HELPFUL HINTS
For Fastener Design and Application

Supply Technologies™

The efficiency experts.

www.supplytechnologies.com

HELPFUL HINTS

For Fastener Design and Application

Supply Technologies is a Strategic Partner in Total Supply Management™ to over 400 Leading Manufacturers in over 100 industries across the globe. A critical function of our Technical Services team is its ability to work closely with our customers' engineers to design efficiencies into existing and new products. We are an extension of your own departments by bringing experience to facilitate design development, improve assembly line ergonomics, and reduce costs. Supply Technologies specializes in a proactive approach to managing efficiencies in all aspects of supplying parts and materials that ultimately are delivered to production floors. This includes planning, implementing, and managing the flow of product for any size world-class manufacturing company by enabling them to build products better, smarter, and faster than ever before.

Italicized words throughout the booklet are defined in the Glossary.

HELPFUL HINTS 9TH EDITION

People talk about "getting down to the nuts and bolts" when describing something in minute detail. An apt phrase, because it recognizes that fasteners are the basic and fundamental component of assembled products.

Fasteners are often taken for granted. Seldom are they viewed as the highly engineered items that they are. All too often, fasteners are applied by "rule of thumb" techniques, not sound engineered practice. As a result, the performance and economy that has been designed and built into each fastener is not fully realized.

Proper fastener selection and application are critical to ensure proper assemblies. The joining requirements of the assembly and the variety of available products complicates the user's choices. This 9th Edition of "Helpful Hints" contains general information that users can consider in light of their particular requirements. Material in this edition has been updated, rewritten, and regrouped for easier access to the subject matter.

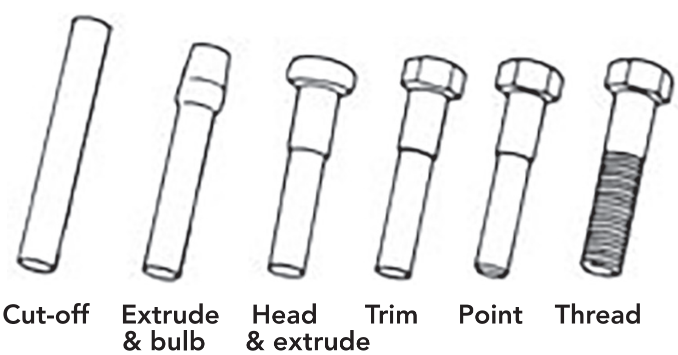
COLD HEADING

Cold heading and *cold forming*, the basic methods of manufacturing most bolts and nuts, is the process of forcing unheated metal to flow into *dies* to change its shape. The machines used are called *headers* and *formers*, which are actually high-speed multi-*blow* presses.

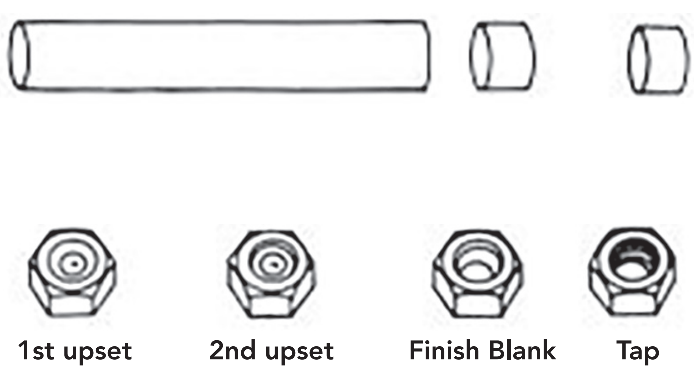
Metal flow lines of a cold-headed product

As illustrated on the right, heading machines cut a length of *wire* from a *coil*, place the cut *blank* in a *die*, and force it into the desired configuration using a series of heading or forming *blows*. The heading operation *upsets* the head. The *die extrudes* the *shank*. Secondary operations, such as pointing, *trimming*, threading, and *tapping* may also be employed.

