuts

Heavy hex nuts shall meet the requirements of ASTM A563. The permitted grade of such nuts shall be as given in the following table.

| Permitted Nut Grades | | | |
|---|---|---|---|
| Group Designation | Bolt Type | Coating* | ASTM A563 Nut Grade |
| 120 | 1 | Plain | C, C3, D, DH, and DH3 |
| | | Coated | DH |
| | 3 | Plain | C3 and DH3 |
| 144 and 150 | 1 | Plain | DH and DH3 |
| | | Coated | DH |
| | 3 | Plain | DH3 |

* - for coating compliance, please see the specification by RCSC.

ASTM A194 Grade 2H nuts are permitted as substitutes for ASTM A563 Grade DH nuts.

**Spline End Matched Bolting Assemblies**

Spline end fixed matched bolting assembly is a matched bolting assembly with a spline end that is to remain attached to the bolt once the installation is complete.

Spline end twist-off matched bolting assembly is a matched bolting assembly with a spline end that is to be sheared off by the installation wrench when using the twist-off tension control bolt method for installation.

Group 120 and 150 spline end twist-off matched bolting assemblies shall meet the requirements of ASTM F3125 Grade F1852 and Grade F2280, respectively.

Group 144 spline end fixed matched bolting assemblies shall meet the requirements of ASTM F3148 Grade 144.

**Washers**

Flat circular washers and beveled washers shall meet the requirements of ASTM F436.

**Washer-Type Indicating Devices**

Direct tension indicator is a washer-shaped device incorporating small arch-like protrusions on the bearing surface that are designed to deform in a controlled manner when subjected to a compressive load.

Compressible-washer-type direct tension indicators (DTI) shall meet the requirements of ASTM F959. The type of direct tension indicators shall be as given in the following table.

| **Permitted Materials for Direct Tension Indicators** | | |
|---|---|---|
| Group Designation | Bolt Type | DTI Type |
| Group 120 | 1 | ASTM F959, Type 325-1 |
| | 3 | ASTM F959, Type 325-3 |
| Group 144 | 1 | ASTM F959, Type 490-1 |
| | 3 | ASTM F959, Type 490-3 |
| Group 150 | 1 | ASTM F959, Type 490-1 |
| | 3 | ASTM F959, Type 490-3 |

**Bolt Length**

Grip is the total thickness of material a bolt passes through, exclusive of washers or direct tension indicators.

The bolt length used shall be such that, when installed, sufficient thread engagement is achieved. Following table may be used for selecting the bolt length.

| **Bolt Length Selection** | |
|---|---|
| Nominal Bolt Diameter, $d_b$, in. | To Determine the Required Bolt Length, Add to *Grip + Washer + Direct Tension Indicator*, in. |
| ½ | 11/16 |
| ⅝ | ⅞ |
| ¾ | 1 |
| ⅞ | 1⅛ |

| | |
|---|---|
| 1 | 1¼ |
| 1⅛ | 1½ |
| 1¼ | 1⅝ |
| 1⅜ | 1¾ |
| 1½ | 1⅞ |

**Storage and Lubrication**

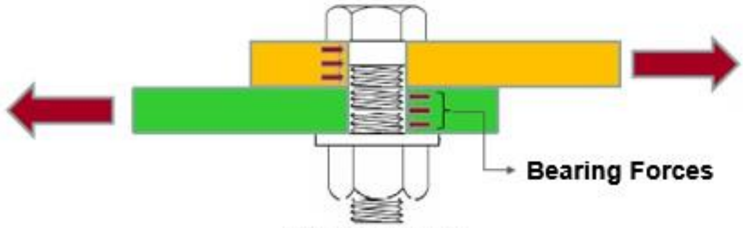
Once received at the installation site, bolting components and bolting assemblies shall be kept in protected storage.

## Joint Types

An assembly of one or more joints that is used to transmit forces between two or more members is called a connection. The area of a connection in which one weld or one group of bolting assemblies joins two or more members or connection elements is called a joint. In a connection the contact surface between two connected elements is called a faying surface.

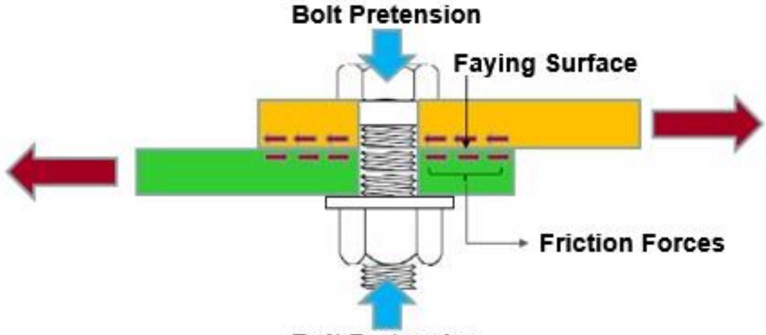
Two basic types of joints are covered by the American Institute of Steel Construction (AISC) specification for structural applications. They are shear joints and tension joints.

Shear joints resist loads at right angles to the bolt axes. The threads of a bolt may either be included in the shear plane or excluded from the shear plane. The capacity of a bolt is greater with the threads excluded from the shear plane. Shear joints are subdivided into two types: bearing joints and friction joints also known as slip-critical joints.




Bearing Joint

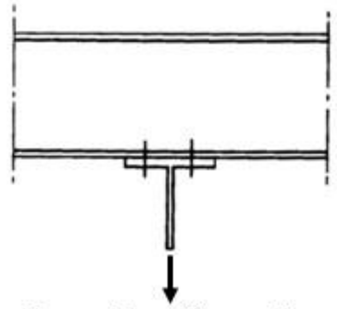
In a bearing joint, shown in above figure, the load is transferred between members by bearing on the bolts. The connected elements are assumed to slip into bearing against the body of the bolt. If the joint is designed as a bearing joint the load is transferred through bearing whether the bolt is installed snug-tight or pretensioned. The resistance to slip in bearing joints is provided by the bolts acting as pins. The bolts carry the load in shear and bear across the thickness of the joint plies.




Slip-Critical Joint

As shown in above figure, in slip-critical joints, the load is transferred between the members by friction in the joint at the joint's faying surface. In slip-critical joints, the bolts are highly pretensioned to cause a clamping force between the connected elements, which develop frictional resistance called friction forces between them. The friction forces allow the joint to withstand loading without slipping into bearing against the body of the bolt, although the bolts must still be designed for bearing. Frictional forces depend on high bolt pretension and the coefficient of friction between the joint surfaces. Hence, the faying surfaces in slip-critical joints require special preparation.

Thus shear joint (sometimes also called bearing joint) is snug-tightened joint or pretensioned joint with bolts that transmit shear loads and for which the design criteria are based upon the shear strength of the bolts and the bearing strength of the connected materials.



**Hanger Type Connection (A Tension Joint)**

A tension joint carries the load which is applied parallel to bolt axes. A hanger type connection, shown in above figure is one of the few examples where mechanical fasteners are used in direct tension. A tension joint is snug-tightened (for static loading only) or pretensioned (all other conditions of tension loading, for example, cyclic loads).

According to the Research Council on Structural Connections (RCSC) and American Institute of Steel Construction (AISC), for structural applications there are generally three types of joints in which a bolt is used: snug-tightened joints, pretensioned joints, and slip-critical joints.

**Snug-Tightened Joints**

Snug-tightened joint is a joint in which the bolting assemblies have been installed to the snug-tight condition.

Snug-tight condition is the joint condition in which the plies have been brought into firm contact and each bolting assembly has at least the tightness attained with either a few impacts of an impact wrench, or the full effort of an ironworker using an ordinary spud wrench.

**Spud Wrench**

Spud wrench, also called podger spanner, shown in above figure is a tool incorporating a wrench at one end and taper on the other end. The taper/pointed end is used to align the bolt holes while the wrench end is used to tighten the nuts.

**Pretensioned Joints**

Pretensioned joint is a joint that transmits shear and/or tensile loads in which the bolts have been installed in accordance with a specification (for example, RCSC specification) to provide a minimum specified pretension in the installed bolt.

Pretension is a level of tensile force achieved in a bolting assembly through its installation, as required for pretensioned and slip-critical joints.

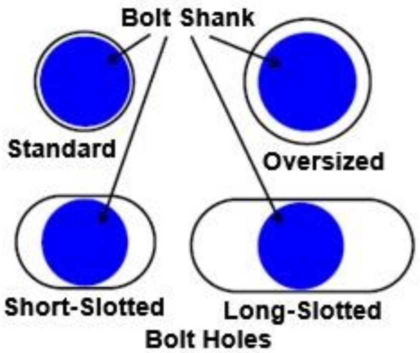
**Slip-Critical Joints**

Slip-critical joint is a joint that transmits shear loads or shear loads in combination with tensile loads in which the bolting assemblies have been installed in accordance with a specification (RCSC specification) to provide a pretension in the installed bolt (clamping force on the faying surfaces), and with faying surfaces that have been prepared to provide a calculable resistance against slip.

Slip-critical joints are required in the following applications involving shear or combined shear and tension:

1. Joints that are subject to fatigue load with reversal of the loading direction;
2. Joints that utilize oversized holes;
3. Joints that utilize slotted holes, except those with applied load approximately perpendicular to the direction of the long dimension of the slot; and
4. Joints in which slip at the faying surfaces would be detrimental to the performance of the structure.

## Bolt Holes



Bolt Holes

As shown in above figure, for high-strength bolts; standard, oversized, short-slotted, and long-slotted holes are used. For their nominal dimensions, please see RCSC specification.

## Use of Washers

Washers are not required in snug-tightened joints, except:

- When the outer face of the joint has a slope that is greater than 1:20 (3°) with respect to a plane that is normal to the bolt axis, an ASTM F436 beveled washer shall be used to compensate for the lack of parallelism.
- When a slotted hole occurs in an outer ply, an ASTM F436 washer or $\frac{5}{16}$ in. thick common plate washer shall be used as required to completely cover the hole.

Washers are not required in pretensioned joints and slip-critical joints, except:

- When Group 144 or Group 150 bolts (as per RCSC specification) are pretensioned in connected material with specified minimum yield strength less than 40 ksi, ASTM F436 washers shall be used under both the bolt head and nut, except that a washer is not needed under the head of an ASTM F3125 Grade F2280 round head bolt or an ASTM F3148 Grade 144 round head bolt.
- When the calibrated wrench method for pretensioning is used, an ASTM F436 washer shall be used under the nut.
- When the twist-off tension control bolt method for pretensioning is used, an ASTM F436 washer shall be used under the nut as part of the bolting assembly.
- When the combined method for pretensioning is used, an ASTM F436 washer shall be used under the nut.
- When the direct tension indicator method for pretensioning is used, and the direct tension indicator is located under the turned element, an ASTM F436 washer shall be used between the turned element and the direct tension indicator.
- When an oversized or slotted hole occurs in an outer ply, the washer requirements shall be as given shown in the following table. The washer used shall be of sufficient size to completely cover the hole.

<table>
<tr><th colspan="5">Washer Requirements for Pretensioned and Slip-Critical Bolted Joints<br>with Oversized and Slotted Holes in the Outer Ply</th></tr>
<tr><td rowspan="2">Bolt Group<br>(RCSC specification)</td><td rowspan="2">Nominal Bolt<br>Diameter, $d_b$, in.</td><td colspan="3">Hole Type in Outer Ply</td></tr>
<tr><td>Oversized</td><td>Short-Slotted</td><td>Long-Slotted</td></tr>
<tr><td>Group 120</td><td>½ - 1½</td><td colspan="2" rowspan="2">ASTM F436</td><td rowspan="2">5/16 in. thick plate washer or continuous bar[a,b]</td></tr>
<tr><td rowspan="2">Group 144 and 150</td><td>≤ 1</td></tr>
<tr><td>> 1</td><td colspan="2">ASTM F436 extra thick[a,c]</td><td>ASTM F436 washer with either a 3/8 in. thick plate washer or continuous bar[a,b]</td></tr>
</table>

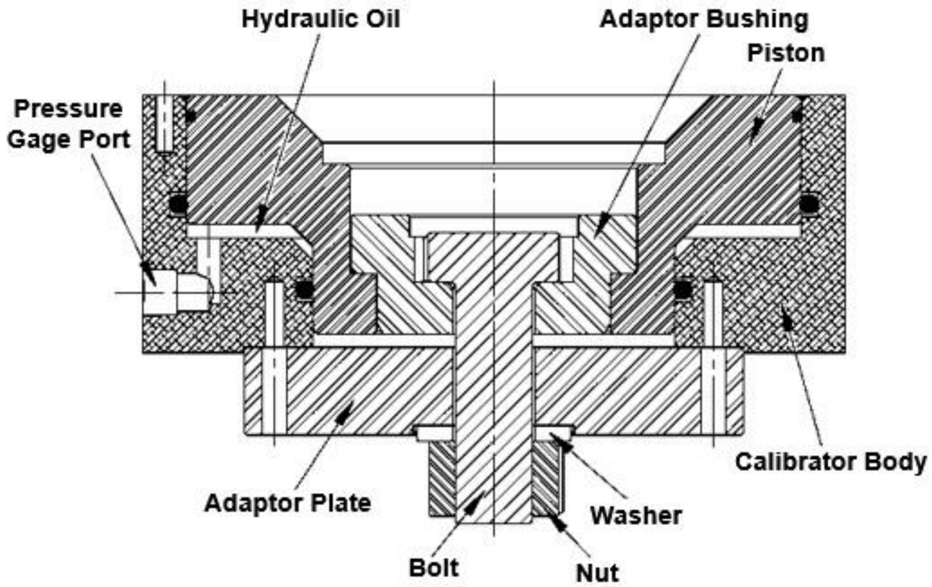
[a] - Multiple washers with a combined thickness of 5/16 in. or larger do not satisfy this requirement.
[b] - The plate washer or bar shall be of structural-grade steel material, but need not be hardened.
[c] - Alternatively, a 3/8 in. thick plate washer and an ordinary thickness F436 washer may be used. The plate washer need not be hardened.

## Bolt Tension Measurement Device

**Bolt Tension Measurement Device**

Bolt tension measurement device is a calibrated device that is used to verify that the bolting assembly, the pretensioning method, and the tools used are capable to achieve the required tensions when a pretensioned joint or slip-critical joint is specified.

The accuracy of the bolt tension measurement device shall be confirmed through calibration at least annually.



Typical Bolt Tension Measurement Device Cross-Section

As shown in above figure, a bolt tension measurement device, also called calibrator is an oil filled hydraulic load cell with a hole through the center of the piston & body. A sample bolt, nut, and washer, as shown in the figure, are installed thru the piston hole using the appropriately sized adaptors. As the nut is tightened, the bolt stretches and the piston applies a compressive force to the hydraulic oil contained between the piston and calibrator body. This force causes a pressure increase in the hydraulic oil that is proportional to the increase in bolt tension. A specially calibrated gage measures the pressure and provides a reading of equivalent tensile force developed in the bolt.

Skidmore-Wilhelm are the leading manufacturer of bolt tension measurement devices. For more information on them, please see www.skidmore-wilhelm.com.

## Connected Plies and Faying Surfaces

Unless otherwise approved by the designer, all connected plies in a joint that are within the grip of the bolt and any materials that are used under the bolt head or nut shall be steel.

The slope of the surfaces of parts in contact with the bolt head and nut shall be equal to or less than 1:20 (3°) with respect to a plane that is normal to the bolt axis.

Faying surfaces and surfaces adjacent to the bolt head and nut shall be free of dirt and other foreign material.

The faying surfaces of slip-critical joints, including those of filler plates and finger shims, shall meet the following requirements:

Uncoated faying surfaces shall be free of scale (except tight mill scale), coatings, and overspray (in areas closer than one bolt diameter but not less than 1 in. from the edge of any hole, and in all areas within the bolt pattern), or shall be blast cleaned prior to assembly.

Coated faying surfaces shall first be blast cleaned and subsequently coated with a coating that is qualified in accordance with the specification.

Galvanized faying surfaces shall be hot-dip galvanized in accordance with the requirements of ASTM A123. Power or hand wire brushing of the surfaces is not permitted.

## Bolts Installation (Bolts Tightening)

It is important to note that adequate bolt tightening requires proper handling and storage of all components, as well as following appropriate tightening procedures regardless of the method being used.

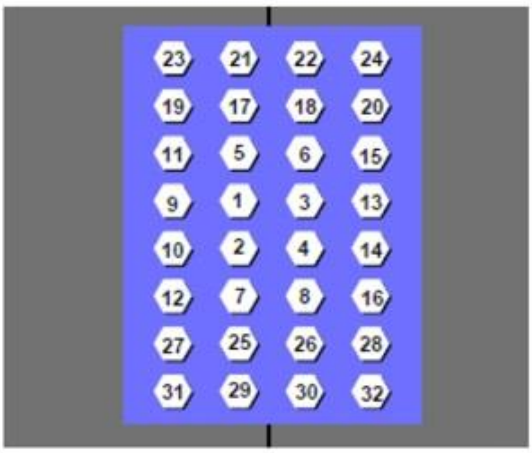
### Snug-Tightened Joints

All bolt holes shall be aligned to permit insertion of the bolts without undue damage to the threads. Drifting to align the bolt holes shall be done in such a way as not to bend or damage the parts nor enlarge the holes.

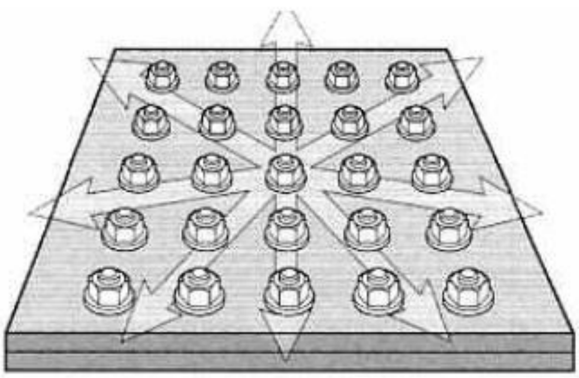
Bolts shall be placed in all holes with washers positioned as required and nuts threaded to complete the assembly.

Compacting of the joint shall progress systematically from the most rigid part of the joint and installed to the snug-tight condition with sufficient thread engagement.

In snug-tightened joints the bolts are tightened (by a few hits of an impact wrench or the full effort of an ironworker using an ordinary spud wrench) so that the nuts cannot be turned without the use of a wrench.




Typical Sequence of Bolts Installation



Installation from Most Rigid Part to Free Edge

Above figure shows typical sequence of bolts installation from the most rigid part of the connection to the free edge. More than one cycle through the bolt pattern may be required to achieve the snug-tightened condition.

It may be noted that the tightening progress from the center of the splice to the outside of the splice, in the horizontal and vertical direction.

**Pretensioned Joints and Slip-Critical Joints**

Installation beyond snug-tight condition is called pretensioning.

AISC and RCSC specifications for structural joints mandates minimum preload values of 70% of the ultimate tensile strength of the bolts for the bolts used in pretensioned or slip-critical joints.

In accordance with the RCSC specification, minimum bolt pretension (clamping force) shall be as per the following table.

| **Minimum Bolt Pretension for Pretensioned Joints and Slip-Critical Joints** | | |
|---|---|---|
| Nominal Bolt Diameter, $d_b$, in. | Minimum Bolt Pretension, kips | |
| | Group 120 | Group 144 and Group 150 |
| ½ | 13 | 16 |
| ⅝ | 20 | 25 |
| ¾ | 29 | 37 |
| ⅞ | 41 | 51 |
| 1 | 54 | 67 |
| 1⅛ | 67 | 84 |
| 1¼ | 85 | 107 |
| 1⅜ | 102 | 127 |
| 1½ | 124 | 155 |

The required pretensions for bolts above 1” in diameter can be difficult to attain. For this reason, fully tensioned bolts larger than 1” in diameter are not often used.

Due to the uncertainties involved with torque, it is not valid to use published values based on a torque-tension relationship from a formula to obtain the accuracy required for structural connections requiring a minimum clamping force of 70% of the ultimate tensile strength of the bolt.

There are five methods of installation procedures recognized by the RCSC:

1. Turn-of-nut method
2. Calibrated wrench method
3. Twist-off tension control bolt method
4. Direct tension indicator method
5. Combined method

As per RCSC, for Group 120 or 150 bolting assemblies, any one of the above five pretensioning methods may be used while for ASTM F3148 Grade 144 matched bolting assemblies, the combined method pretensioning (Sr. No.5) shall be used.

In all above five methods of installation, pretensioning should be carried out progressing systematically from the most rigid part of the joint in a manner that will minimize relaxation of previously pretensioned bolting assemblies.

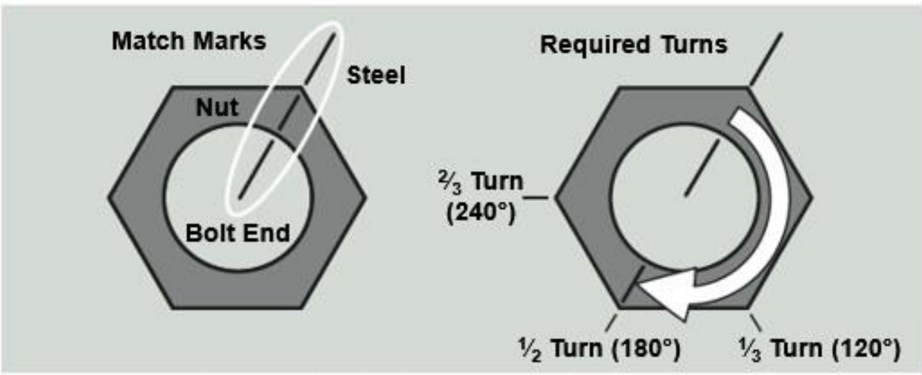
For all methods, the part not turned by the wrench shall be prevented from rotating during pretensioning. When it is impractical to turn the nut, pretensioning by turning the bolt head is permitted while rotation of the nut is prevented, provided that the washer requirements are met and the calibrated wrench method of pretensioning is not used. Upon completion of the pretensioning, it is not permitted to turn the nut or the head in the loosening direction.

In large splice connections or connections with many plies, it may be necessary to use drift pins to bring connections into alignment or temporary bolts to bring all plies of the connection into firm contact. If temporary bolts are used, the remaining holes should be filled with permanent bolts fully tensioned before replacing temporary bolts with permanent ones.

The pre-installation verification procedures specified in RCSC specification shall be performed using bolting assemblies that are representative of the condition of those that will be pretensioned in the work. For information on pre-installation verification procedures, please see the RCSC specification.

## Turn-of-Nut Method Pretensioning

Turn-of-nut method is based on the application of a specific minimum elongation as a means of controlling high-strength bolt pretension.




**Match Marks on Nut, Bolt and Steel Surface (Turn-of-Nut Installation)**

Turn-of-nut pretensioning involves several steps:

1. The bolts are snug-tightened
2. Match marks are placed on each nut, bolt, and steel surface in a single straight line (though match marking is not a RCSC requirement, it allows for an easy visual inspection after final tightening)
3. The part not turned by the wrench is prevented from turning
4. The bolts are tightened with a prescribed rotation past the snug-tight condition

The specified nut rotation varies based on diameter and length of the bolts (between ⅓ and 1 turn) as shown in following table.

| **Nut Rotation from Snug-Tight Condition for Turn-of-Nut Method Pretensioning**[a,b] | | | |
|---|---|---|---|
| | Disposition of Outer Faces of Bolted Parts | | |
| Bolt Length[c] ($d_b$ is nominal diameter of bolt) | Both Faces Normal to Bolt Axis | One Face Normal to Bolt Axis, Other Sloped Not More Than 1:20 (3°)[d] | Both Faces Sloped Not More Than 1:20 from Normal to Bolt Axis[d] |
| Not more than $4d_b$ | ⅓ turn | ½ turn | ⅔ turn |
| More than 4db but not more than $8d_b$ | ½ turn | ⅔ turn | ⅚ turn |
| More than 8db but not more than $12d_b$ | ⅔ turn | ⅚ turn | 1 turn |

[a] - Nut rotation is relative to bolt regardless of the element (nut or bolt) being turned. For all required nut rotations, the tolerance is plus 60 degrees (⅙ turn) and minus 0 degrees.
[b] - Applicable only to joints in which all material within the grip is steel.
[c] - When the bolt length exceeds $12d_b$, the required nut rotation shall be determined by actual testing in a suitable bolt tension measurement device.
[d] - Beveled washer not used.

The actual pretension developed depends on how far the nut is turned as well as how much clamping force was established prior to the turning. A bolt in snug-tight condition will carry no less than 10% of its pretension load.

## Calibrated Wrench Method Pretensioning

Calibrated wrench method uses a torque / impact wrench to tighten the bolt to a specified tension after the snug-tightening operation has been performed. A bolt tension measurement device (calibration device) is used to calibrate the torque / impact wrench to the installation torque level which will achieve the specified tension. It is prohibited to use this method by turning the bolt head.

Since this method is a torque-controlled method, many variables could result in an inadequately tightened bolts. Hence, the calibrated wrench method is valid only if three sample of fastener assemblies' representative of those to be used in the connections are tested daily to verify that the correct tension will be achieved.

## Twist-Off Tension Control Bolt Method Pretensioning

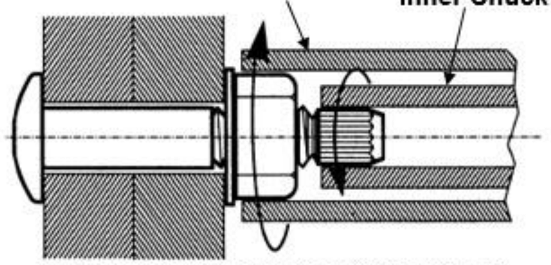
Twist-off tension control bolt method requires the use of spline end twist-off matched bolting assemblies, also called tension-control (TC) bolts and a specially designed wrench (typically electrically powered) that has two coaxial chucks.


Spline End Twist-Off Matched Bolting Assembly

As shown in above figure, spline end twist-off matched bolting assemblies have a splined end that extends beyond the threaded portion of the bolt, a suitable nut and a washer.