Stages in Making a Cap Screw



Stages in Cold Nut Manufacturing



Nine Guidelines to Use

Compared to *machining, cold heading* yields stronger pieces at less cost. Also, *cold heading* automatically controls quality because unsound material cannot be used. While the capability of cold *headers* is broad, it pays to design for the process right at the start.

Cold Heading Guidelines

1. Money can be saved by ordering in typical production run quantities as follows:

Diameter/Length	Approx. Pieces
¼ – ⅜ in. x 3 in. long	100,000 +
7⁄16 – ½ in. x 5 in. long	60,000 +
⅝ – ¾ in. x 6 in. long	40,000 +
⅞ – 1 in. x 6 in. long	25,000 +

2. Maximum volume of *upset* is equivalent to length of stock 4½ times its own diameter in two-*blow* heading. *(With special operations, up to 26 diameters have been achieved.)*
3. Various metals and alloys are suitable. Carbon content in steel should be kept under 0.45%.
4. Concentric pieces are easier to form, though eccentric and serrated shapes can be practical to cold form.
5. Avoid sharp corners and allow for generous radii.
6. Since *upsets* are usually cylindrical, oval or round shapes take less *trimming* than square or rectangular shapes.
7. Hollow *upsets* tend to form cracks at edges of recess, so avoid them.
8. *Embossing* raises costs.
9. No problem *heat treating* short sections, but long sections are apt to be distorted. When in doubt, contact an expert in *cold heading* or *heat treating*.

Materials

Materials commonly used in *cold heading* and *cold forming* commercial grade products include:

- Carbon steels up to about 0.45 carbon
- Low carbon martensitic boron steels
- Alloy steels up to about 0.45 carbon
- Copper
- Brass – 60% copper and higher
- Silicon bronze (97% copper)
- Monel
- 300 series stainless steels
- 400 series stainless steels
- Aluminum

Low carbon steels (AISI C1020 and under) are frequently furnished bright, not heat treated, or with a simple stress relief. Higher carbon and alloy products are customarily heat treated.

Some commonly used steels include:

AISI	
C1010	Machine screws, rivets
C1110	Hex nuts
C1018-20	Grade 2 cap screws (made cold)
C1030-38	Grade 5 cap screws (made cold), A 325 Bolts
C4140, C4037, C8367	Grade 8 cap screws, A 490 bolts

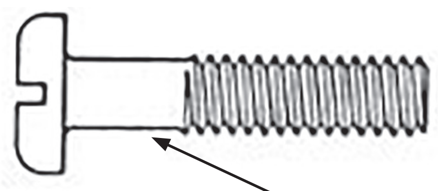
Rolled Threads

Bolt *blanks* are roll threaded by progressively squeezing them between rotating or reciprocating *dies*. Displaced metal then flows into the threading *dies*.

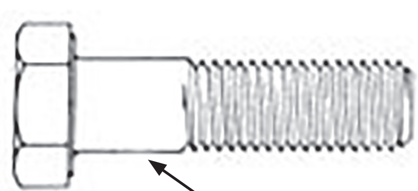
Rolled threads date back to 1851. At that time, rolled threading produced threads with a larger diameter than the *shank*. As a result, it was necessary to cut threads to have the same thread and *shank* diameter. Rolled or cut threads were used to identify cap screws with undersize or full-size body *shanks*, respectively.

This is no longer true. Today all types of threads can be precision rolled. In most instances, rolled threads are better than cut threads, because cold working the metal results in continuous flow lines and improves physical properties – as well as burnishing the surface of the thread.

A high-production process, roll threading is used on 95% of today's output of screws and bolts.



Reduced body diameter
Unthreaded *shank* has a smaller diameter (about equal to the *pitch diameter*) than the OD of the thread.



Full body diameter
Unthreaded *shank* has the same diameter as the OD of the thread.