In general, two steps are involved during installation: compacting the joint to a snug-tight condition and then systematically twisting off the splines with the special wrench to achieve the prescribed tension.

After the snug-tightening operation is performed, the installer shall verify that the splined end has not been severed, and if this has occurred, the bolting assembly shall be removed and replaced.




Two Coaxial Chucks for Pretensioning

As shown in above figure, for pretensioning, the inner chuck of the wrench engages the splined end of the bolt and the outer chuck of the wrench engages with the nut. The two chucks then turn opposite to one another to tighten the bolt. When the tension is sufficient (specified) in the fastener, the spline end of the bolt simply shears off (twists-off), leaving the tightened bolt correctly installed in the connection. The bolt manufacturer calibrates the bolts so that the spines shears off when the bolt pretension has reached the specified level

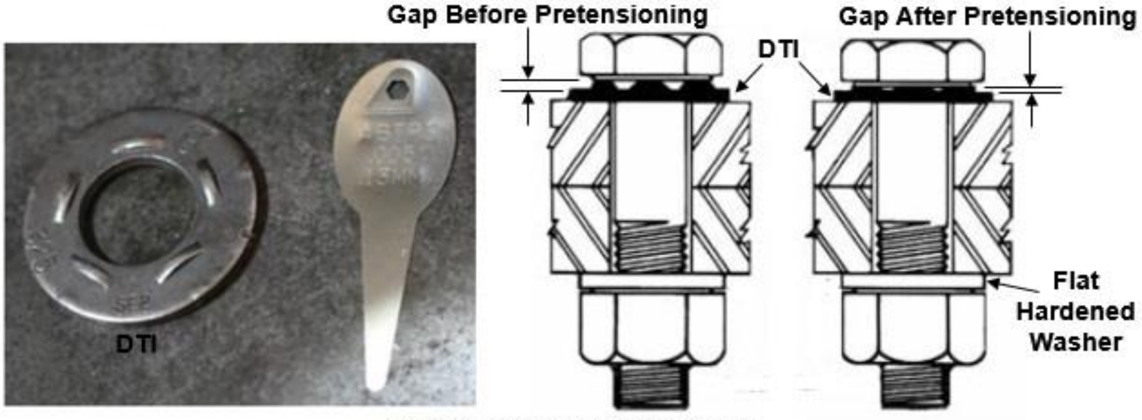
RCSC specification stipulates that because this method of pretensioning is a torque-based system, bolt assemblies must be used in as-received, cleaned, lubricated condition; re-lubrication in the field is not allowed.

Though tension-control (TC) bolts are generally more expensive than conventional high-strength bolts and requires special wrenches for their installation, this method offers an easy one-side installation and a quick visual inspection.

**Caution**

When using the calibrated wrench method or tension control bolts, care must be taken to prevent the nut from running up onto the thread runout portion of the bolt. If this happens, the torque measurement will be erroneous in case of calibrated wrench method or the spine of tension control bolt will shear off without inducing the desired pretension.

## Direct Tension Indicator Method Pretensioning




Direct Tension Indicator (DTI)

A direct tension indicator (DTI) is a compressible, washer-type device with small arch-like protrusions/bumps on one face that allow a gap between a fastener-bearing surface and the washer. It is essential that a DTI be properly oriented in the assembly. For proper orientation, the protrusions must bear against the unturned component in the assembly.

Once a snug-tight condition is achieved, the connection is tightened causing plastic yielding of the protrusions. During the process, the DTI is compressed to a gap that is less than the gap specified by the manufacturer's instructions and in concert with the RCSC specification. A feeler gage is used to verify if the DTI has been properly compressed to an adequate gap, which is indicative of meeting a required tension. Once the gap is reduced to a prescribed amount, the bolt is considered to be properly tightened and the tightening process can stop.

Job inspection gap is a gap between a DTI and the hardened surface on which it bears that is less than the gap measured in a bolt tension measurement device when a tension equal to 1.05 times the minimum required pretension is applied to the bolting assembly.

As per RCSC specification, after the snug-tightening operation is performed, the installer shall verify that the DTI protrusions have not been compressed to a gap that is less than the job inspection gap in half or more of the locations, and if this has occurred, the DTI shall be removed and replaced. After all bolts in the joint are pretensioned, the installer shall verify that the DTI protrusions have been compressed to a gap that is less than the job inspection gap in more than half of the locations.

It may be noted that a DTI washer can only indicate the minimum tension needed to close the gap. In the case of over tightening, DTIs are not capable of indicating the amount of over tensioning. Also, the use of a DTI does not allow for directly monitoring bolt relaxation because the deformation of the protrusions is plastic (i.e., not returning to their original dimensions).

Squirter DTI Washer


Use of Squirter DTI Washer for Pretensioning

As shown in above figure, a Squirter DTI washer is the same as other DTI washer except a Squirter DTI washer fills the voids under the protrusions with colored (orange in above figure) silicone.

Above figure shows use of Squirter DTI washer for pretensioning. As the bolt is tightened the protrusion compress pushing the silicone to edge of the washer indicating to the installer that the correct tension has been achieved.

## Combined Method Pretensioning

The combined method relies on an established relationship between fastener torque and tension to achieve or surpass the prescribed initial tension by torque tightening the bolts to the initial torque value provided by the fastener supplier. After initial tensioning, the bolt or nut is rotated by a designated additional amount relative to the bolt to reliably achieve the minimum specified pretension. This final pretensioning step is similar to the turn-of-nut method, but the angle of rotation is different and likely less because it is relative to the initial tension condition of the combined method, which is usually higher than the minimum snug condition required for the turn-of-nut method.

Thus, for initial tensioning, the initial torque provided by the fastener supplier shall be applied to the nut. If the initial torque has not been provided by the supplier, then the torque (default initial torque as per RCSC) given in the following table shall be used.

| Default Initial Torque Range for Combined Method | | | | |
|---|---|---|---|---|
| Nominal Bolt Diameter, $d_b$, in. | Torque Range, lb-ft[a] | | | |
| | Group 120 | | Group 144[b] and Group 150 | |
| | Min | Max | Min | Max |
| ½ | 45 | 50 | 60 | 75 |
| ⅝ | 100 | 120 | 120 | 145 |
| ¾ | 170 | 205 | 210 | 250 |
| ⅞ | 260 | 310 | 335 | 400 |
| 1 | 405 | 480 | 510 | 605 |
| 1⅛ | 570 | 680 | 710 | 845 |
| 1¼ | 810 | 965 | 1010 | 1200 |
| 1⅜ | 1060 | 1260 | 1325 | 1575 |
| 1½ | 1390 | 1655 | 1735 | 2065 |

[a] - This table shall not be used in lieu of supplier-provided torque values and shall only be used when torque has not been provided for a bolting assembly by the bolt supplier.
[b] - F3148 Group 144 bolting assemblies are only available up to 1¼ in. diameter.

After the application of the initial torque and when the plies have been brought into firm contact, the rotation specified in the following table shall be applied to all bolting assemblies in the joint.

| Nut Rotation from Initial Torque for Combined Method Pretensioning[a,b] | |
|---|---|
| **Bolt Length[c]** | **Rotation** |
| Not more than $4d_b$ | 90° (¼ turn) |
| More than $4d_b$ but not more than $8d_b$ | 120° (⅓ turn) |

[a] - Nut rotation is relative to bolt regardless of the element (nut or bolt) being turned. For all required nut rotations, the tolerance is plus 45 degrees (⅙ turn) and minus 0 degrees.
[b] - Applicable only to joints in which all material within the grip is steel.
[c] - When the bolt length exceeds $8d_b$, the required nut rotation shall be determined by actual testing in a suitable bolt tension measurement device.

The bolting assemblies used for the combined method should be treated as matched bolting assemblies.

## Preinstallation Verification

The RCSC specification requires that all fastener assemblies utilizing high-strength bolts for pretensioned and slip-critical joints be tested prior to installation.

Preinstallation verification testing is frequently conducted through the use of a tension calibrator. The most commonly used device is the Skidmore-Wilhelm bolt tension calibrator, while other similar devices can be used.

In general, a tension calibrator is used to verify that fasteners meet the minimum bolt tension requirement, wrenches are properly calibrated to achieve proper tension, twist-off-type tension control bolts sheer off at the correct tension, protrusions of DTIs properly deform, and workers understand how to achieve the proper pretension. For detail information, please see the specification.

While a conventional tension calibrator can serve various purposes, its use may be limited, particularly when the bolts being tested are too short to fit into a calibration device. In this case, DTIs can be used in pre-installation verification testing as an alternative. If used for verification purposes, the DTIs must first be calibrated in conformance with the procedures outlined in the RCSC specification. However, the use of DTIs may not be a suitable option with the turn-of-nut pretensioning method. This is because the force required to compress DTIs may consume part of the turns required for the turn-of-nut procedure. When utilizing the turn-of-nut method and if bolts are too short to fit into a tension calibrator, ensuring that proper components are used in a fastener assembly and applying a required turn would be an adequate preinstallation verification.

## Inspection

Inspection tasks prior to, during, and after bolting shall be performed in accordance with the invoking specification or standard. For more information, please see RCSC specification.

**Important**

In case of pretensioning by turn-of-nut method, inspector may instruct bolting crews NOT to back off any nut turn that is greater than what they were supposed to rotate from snug-tight condition. Over rotation is not a cause for rejection or rework. Compensating for over rotation, by backing the nut off, will result in less than the required pretension.

## Other Standards and Installation Specifications for Structural Fasteners

Fasteners used for structural steelwork are also called preloaded bolting assemblies and high strength friction grip bolting. Various other standards for structural fasteners and installation specifications are as under.

**Europe - European Standards**

EN 14399: High-strength structural bolting assemblies for preloading, Pare 1 to Part 10
EN 1090-2: Execution of steel structures and aluminium structures - Part 2: Technical requirements for steel structures.

It may be noted that on structural bolting in Europe two technical solutions exist to achieve the necessary ductility of bolt/nut/washer assemblies. These solutions utilize different systems (HR and HV) of bolt/nut/washer assemblies. Both systems are well proved and it is up to the experts responsible for structural bolting whether they use the one or the other system. It is, however, important for the performance of the assembly to avoid mixing up the components of both systems.

**India - Indian Standards**

IS 3757: Specifications for high strength structural bolts.
IS 6623: High Strength Structural Nuts - Specifications.
IS 6649: Specification for Hardened and Tempered Washers for High Strength Structural Bolts and Nuts.
IS 4000: High Strength Bolts in Steel structures - code of Practice.

**Australia - Australian Standard**

AS/NZS 1252: High strength steel bolts with associated nuts and washers for structural engineering.
AS 4100: Steel structures.

# Locking of Threaded Fasteners

Threaded fasteners are the fastener of choice in multiple industries and applications for the simple reason that they are easy to dismantle. However, this also makes them vulnerable to slackening and self loosening resulting in loss of preload.

Threaded fasteners can come loose on occasions without human intervention. This loosening can be due to slackening for example due to creep, embedding, stress relaxation (in gaskets) or the fastener self-rotating called self loosening (often also called vibration loosening). Creep, embedding and stress relaxation will generally not completely loosen a fastener, these loosening mechanisms occur without the nut rotating relative to the bolt. The term self loosening is used for the nut rotating relative to the bolt without human intervention. It is known that the fastener can self rotate under the action of transverse joint movement due to cyclic loading like shock, vibration, dynamic load, etc. that can completely loosen a tightened fastener such that the nut will become detached from the bolt. In these situations, to prevent self loosening locking of fastener may be required.

Depending on the application, bolt loosening can have profound consequences. One loose bolt can bring a whole production plant to a standstill and cost some company thousands, while in other applications loose bolts can pose a significant safety hazard. In view of this, information about various methods/solutions for locking of threaded fasteners to prevent self loosening is given in this chapter.

## Why Do Fasteners Self Loosen?

The most influential research published on the subject to-date is by German engineer Gerhard Junker in 1969 in which he reports on a theory he developed as to why fasteners self loosen under vibratory loading. Junker found that transverse dynamic loads resulting in relative movement (slip) between the joined parts generate a far more severe condition for self loosening than dynamic axial loads and can completely loosen fasteners.

Testing methodology of Junker to determine at which point a fastener starts rotating loose when subjected to vibration, is now universally known as the Junker Test and has been adopted as the international standard, such as the DIN 65151 and ISO 16130.

Hence, to prevent self loosening, the slip between the joined parts needs to be eliminated or minimized. This can be achieved by either increasing the bolt tension (preload) - high preload or bolt tension provides a high normal force, which, in turn, creates high frictional forces; increasing the friction between the clamped parts; or decreasing the cyclic loading. Another common method is to increase the friction between the bolt and nut threads.

If you're designing the joint, one way to minimize slip is to orient the bolts and joints so that bolt axes are parallel to the expected direction of vibration because axial vibration is far less of a problem than transverse vibration.

## Lock Washers

Lock washers are a very popular locking choice especially with small sized fasteners. These washers are placed under the nut or screw head. Essentially, there are two types: spring type and tooth type.

Spring type washers include split (helical spring washer) and conical (Belleville). They compress as the fastener is tightened and the spring back tension deters loosening. A tooth lock washer - internal, external, internal-external and countersunk external creates a ratchet action by biting into the nut or screw head and the surface it contacts.

**Helical Spring Washers**

Helical Spring Washers

Typical helical spring washers shown in above figure are made of slightly trapezoidal wire formed into a helix of one coil so that the free height is approximately twice the thickness of the washer cross section. They are usually made of hardened carbon steel.

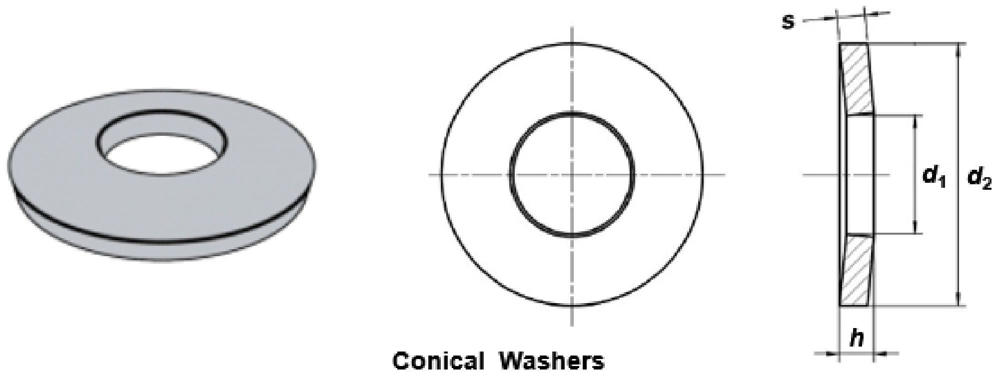
Steel helical spring washers serve to counteract the loss in inherent tension caused by setting or creep of a bolt/nut assembly provided that they are sufficiently resilient to increase the overall resilience of the assembly and that their inherent springiness can compensate for any loss in tension so that the clamping force required to ensure the reliability of the assembly is maintained.

Since these washers are not highly reliable, they are generally used for non-critical applications only. The section on lock washers in NASA Reference Publication 1228 (1990) - "Fastener Design Manual" states the following.

"The helical spring washer serves as a spring while the bolt is being tightened. However, the washer is normally flat by the time the bolt is fully torqued. At this time, it is equivalent to a solid flat washer, and its locking ability is nonexistent. In summary, a lock washer of this type is useless for locking."

Some helical spring washers are having bent (deflected or tang) ends. When the washer is tightened, the sharp edges of the washer are supposed to dig into the nut and mounting surface to prevent counter-clockwise rotation. In practice bent end is unable to dig into hard surfaces and does not actually prevent rotation.

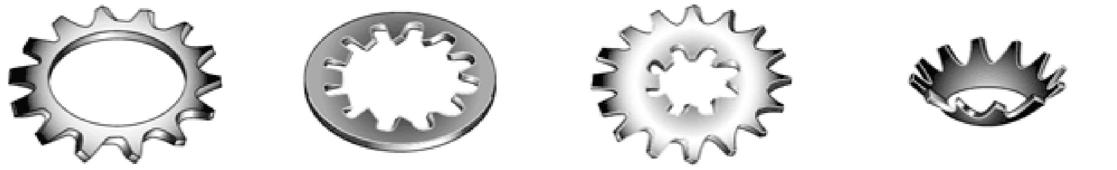
**Conical Washers (Belleville Washers)**



Conical Washers

Conical washers are also known by their alternative names such as Belleville washers or cupped spring washer. They are intended to counteract the effect of setting which results in bolt/nut assemblies working loose. However, unless they have serrations on their surfaces, they have no significant locking capability. Of course, the serrations will damage the mating surfaces under them.

**Tooth Lock Washers**

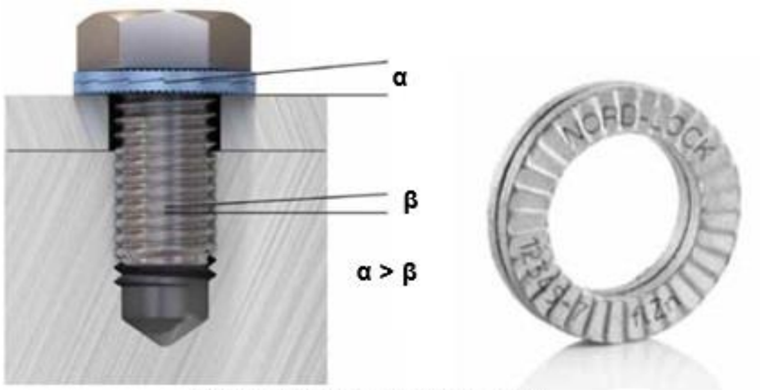


External Tooth | Internal Tooth | External-Internal Type | Countersunk Type

Tooth Lock Washers

Tooth lock washers are used with screws and nuts for some spring action but mostly for locking action. They serve to increase the friction between the screw and the assembly. The teeth are formed in a twisted configuration with sharp edges. One edge bites into the bolt/screw head (or nut) while the other edge bites into the mating surface. Although this washer does provide some locking action, it damages the mating surfaces.

**Nord-Lock Washer Assembly**




Nord-Lock Washer Assembly

Nord-Lock washer assembly consists of two pieces: a top and a bottom washer. These reusable washers have wedges on one side, which should be installed interlocked, and radial teeth on the opposite side. When the screw or nut is tightened, the teeth bite into the screw head or nut and the mating material to prevent slippage. Since the cam angle α is greater than the thread pitch β a wedge effect is created by the cams, which increase tension and prevents the bolt from rotating loose. For more information on Nord-Lock washers, please see website of Nord-Lock International: www.nord-lock.com.

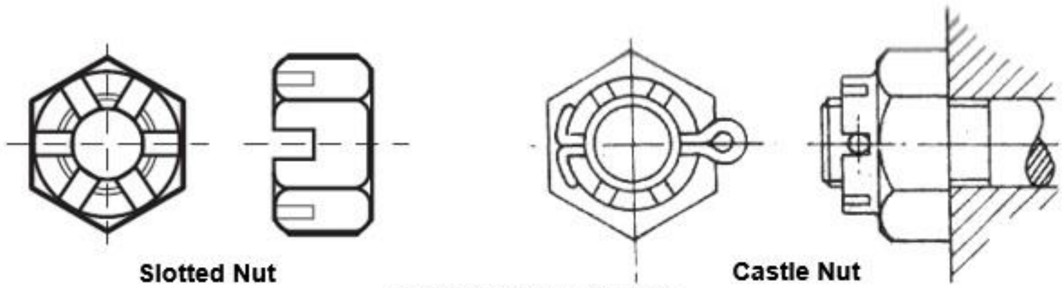
**Sems**

Sems are screws that have captive lock washers beneath their heads that can't be removed without damage. This one-piece approach saves time during assembly, but if the screw or lock washer becomes damaged, you have to replace both.

## Locknuts

Basically, there are two types of locking nuts: free running and prevailing torque. Free running types often rely upon an additional component, such as a jam nut. Prevailing torque types are designed to create friction by wedging (or binding) the nut threads to the bolt threads. Hence, in prevailing torque types there is substantial friction during assembly. Even after slight loosening, the friction continues to resist further loosening.

**Slotted and Castle Nuts**

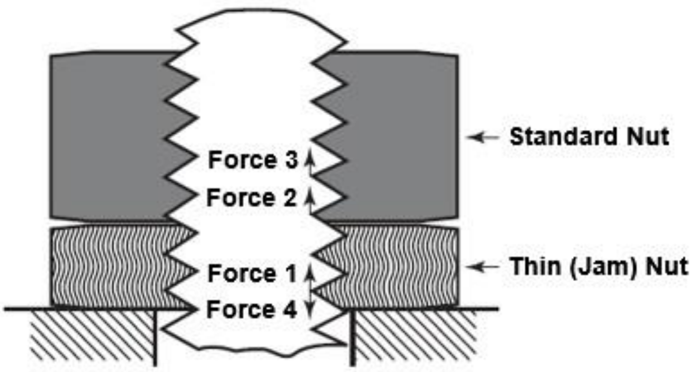



Slotted and Castle Nuts

As shown in above figure, slotted nuts and castle nuts are normally having six slots in the top face. A nut without castellation is called slotted nut where as a nut with castellation is called castle nut. Slotted and castle nuts are used with bolts or studs that have a hole drilled in their threaded section. Once the nut is installed, a cotter/split pin is inserted through opposing slots in the nut and the bolt's hole to prevent the nut from loosening. This method requires that the hole be drilled in the correct place. And because the nut slots are spaced at 60 degree intervals, it may not be possible to tighten the nut precisely - it may have to be over or under tightened for the hole and a slot to align.

It may be noted that slotted nuts are often used in a non-tensioned assembly because full thread engagement gets compromised by the slots.

**Jam Nuts**




Jam Nut

A jam nut (also called a locknut) is a nut used in combination with another nut to prevent the fastener from loosening. The process is sometimes called double nutting. The jam nut can be thinner or of the same thickness as the standard nut. In assemblies with jam nut, the thin nut is first tightened against the assembled parts and then the standard nut (regular or high nut) is tightened against the thin nut. When used in conjunction with a standard nut with the intention of ‘locking’ the assembly, these nuts are commonly assembled incorrectly.

The correct assembly method is to install the thin (jam) nut FIRST as shown in above figure and tighten it to snug tight condition, Force 1 (it should not be tightened severely to produce a high tension in the bolt). Next install the standard nut and snug tighten it, Force 2. Now, holding the thin nut against rotating, further tighten the standard nut to full design tension, Force 3. As the standard nut is tightened, the threads of the thin nut must first bear upward on the bolt threads, then are free, and finally bear downward on the bolt threads, Force 4; while the threads of the standard nut bear upwards on the bolt threads, Force 3. Thus the two nuts are bearing in opposite directions (direction of Force 4 is opposite to the direction of Force 3) on the bolt threads and are jammed. The upper nut has to carry the higher load and therefore, has to be the thicker of the two. These nuts will remain locked even if tension in the assembly is lost.

This locking principle can also be used to fix a nut in any position on the male screw thread and therefore create a shoulder.

**Keps**

Also known as K-Lock nuts, Keps are similar to Sems in that they are nuts with pre-assembled lock washers (usually external tooth). It can save time and make the job easier, handling a single piece.

## Serrated Fasteners

Serrated Fasteners

Serrated fasteners, available as screws and as nuts, have serrated teeth that bite into the mating surface as they are tightened. In that respect, they act like tooth lock washers because the ratchet action they create resists loosening. But, also like tooth washers, they damage the mated surface.

Serrated fasteners are marketed by UNBRAKO as Durlok® fasteners. For more information on them, please see website of UNBRAKO: www.unbrako.com.

## Prevailing Torque Type Fasteners

Prevailing torque type locking fasteners use some form of interference to create friction with the mating threads. Considering the two options - nuts and screws - nuts are more common. A single nut can be used with many different bolt and screw lengths and head styles, thus reducing inventory. Prevailing torque screws, however, must be used in situations like tapped holes.

There are two ways to make prevailing torque fasteners: using all metal, or by adding a non-metallic element. Prevailing torque all metal type nut has a one piece or a multiple piece metal construction and derives its prevailing torque characteristics from a controlled distortion of the nut thread and/or body and/or from metallic insert(s). Prevailing torque non-metallic insert type nut has a multiple piece construction and derives its prevailing torque characteristics from insert(s) of non-metallic material retained in the nut. Prevailing torque screws use a nylon patch or pellet 2 to 3 threads from the end. Some of the more common prevailing torque type nuts are covered here.

**Split Beam Locknut**

A split beam locknut, shown in above figure, a prevailing torque type all metal nut, has slots in the top, and the thread diameter is undersized in the slotted portion. The nut spins freely until the bolt threads get to the slotted area. On further tightening, the split "beam" segments are deflected outward by the bolt, and a friction load results from binding of the mating threads.

Split beam locknut are marketed by SPS Technologies, LLC. as FLEXLOC®. For more information on FLEXLOC®, please see website of SPS Technologies: www.spstech.com.

**Elliptical Collar Out-of-Round Collar**
**Deformed Thread Locknuts (Cleveloc Nuts)**

Above figure shows typical deformed thread prevailing torque type all metal nuts. The collar of the nut is elliptical or out-of-round (three crimps) in cross section and it is this that provides the flexible locking element. These type of nuts are commonly called Cleveloc nuts.

**Nylock Nut**

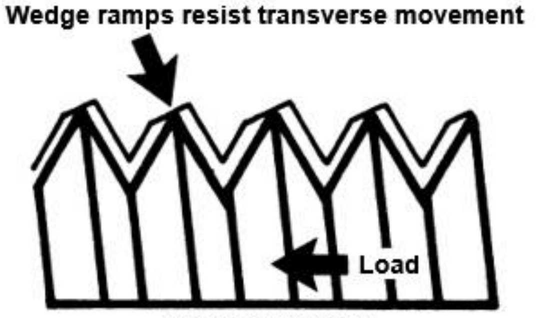
Above figure shows a typical prevailing torque non-metallic insert type nut, commonly called Nylock nuts. These nuts have a nylon or any other polymer ring swaged into the nut just after the threads which has an interference fit on the male thread that causes resistance to nut turning. Because the non-metallic ring tightly conforms to the mating threads, it also provides an effective seal. The main drawback of the non-metallic ring is its maximum operating temperature, which is approximately 121°C (250°F). The non-metallic ring will also be damaged quickly by reassembly. These type of nuts are sometimes also called a nylon insert nut or an elastic stop nut.

A cost-saving method sometimes used is to bond a nylon patch instead of a nylon ring on the threads of either the nut or the bolt to get some locking action. Nylok is the leading manufacturer of mechanical locking and thread sealing products including nylon patch and nylon pellet. For more information on them, please see their website: www.nylok.com.

It may be noted that a prevailing torque type locking nut may loosen in service. However, once most of the clamp load has been relieved, the nut will not loosen further, and will still stay attached to the bolt and prevent loss of the bolt.

The prevailing torque must be added to the required clamping torque when determining the assembly torque for a prevailing torque locknut.

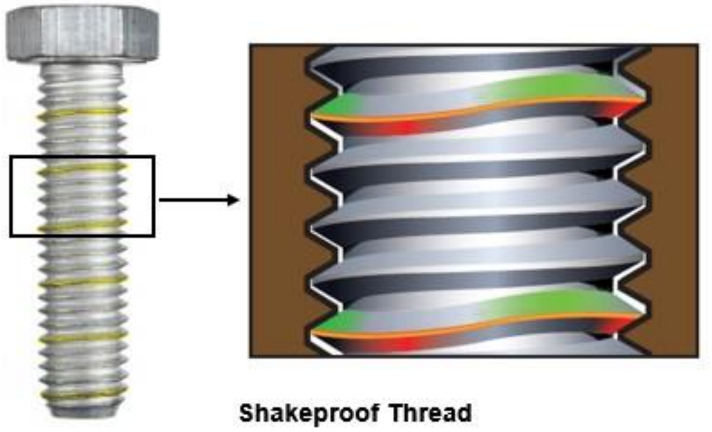
## Spiralock Thread Nut




Spiralock Thread

As shown in above figure, the Spiralock thread form has a 30° wedge ramp at its root. Under clamp load the crests of the male threads are wedged tightly against the ramp. This makes lateral movement, which causes loosening under vibration, nearly impossible. A Spiralock thread nut is free spinning until seated. This thread form requires 20% more torque than a standard thread form to achieve a given preload. The extra torque is required to pull the male threads up the root ramps of the female threads.

## Shakeproof Thread



Shakeproof Thread

In general, fasteners loosen due to standard tolerance clearance. As shown in above figure, the Shakeproof™ Thread design integrates a curved thread feature which creates a mechanical interference with the mating internal threads, thus eliminating the tolerance clearance between the threads of a fastener to create a vibration-proof joint with as few as four engaged pitches. For more information on Shakeproof Thread, please see website of *ITW* Shakeproof Industrial: www.shakeproof.com.

## Locking Adhesives

One very good way to resist vibrations is to ''cement'' the nut and bolt together. The most common way to do this is with an anaerobic adhesive, a material which is activated (hardens) when subjected to high pressure in the presence of metal and the absence of air. It is applied to fastener threads much as a lubricant would be applied. It ''glues'' the threads together when they are tightened. It can, however, be overcome if required subsequently to take the joint apart. The material does no permanent damage to the threads.

A wide variety of anaerobic adhesives are available. Selection would be based on such things as the size of the gap to be filled (gap filling prevents relative side slip), adhesive strength (off-torque requirements), size of the fastener, method of applications, etc. These adhesives are usually effective as thread sealers as well. It's best to consult the manufacturers for details. The first company to provide anaerobic adhesives is Henkel Loctite (website: www.loctite.com).

In many cases the adhesives are applied by the user before assembly; in other cases, they're pre-applied by the manufacturer. If the adhesive is pre-applied by the manufacturer, it is applied via a sophisticated microencapsulation process. When the fastener is rotated against its mating part the capsules of adhesive burst, releasing the material in and around the thread flanks, which then cures and forms the locking bond. Properly applied, an aerobics adhesive creates a nut factor of 0.14 to 0.17 on steel, and so can serve as an assembly lubricant. They can be used at service temperatures up to 200°C.

It may be noted that sometimes a mixture of an epoxy resin and a hardening agent is applied by users before assembly for permanent, one-time application.

Other well-known manufacturers of anaerobic and epoxy adhesives are Permabond (www.permabond.com), Nylock (www.nylok.com) and 3M (www.ndindustries.com).

## Mechanically Locked Fasteners

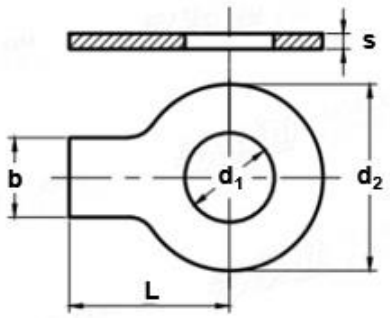
Sometimes self loosening would threaten safety and we want to be absolutely sure that the fasteners won't come loose. In such situations we can consider the use of fasteners in which the nut and bolt are mechanically locked together.

### Lockwiring

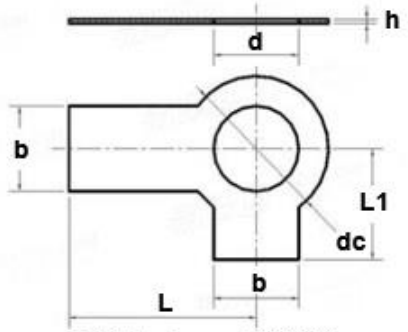

Although lockwiring is a laborious method of preventing bolt or nut rotation, it is still used in critical applications, particularly in the aerospace field. The nuts usually have drilled corners, and the bolts either have through holes in the head or drilled corners to thread the lockwire through. Above figure shows a typical bolt head lockwiring assembly.

Note that two strands of the lock wire are normally twisted together, as shown, with one strand passing through a drilled hole in the head of the fastener. The wiring is installed in such a way that any tendency for the fastener to loosen would require stretching of the wires.

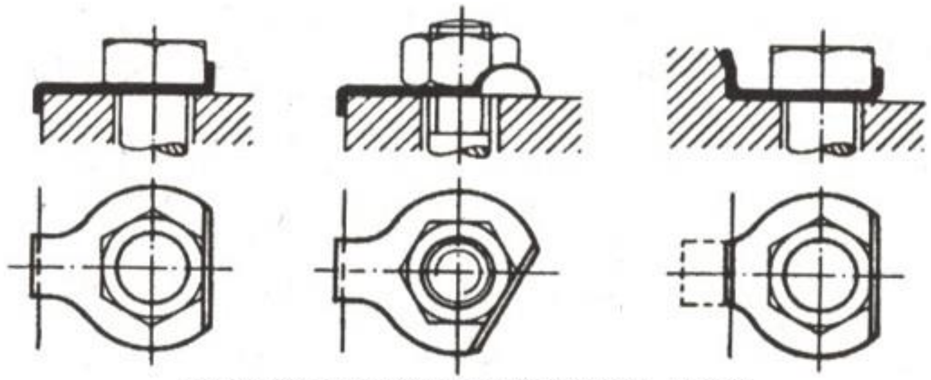
**Tab Washers**




Tab Washers - DIN 93




Tab Washers - DIN 463



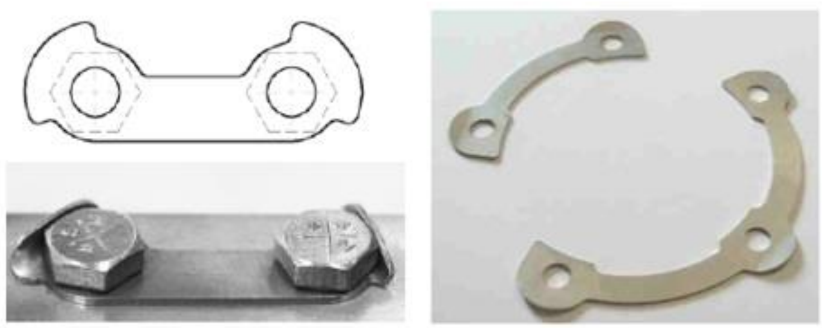
Examples of Application: Tab Washers - DIN 93

Tab Washers

Tab washers (DIN 93, DIN 463, etc.) are thin metal washers designed with tabs which project from the outside diameter. The washer is placed below the head of the bolt or the nut and following tightening one tab is bent down over an edge of the assembly and a second tab bent up to lock the nut or the bolt head in position.

As this method fixes the nut or bolt head to the adjacent surface, use two tab washers (one for bolt and other for nut) to lock a set of fastener if the other component is not secured by some other method.

**Lock Plates**

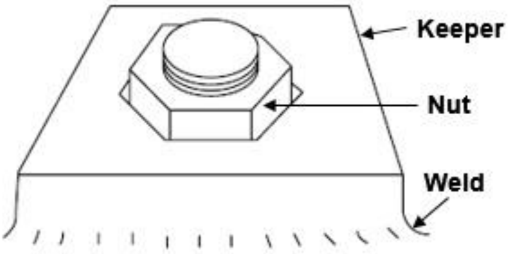


Lock Plates

Lock plates are used to prevent fasteners rotation by fitting them over two or three bolts. As shown in above figure, a bolt head or a nut is locked by bending up the small tab of the lock plate.

**Welding**

Nuts can be welded to bolts, at least if they're large enough. The normal procedure is to tack weld the nut to the end of the bolt. Another procedure is to tack weld both the nut and the head of the bolt to joint surfaces. Either procedure makes removal of the nut (for example, maintenance purposes) very difficult and will probably make it necessary to replace them with new parts if they are removed.



Keeper Welding

A related procedure, which preserves the parts, is to place a square plate with a hexagonal hole in it (called ''keeper'' plate) over the tightened nut and then weld the square plate to the joint as shown in above figure.

It may be noted that tack welding can be done on carbon steel fasteners, but it is not recommended for hardened alloy materials.

**Staking of Threads**

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

**Note**

It is recommended to use a fine pitch thread instead of a coarse pitch thread to prevent self loosening of threaded fasteners. As per a report on an experiment, fine pitch threads endured twice as many vibration cycles as coarse pitch threads.

# Helical Coil Inserts

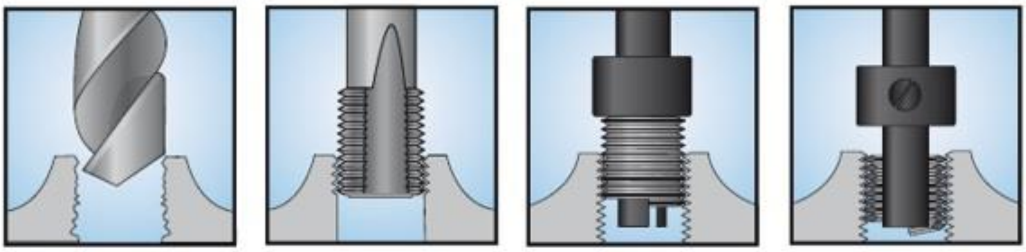
Helical coil inserts are screw thread bushings coiled from wire of diamond shape cross-section. They are screwed into STI (Screw Thread Insert) tapped holes to form nominal size internal threads. They are utilized in a wide variety of original equipment and repair applications. In view of this, information about them is given in this chapter.

## Uses

Helical coil insert increases the possibility of higher torque application in weaker/softer metals to give them steel like thread performance. This has enabled industries to be able to use lighter weight materials such as Aluminum without having to give up thread strength of their fasteners.


Repair of Stripped Threads

As shown in above figure, internal threads on a machine assembly can become stripped or cross-threaded due to repeated use, over tightening or by cocking a bolt while reassembling. An old-timer's fix is to drill the hole out to allow tapping to the next size screw thread and use a larger bolt. While this certainly will produce a quality repair, this fix is not always possible because of space or assembly requirements. In such cases the helical coil insert is a good maintenance fix.

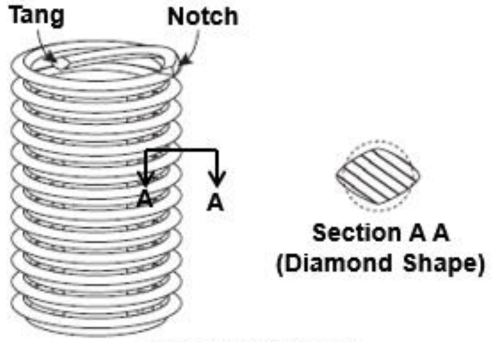



Four Step Repair Procedure

The thread repair procedure consists of choosing an insert that has the same thread size as the damaged thread. The old threads are drilled out and the hole is tapped with a STI (Screw Thread Insert) tap. Special tools (installation tool and tang break tool if supplied) are used to insert the proper helical coil insert and to break off the driving tang. Now the hole can be reused for assembly using the original sized bolt. Above figure shows the four steps of thread repair procedure.

Screw-lock inserts provide prevailing torque on the screw or bolt for use in vibration applications.

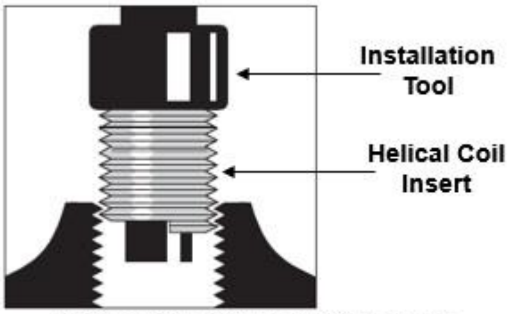
## Construction and Installation



Helical Coil Insert

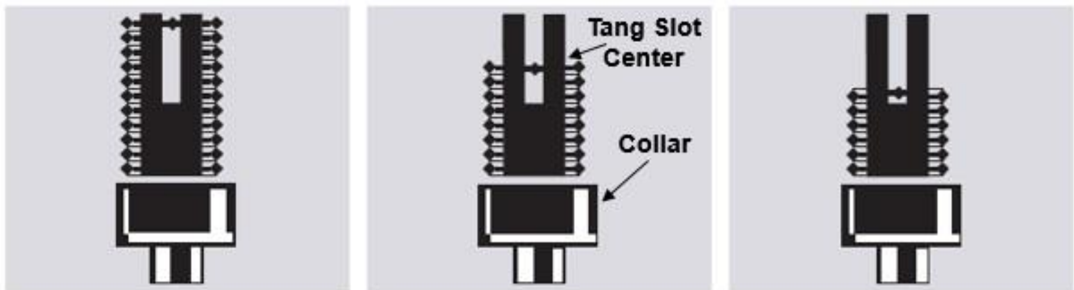
As shown in above figure, helical coil inserts are helically wound, precision formed coils of extremely hard, diamond shaped 304 stainless steel wire. The wire is having an ultimate tensile strength of 200,000 - 250,000 PSI, mirror like surface finish of 8-16 micro inches and flank hardness of RC 43-50. Generally, a diametral tang is provided for their installation. This tang is notched for its removal after installation. When installed into a STI (Screw Thread Insert) tapped hole, they provide permanent conventional 60° internal screw threads that accommodate most standard bolts or machine screws. Inserts are also available in a variety of materials (Inconel X750, Phosphor Bronze, Titanium, etc.) and special plating and coatings (Cadmium Plating, Dry Film Lubricant, etc.).

In many cases, the helical coil insert is stronger than the original threaded hole, and at the same time it significantly reduces the possibility of thread wear, seizing and corrosion.



Installation of Helical Coil Insert

Inserts are installed by torquing them through the diametral tang. For installation, an insert is place on mandrel of the installation tool so that the insert tang is centered in the tang slot as shown in the following figure. Now the insert is screwed in the tapped hole with light downward pressure until it is 1/4 to 1/2 turn below the surface.



Collar Adjusted Too Low Collar Adjusted Correctly Collar Adjusted Too High

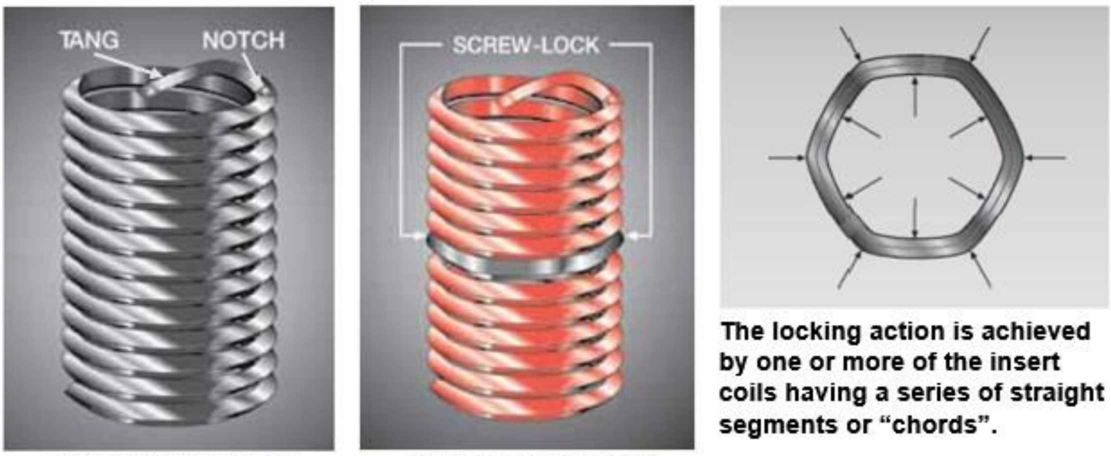
Placing of Insert Tang at Center of Tang Slot

The tang is broken off after installation by simply striking it with the installation tool, tang breakoff tool or a piece of rod. In sizes over ½ inch and all spark plug applications, a long-nosed plier is used to bend it up and down until it snaps off at the notch.

Helical Coil Insert in Semi-Installed Position

As shown in above figure, in the free state, inserts are larger in diameter than the tapped hole into which they are installed. In the assembly operation, the torque applied to the tang reduces the diameter of the leading coil and permits it to enter the tapped thread. The remaining coils are reduced in diameter as they, in turn, are screwed into the tapped hole. When the torque or rotation is stopped, the coils expand with a spring-like action anchoring the insert in place against the tapped hole.

## Types of Helical Coil Inserts




The locking action is achieved by one or more of the insert coils having a series of straight segments or "chords".

Free Running Insert

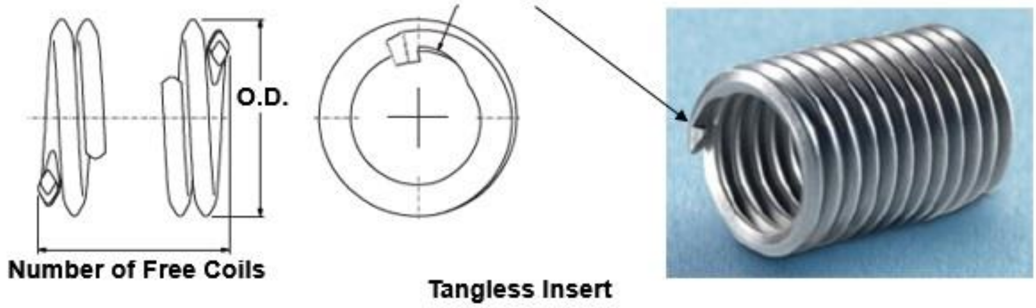
Screw-Locking Insert

Types of Helical Coil Inserts

As shown in above figure, there are two types of inserts. The standard or **Free Running** insert provides a smooth, hard, and free-running thread; and the **Screw-Locking** insert provides self-locking torque on the male member by a series of "chords" on one or more of the insert coils.

Screw-locking inserts are generally dyed red for identification, except when they are cadmium plated or dry film lubricant coated. The dye may completely or only partially cover the insert. However, it must be sufficient to identify the insert when it is installed in its threaded hole.

It is recommended that a cadmium plated or dry-film lubricated screw/bolt is used for screw-locking inserts applications.



Tangless Insert

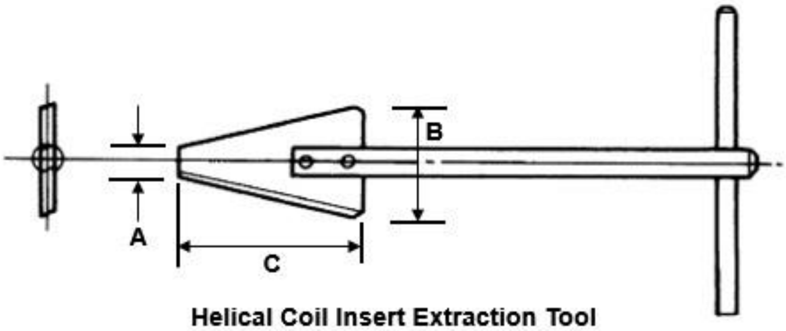
Tangless inserts eliminate the need for tang break-off after insert installation. No tang to break off eliminates a stage in the installation process, ideal for automatic installation in high volume applications. They are available in free running and screw-locking types.

## Nominal Length

Normally, the engaged length of bolt in an insert is determined by strength considerations.

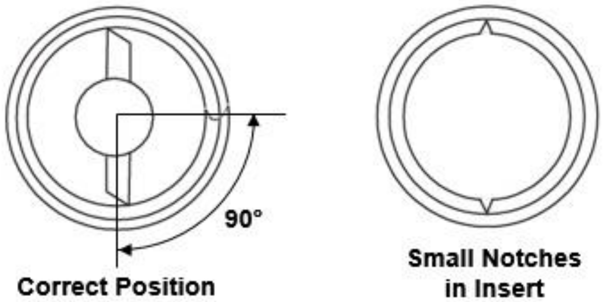
Each nominal insert size is standardized in five lengths which are multiples of the insert's nominal diameter. These are 1, 1.5, 2, 2.5 and 3 times nominal diameter. Each nominal length is the minimum through-hole length (material thickness), without countersink, into which that insert can be installed. It is a reference value and cannot be measured.

## Helical Coil Insert Extraction Tool



Helical Coil Insert Extraction Tool

Occasionally helical coil inserts must be removed. Inserts can be removed manually with little effort. This is done by inserting the blade of the extraction tool (shown in above figure) into the helical coil insert so that the V section of the blade is toward the top end of the insert. To remove an insert, select the extraction tools based on the screw/insert size and strike the head of the tool with a light blow. Now, maintaining a steady pressure of the blade against insert, turn the extracting tool counterclockwise until the insert is removed.



Correct Position of Insert Extraction Tool

As correct positioning of the extraction tool will make the extraction easier, the extraction tool should be turned 90° from the start of the coil allowing easy winding out of the insert. If the extraction tool is not gripping the insert, the edges can be resharpened. Should the extraction tool not grip the insert, file two small notch in the insert for the tool to bite into as shown in above figure.

## Industry Standards

Following are ASME standards for helical coil inserts and tooling.

**ASME B18.29.1: Helical Coil Screw Thread Inserts - Free Running and Screw Locking (Inch Series)**

This Standard is intended to delineate the dimensional data for the inch series helical coil screw thread insert and the threaded hole into which it is installed. Appendices that describe insert selection, STI (Screw Thread Insert) taps, gages and gaging, insert installation, and removal tooling are also included.

**ASME B18.29.2M: Helical Coil Screw Thread Inserts, Free Running and Screw Locking (Metric Series)**

This Standard delineates the dimensional, mechanical, and performance data for the metric series helical coil screw thread insert and threaded hole into which it is installed. Appendices that describe insert selection, STI (screw thread insert) taps, insert installation, and removal tooling are also included.

## Helical Coil Thread Repair Kits

Thread repair kits are available in metric, inches, and spark plug series and each kit comes with a quantity of helical coil inserts, the appropriate size drill, high speed steel STI tap, installation tool, and a tang removal tool. Many manufacturers supply such kits.

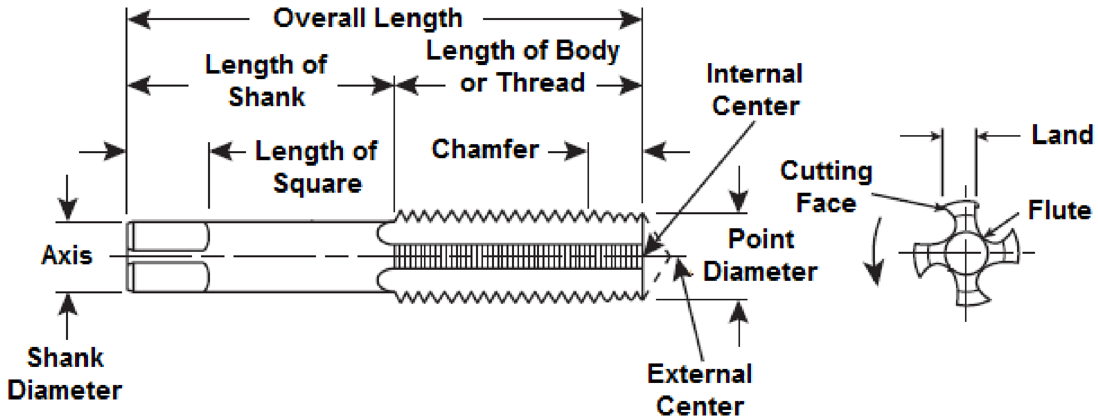
Heli-Coil® is a registered trademark of Emhart Teknologies, Inc. Emhart Teknologies are one of the leading suppliers of helical coil inserts and thread repair kits. For more information on helical coil inserts and thread repair kits, please see website of Emhart Teknologies, Inc.: www.emhartamericas.com, or website of Noble Aerospace Pvt. Limited: www.noblefix.com.

Recoil® is a registered trademark of Alcoa Fastening Systems. They also supply helical coil inserts and thread repair kits. For more information on their products, please see their website: www.afsrecoil.net.

# Screw Thread Tapping

Cutting internal screw threads with a hand tap, called tapping, is an operation frequently performed by mechanics. Often this is a troublesome operation involving broken taps and time consuming efforts to remove them. Some of these troubles may be avoided by a better understanding of the tapping operation and the tools involved. In view of this, information about hand taps, tapping, tap-drill size charts and tap extractor is given in this chapter.

## Hand Taps



Typical Hand Tap

As shownn in above figure, a typical hand tap basically consists of a shank, which has flats on the end to hold and drive it, and a threaded body, which does the thread cutting. The threaded body is composed of lands, which are the cutters, and flutes or channels to let the chips out and permit cutting fluid to reach the cutting edges. The threaded body is chamfered or tapered at the point to allow the tap to enter a hole and to spread the heavy cutting operation over several rings of lands or cutting edges. Applicable standards for hand taps are ISO 529 and IS 6175.

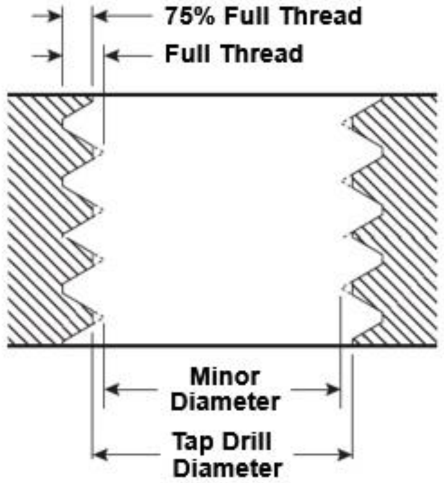
Because no single tap could possibly meet all the difficult tapping requirements, the manufacturers of threading tools modify the basic tap design in several ways, making tools that are especially suitable for particular tapping needs. The number of flutes may vary from two to as many as nine. Unless specific reference to the number of flutes is made, taps are supplied having the standard number of flutes for a given size and type. The most widely used sizes for general use are made with four flutes.

The chamfer (also called lead) on a tap refers to the angular reduction in diameter of the leading threads at its point. This chamfer allows the tap to enter the hole and the gradually increasing diameter of the threads serve to lead/guide the threaded body of the tap into the hole as it is turned. The longer the chamfer, the smaller the chip each thread must cut, as the load is distributed over a greater number of cutting edges. The three types of hand taps based on different chamfer lengths in common use are taper, plug or second, and bottoming.

As shown in the following figure, taper taps have the longest chamfer, 10 to 12 threads (4° per side) and are usually used for starting to tap a hole. The taper tap is frequently referred to as a “starting” tap. The plug tap has a 6 to 8 threads chamfer (8° per side) and is used to provide full threads more closely to the bottom of a hole than is possible with a taper tap. The bottoming tap has 1.5 to 2 threads chamfer (22° per side), but when carefully used, it can tap to the bottom of a hole if it is preceded by a plug tap. The taper taps should always be used to start tapping a hole. While tapping may be started with a plug tap if care is used, the load on the leading threads is extremely heavy because of the short chamfer. Do not attempt to start tapping a hole with a bottoming tap.

Types of Hand Taps

## Hand Tapping



75 Percent Full Thread

The first operation in internal threading with a hand tap is to drill the proper diameter hole. The usual method of selecting the drill size is to refer a “tap-drill size chart”. Note that the charts carry the statement “Based on approximately 75 percent full thread”. This notation means that the drill diameter is larger than the minor diameter of the thread to be tapped. The drill size will produce a hole enough oversize so that 25 percent of the thread at crest of the internal thread will be missing.

An oversize hole is made to provide clearance between the hole and the minor diameter of the tap. If this is not done there will be no clearance and the tap will turn hard, tear the threads, and run a high risk of breakage. The 25 percent that is missing from the crest of the internal thread does not appreciably reduce its strength.

To tap a hole with a hand tap, begin with the taper tap. Start the tap by placing it in the hole and carefully turn it about one half turn until it starts to cut. Check the position of the tap by eye to keep it square with the work surface. Back it up to break the chip, and again turn it in about one half turn. Each time, check the tap to be sure it is square. After the tap is well started, check it with a square. If it is not true, pressure must be exerted in the opposite direction to correct its position on the next several cuts. A jig may be used to ensure that the tap is vertical. Use plenty of lubricant and occasionally completely remove the tap from the hole to clean out chips. Continue alternately cutting and reversing the tap to break the chips.

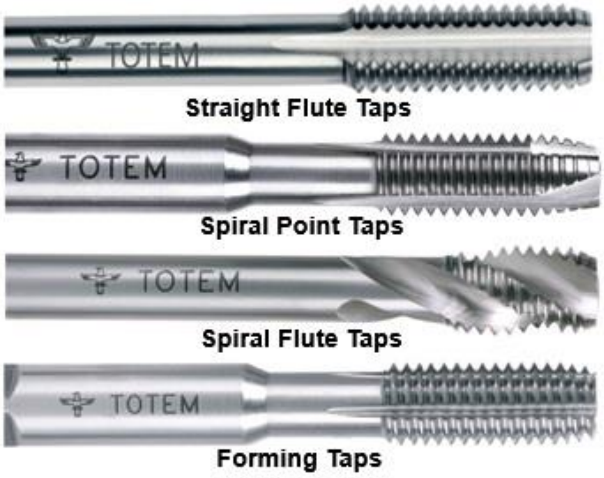
When the tap is cutting properly, one can "feel" it. If the tap resists turning and there is a springy feeling, it is not cutting properly and should be removed and inspected.

To tap a blind hole, use first the taper tap, then the plug tap, and finally the bottoming tap. Each tap must be carefully turned in until bottomed. Because the chips cannot fall through, special attention must be paid to chip removal. The tap must be removed more frequently than for a through hole, and the work piece must be inverted and jarred to remove the chips. If the work piece cannot be inverted, blow the chips out with air or remove them with a magnet. In special cases, it might even be necessary to use a vacuum for chip removal.

It may be noted that a twist drill is a roughing tool that may be expected to drill slightly oversize and that some variations in the size of the tapping holes are almost inevitable. When a closer control of the hole size is required, it must be reamed. Reaming is recommended for the larger thread diameters and for some fine pitch threads.

## Machine Taps

For machine tapping, in addition to straight flute taps, three more types of taps used are: spiral point taps, spiral flute taps and forming taps.



Machine Taps

Straight flute machine taps are dimensionally same as hand taps and with a chamfer of 1.5 threads. These taps are recommended for tapping on grey cast iron, S G iron and steel. The chips produced are short or powder type.

Spiral point taps are also known as gun nose or bullnose or chipdriver. These taps are dimensionally the same as a hand tap, but have the cutting face ground back relative to the axis of the tap, for the lead portion. This gives the flute a better cutting action, requiring less power, and pushes the cut material forward, allowing free flow of coolant along the flute to the cutting edge. The flutes are not ground as deeply as for hand taps, giving the tap greater strength. It can therefore be run at higher speeds. Spiral point taps are ideal for machine tapping of through holes, or blind holes where there is enough clearance beyond the threaded portion to accommodate the swarf (fine chips).

Spiral flute taps are used to produce a thread close to the bottom of a blind hole and therefore have a very short chamfer. The right hand spiral cut of the flute acts to force the swarf away from the cutting teeth to the rear of the flute and out of the hole. They are better on materials which form long continuous stringy swarf, rather than chips. They are also better to tap a thread in a hole where there is a break in the material, e.g. another hole, as the spiral fluting helps the tap to pick up on the other side.

Forming taps are also known as roll, flute less taps. These taps do not have flute or cutting edges but have special roll forming lobes with circular land. They are used for the chip less production of threads in ductile materials such as copper, aluminium, soft brass or steel (up to 36 HRC). Threads produced by flute less taps are formed by plastic deformation; they are not cut. They are operated at high speeds and are better at maintaining gauge qualification.

## Removing a Broken Tap


Broken taps can be removed by using a tap extractor, which has hardened steel fingers that enter the flutes of the tap. The tap is backed out of the hole by turning the extractor with a wrench. Sometimes the injection of a small amount of a proprietary solvent into the hole will be helpful. A solvent can be made by diluting about one-part nitric acid with five-parts water. The action of the proprietary solvent or the diluted nitric acid on the steel loosens the broken tap so that it can be removed with pliers or with a tap extractor. The hole should be washed out afterwards so that the acid will not continue to work on the part. Another method is to add, by electric arc welding, additional metal to the shank of the broken tap, above the level of the hole. Care must be taken to prevent depositing metal on the threads in the tapped hole. After the shank has been built up, the head of a bolt or a nut is welded to it and then the tap may be backed out. Following figure shows the 3 steps for removing broken taps.

**Step 1**
**Thoroughly remove all chips of the broken tap. Insert the extractor fingers into the flutes of the broken tap, pushing them gently but firmly into position.**

**Step 2**
**Push the holder down until it touches the broken tap. Slide the sleeve down until it touches the work. (Both are important.)**

**Step 3**
**Apply a tap wrench to the square end of the holder. Twist forward and backward a few times to loosen, then back out the broken tap.**

**3 Steps for Removing Broken Taps**

## US Number and Letter Gauge Drill Bit Sizes

Number drill bit gauge sizes are analogous to, but different from, American wire gauge.

Number gauge is routinely used from size 80 (the smallest) to size 1 (the largest) followed by letter gauge size A (the smallest) to size Z (the largest). Number gauge is actually defined at least down to size 97, but these smaller sizes are rarely used.

The gauge-to-diameter conversion does not follow a set formula, but rather was defined as a useful and practical measure.

Number and letter gauge drill bits are still in common use in the U.S. In the past, they were popular elsewhere, but now have been largely discarded in favor of metric sizes. They are sized to provide proper clearance holes for screws and bolts according to ASME B18.2.8.

Following tables shows decimal size in inch for drill number and letter sizes.

| Number Drill Sizes | | | | | | | |
|---|---|---|---|---|---|---|---|
| Drill No. | Decimal Size, in. | Drill No. | Decimal Size, in. | Drill No. | Decimal Size, in. | Drill No. | Decimal Size, in. |
| 1 | 0.2280 | 21 | 0.1590 | 41 | 0.0960 | 61 | 0.0390 |
| 2 | 0.2210 | 22 | 0.1570 | 42 | 0.0935 | 62 | 0.0380 |
| 3 | 0.2130 | 23 | 0.1540 | 43 | 0.0890 | 63 | 0.0370 |
| 4 | 0.2090 | 24 | 0.1520 | 44 | 0.0860 | 64 | 0.0360 |
| 5 | 0.2055 | 25 | 0.1495 | 45 | 0.0820 | 65 | 0.0350 |
| 6 | 0.2040 | 26 | 0.1470 | 46 | 0.0810 | 66 | 0.0330 |
| 7 | 0.2010 | 27 | 0.1440 | 47 | 0.0785 | 67 | 0.0320 |
| 8 | 0.1990 | 28 | 0.1405 | 48 | 0.0760 | 68 | 0.0310 |
| 9 | 0.1960 | 29 | 0.1360 | 49 | 0.0730 | 69 | 0.02925 |
| 10 | 0.1935 | 30 | 0.1285 | 50 | 0.0700 | 70 | 0.0280 |
| 11 | 0.1910 | 31 | 0.1200 | 51 | 0.0670 | 71 | 0.0260 |
| 12 | 0.1890 | 32 | 0.1160 | 52 | 0.0635 | 72 | 0.0250 |
| 13 | 0.1850 | 33 | 0.1130 | 53 | 0.0595 | 73 | 0.0240 |
| 14 | 0.1820 | 34 | 0.1110 | 54 | 0.0550 | 74 | 0.0225 |
| 15 | 0.1800 | 35 | 0.1100 | 55 | 0.0520 | 75 | 0.0210 |
| 16 | 0.1770 | 36 | 0.1065 | 56 | 0.0465 | 76 | 0.0200 |
| 17 | 0.1730 | 37 | 0.1040 | 57 | 0.0430 | 77 | 0.0180 |
| 18 | 0.1695 | 38 | 0.1015 | 58 | 0.0420 | 78 | 0.0160 |
| 19 | 0.1660 | 39 | 0.0995 | 59 | 0.0410 | 79 | 0.0145 |
| 20 | 0.1610 | 40 | 0.0980 | 60 | 0.0400 | 80 | 0.0135 |

| Letter Drill Sizes | | | |
|---|---|---|---|
| Drill Letter | Decimal Size, in. | Drill Letter | Decimal Size, in. |
| A | 0.234 | N | 0.302 |
| B | 0.238 | O | 0.316 |
| C | 0.242 | P | 0.323 |
| D | 0.246 | Q | 0.332 |
| E | 0.250 | R | 0.339 |
| F | 0.257 | S | 0.348 |
| G | 0.261 | T | 0.358 |
| H | 0.266 | U | 0.368 |
| I | 0.272 | V | 0.377 |
| J | 0.277 | W | 0.386 |
| K | 0.281 | X | 0.397 |
| L | 0.290 | Y | 0.404 |
| M | 0.295 | Z | 0.413 |

# Tap-Drill Size Charts

ISO 2306 specifies the sizes of drills for ISO metric threads, ISO inch threads and pipe threads. IS 10952 is identical with IS0 2306.

To select drill for tapping threads (for approx. 75% thread) as per Unified inch screw threads and Metric screw threads, use the following tables.

| Tap-Drill Size Chart for Unified Inch Screw Threads | | | |
|---|---|---|---|
| Coarse Threads | | Fine Threads | |
| Tap Size (Thread-Pitch) | Drill Size (No./Letter/Fraction, in) | Tap Size (Thread-Pitch) | Drill Size (No./Letter/Fraction, in) |
| - | - | #0-80 | $\frac{3}{64}$ |
| #1-64 | 53 | #1-72 | 53 |
| #2-56 | 50 | #2-64 | 50 |
| #3-48 | 47 | #3-56 | 46 |
| #4-40 | 43 | #4-48 | 42 |
| #5-40 | 38 | #5-44 | 37 |
| #6-32 | 36 | #6-40 | 33 |
| #8-32 | 29 | #8-36 | 29 |
| #10-24 | 25 | #10-32 | 21 |
| #12-24 | 16 | #12-28 | 15 |
| ¼-20 | 7 | ¼-28 | 3 |
| $\frac{5}{16}$-18 | F | $\frac{5}{16}$-24 | I |
| ⅜-16 | $\frac{5}{16}$ | ⅜-24 | Q |
| $\frac{7}{16}$-14 | U | $\frac{7}{16}$-20 | $\frac{25}{64}$ |
| ½-13 | $\frac{27}{64}$ | ½-20 | $\frac{29}{64}$ |
| $\frac{9}{16}$-12 | $\frac{31}{64}$ | $\frac{9}{16}$-18 | $\frac{33}{64}$ |
| ⅝-11 | $\frac{17}{32}$ | ⅝-18 | $\frac{37}{64}$ |
| ¾-10 | $\frac{21}{32}$ | ¾-16 | $\frac{11}{16}$ |
| ⅞-9 | $\frac{49}{64}$ | ⅞-14 | $\frac{13}{16}$ |
| 1-8 | ⅞ | 1-12 | $\frac{59}{64}$ |
| 1⅛-7 | $\frac{63}{64}$ | 1⅛-12 | $1\frac{3}{64}$ |
| 1¼-7 | $1\frac{7}{64}$ | 1¼-12 | $1\frac{11}{64}$ |
| 1⅜-6 | $1\frac{7}{32}$ | 1⅜-12 | $1\frac{19}{64}$ |
| 1½-6 | $1\frac{11}{32}$ | 1½-12 | $1\frac{27}{64}$ |

| Tap-Drill Size Chart for Metric Screw Threads (All dimensions are in mm) | | | |
|---|---|---|---|
| Coarse Threads | | Fine Threads | |
| Tap Size (Thread x Pitch) | Drill Size, mm | Tap Size (Thread x Pitch) | Drill Size, mm |
| M1.0 x 0.25 | 0.75 | - | - |
| M1.1 x 0.25 | 0.85 | - | - |
| M1.2 x 0.25 | 0.95 | - | - |
| M1.4 x 0.30 | 1.10 | - | - |
| M1.6 x 0.35 | 1.25 | - | - |
| M1.8 x 0.35 | 1.45 | - | - |
| M2 x 0.40 | 1.60 | M2 x 0.25 | 1.75 |
| M2.2 x 0.45 | 1.75 | M2.2 x 0.25 | 1.95 |
| M2.5 x 0.45 | 2.05 | M2.5 x 0.35 | 2.15 |
| M3.0 x 0.50 | 2.50 | M3.0 x 0.35 | 2.65 |
| M3.5 x 0.60 | 2.90 | M3.5 x 0.35 | 3.15 |
| M4.0 x 0.70 | 3.30 | M4.0 x 0.50 | 3.50 |
| M4.5 x 0.75 | 3.70 | M4.5 x 0.50 | 4.25 |
| M5.0 x 0.80 | 4.20 | M5.0 x 0.50 | 5.00 |
| M6.0 x 1.00 | 5.00 | M6 x 0.75 | 5.25 |
| M7 x 1.00 | 6.00 | M7 x 0.75 | 6.25 |

| | | | |
|---|---|---|---|
| M8 x 1.25 | 6.80 | M8 x 1.00 | 7.00 |
| M10 x 1.50 | 8.50 | M10 x 1.25 | 8.80 |
| M12 x 1.75 | 10.20 | M12 x 1.50 | 10.50 |
| M14 x 2.00 | 12.00 | M14 x 1.50 | 12.50 |
| M16 x 2.00 | 14.00 | M16 x 1.50 | 14.50 |
| M18 x 2.50 | 15.50 | M18 x 2.00 | 16.00 |
| M20 x 2.50 | 17.50 | M20 x 2.00 | 18.00 |
| M22 x 2.50 | 19.50 | M22 x 2.00 | 20.00 |
| M24 x 3.00 | 21.00 | M24 x 2.00 | 22.00 |
| M27 x 3.00 | 24.00 | M27 x 2.00 | 25.00 |
| M30 x 3.50 | 26.50 | M30 x 2.00 | 28.00 |
| M33 x 3.50 | 29.50 | M33 x 2.00 | 31.00 |
| M36 x 4.00 | 32.00 | M36 x 3.00 | 33.00 |
| M39 x 4.00 | 35.00 | M39 x 3.00 | 36.00 |

**Rule of Thumb for Tap-Drill Size**

As per ISO 2306, the size of tapping drill diameter is approximately equal to the nominal diameter of the thread minus the pitch.

Hence, as a rule of thumb, for Metric threads, the size of tapping drill can be calculated by subtracting the pitch from the diameter of the thread. Example: for M16 x 2.00 thread, the tapping drill is 16.00 − 2.00 = 14.00 mm.

For Imperial (inch) threads the tapping drill size is calculated in the same way, diameter minus pitch. Example: for ¾ -10 UNC, pitch = 0.1, diameter = 0.75, tapping drill = 0.75 − 0.10 = 0.65 in.

# Screw Extraction

Sometimes a screw/bolt on a machine assembly brakes during its maintenance and need to be extracted. In view of this information about screw extraction is given in this chapter.

## Screw Extractor

Screw Extractor

A screw/bolt extractor is a tool for removing broken screws/bolts. It most commonly takes the form of a tapered hand tap with a reversed thread.

To remove a broken screw, drill about 3 to 6 mm deep hole into the screw head; the depth will depend on the size of the screw extractor you are using. The hole diameter varies according to the extractor size you are using. Follow the recommendations on the package that came with the extractor. Next, insert the tip of the screw extractor into the drilled hole and using a hammer, tap the extractor firmly into the hole. Now apply downward pressure on the extractor and turn it counterclockwise (for right hand screws) with a wrench (or drill). As you turn the screw extractor, its threads will draw it in until it bites into the drilled hole. Once the extractor takes hold, continue turning it counterclockwise and pull it to completely remove the broken screw.

If the extractor slips and loses its bite in the screw, try the following:

- Tap the extractor more firmly to get a better bite into the screw.
- Push down more firmly as you turn the extractor counterclockwise.
- Enlarge the pilot hole slightly and try again.

## Other Methods

Screw extractors are made of hard , brittle steel, and, if too much torque is applied, can break off inside the screw that is being removed. Since the extractor is hard material, a typical drill bit will not be able to drill into it and a larger element of difficulty will get added to the original screw extraction work. One way to avoid this added difficulty is to drill a hole completely through the screw.

Another method is to weld a nut to the screw end and then removing the screw by unscrewing it. Sometimes, you may require to add additional metal to the end of the broken screw before a nut can be welded to it. In such cases, care must be taken to prevent depositing metal on the adjoining machine component.

# Clearance Holes for Bolts and Screws

Clearance holes are larger than the nominal diameter of the bolt or screw and the amount of clearance depends on the desired type of fit. Information about clearance holes for bolts and screws as per ASME 818.2.8 is given in this chapter.

ASME 818.2.8 covers the recommended clearance hole sizes for #0 through 1.5 in. and M1.6 through M100 metric fasteners in three classes of clearance using a close-, normal-, and loose-fit category. The hole sizes for metric fasteners are in agreement with IS0 273: Fasteners - Clearance holes for bolts and screws, except the IS0 273 covers fastener sizes M1 through M150. It may be noted that the three classes of clearance in IS0 273 are called fine, medium and coarse series. The Indian Standard IS 1821 is identical with IS0 273.

The selection of clearance hole size to suit particular design requirements can be dependent upon many variable factors. It is however felt that the normal fit category (medium series) should suit the majority of general purpose applications.

## Clearance Holes for Inch Fasteners

The recommended nominal drill sizes for clearance holes are tabulated by nominal drill designation as letter, numbers, or fractional sizes. The following table lists recommended nominal drill sizes for the different types of fits.

| **Nominal Drill Size for Clearance Holes - Inch Fasteners** | | | |
|---|---|---|---|
| Nominal Screw Size | Nominal Drill Size | | |
| | Normal Fit | Close Fit | Loose Fit |
| #0 | #48 | #51 | $\frac{3}{32}$ |
| #1 | #43 | #46 | #37 |
| #2 | #38 | $\frac{3}{32}$ | #32 |
| #3 | #32 | #36 | #30 |
| #4 | #30 | #31 | #27 |
| #5 | $\frac{5}{32}$ | $\frac{9}{64}$ | $\frac{11}{64}$ |
| ***#6*** | #18 | #23 | #13 |
| #8 | #9 | #15 | #3 |
| #10 | #2 | #5 | B |
| $\frac{1}{4}$ | $\frac{9}{32}$ | $\frac{17}{64}$ | $\frac{19}{64}$ |
| $\frac{5}{16}$ | $\frac{11}{32}$ | $\frac{21}{64}$ | $\frac{23}{64}$ |
| $\frac{3}{8}$ | $\frac{13}{32}$ | $\frac{25}{64}$ | $\frac{27}{64}$ |
| $\frac{7}{16}$ | $\frac{15}{32}$ | $\frac{29}{64}$ | $\frac{31}{64}$ |
| $\frac{1}{2}$ | $\frac{9}{16}$ | $\frac{17}{32}$ | $\frac{39}{64}$ |
| $\frac{5}{8}$ | $\frac{11}{16}$ | $\frac{21}{32}$ | $\frac{47}{64}$ |
| $\frac{3}{4}$ | $\frac{13}{16}$ | $\frac{25}{32}$ | $\frac{29}{32}$ |
| $\frac{7}{8}$ | $\frac{15}{16}$ | $\frac{29}{32}$ | $1\frac{1}{32}$ |
| 1 | $1\frac{3}{32}$ | $1\frac{1}{32}$ | $1\frac{5}{32}$ |
| $1\frac{1}{8}$ | $1\frac{7}{32}$ | $1\frac{5}{32}$ | $1\frac{5}{16}$ |
| $1\frac{1}{4}$ | $1\frac{11}{32}$ | $1\frac{9}{32}$ | $1\frac{7}{16}$ |
| $1\frac{3}{8}$ | $1\frac{1}{2}$ | $1\frac{7}{16}$ | $1\frac{39}{64}$ |
| $1\frac{1}{2}$ | $1\frac{5}{8}$ | $1\frac{9}{16}$ | $1\frac{47}{64}$ |

For information on limiting (Min. and Max.) dimensions for the hole diameter to specify tolerances, please see the standard.

The clearance hole tolerances for both inch and metric holes are based on IS0 286, IS0 System of Limits and Fits, using tolerance class H12 for close-fit, H13 for normal-fit, and H14 for loose-fit clearance holes.

## Clearance Holes for Metric Fasteners

The following table lists recommended nominal drill sizes for the different types of fits.

| Nominal Drill Size for Clearance Holes - Metric Fasteners | | | |
|---|---|---|---|
| Nominal Screw Size | Nominal Drill Size | | |
| | Normal Fit | Close Fit | Loose Fit |
| M1.6 | 1.8 | 1.7 | 2.0 |
| M2 | 2.4 | 2.2 | 2.6 |
| M2.5 | 2.9 | 2.7 | 3.1 |
| M3 | 3.4 | 3.2 | 3.6 |
| M4 | 4.5 | 4.3 | 4.8 |
| M5 | 5.5 | 5.3 | 5.8 |
| M6 | 6.6 | 6.4 | 7 |
| M8 | 9 | 8.4 | 10 |
| M10 | 11 | 10.5 | 12 |
| M12 | 13.5 | 13 | 14.5 |
| M14 | 15.5 | 15 | 16.5 |
| M16 | 17.5 | 17 | 18.5 |
| M20 | 22 | 21 | 24 |
| M24 | 26 | 25 | 28 |
| M30 | 33 | 31 | 35 |
| M36 | 39 | 37 | 42 |
| M42 | 45 | 43 | 48 |
| M48 | 52 | 50 | 56 |
| M56 | 62 | 58 | 66 |
| M64 | 70 | 66 | 74 |
| M72 | 78 | 74 | 82 |
| M80 | 86 | 82 | 91 |
| M90 | 96 | 93 | 101 |
| M100 | 107 | 104 | 112 |

For information on limiting (Min. and Max.) dimensions for the hole diameter to specify tolerances, please see the standard.

### Note

In most cases, cap screw holes have a constant diameter (plain holes), however, in some cases it is desired that the screw head is not sticking-out of the surface. For such cases, counter-bored or countersunk holes may be used.

For information on drill and counterbore sizes for socket head cap screws, please see Appendix A of ASME B18.3.

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