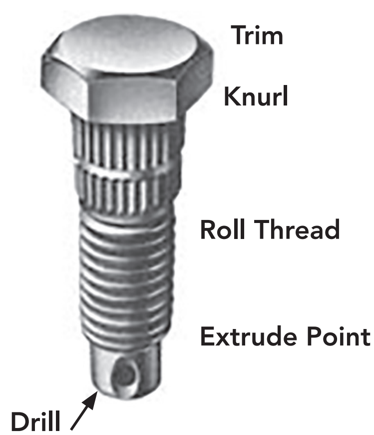
Custom Cold Formed Parts

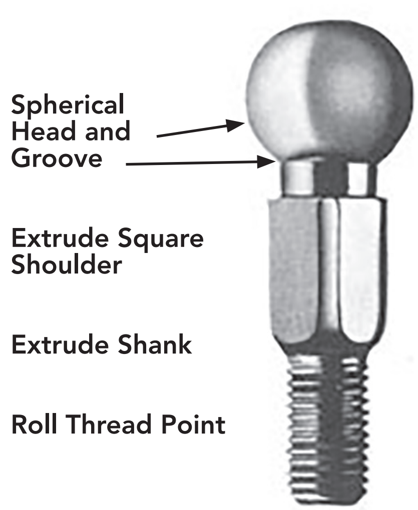
Today's *cold heading* and *cold forming* equipment offer users a source for volume procurement of close-tolerance complex parts. Here are some reasons for designing a special cold headed part:

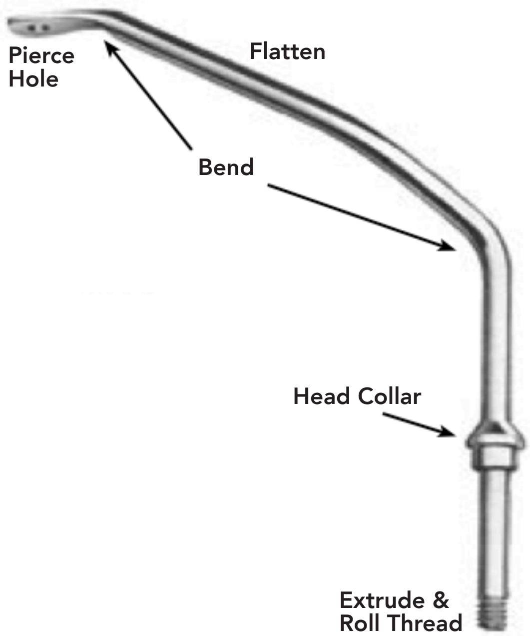
- Design considerations may rule out use of a *standard fastener*.
- A *special part* may prove more efficient in assembly.
- A single special part may replace two or more separate parts.
- A cold headed or cold formed part may be less expensive than a machined part.

When designing a *non-standard part*, try to use a listed mechanical performance standard. This will give your supplier easier access to the raw material, or it may be in their inventory.

Some of the many capabilities of heading and forming equipment are shown on these pages.







Form Flange on Square

Cold Formed Slotted Nut

FASTENER STANDARDS

User-producer groups have developed *standards* for all widely used fasteners. Dimensional *standards* detail product dimensions, and mechanical performance *standards* define materials, hardnesses, physical strengths, and test procedures.

The major fastener specification groups include:

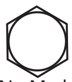
ANSI	**American National Standards Institute, Inc.** https://www.ansi.org/	**ISO**	**International Standards Organization** https://www.iso.org/home.html
ASTM	**American Society for Testing and Materials** https://www.astm.org/	**DIN**	**Deutsches Institut für Normung** https://www.din.de/en
SAE	**Society of Automotive Engineers, Inc.** https://www.sae.org/	**BS**	**British Standards** https://www.bsigroup.com/
IFI	**Industrial Fasteners Institute** https://www.indfast.org/	**JIS**	**Japanese Industrial Standards** https://www.jisc.go.jp/eng/

Product Identification – Fastener Markings

SAE and ASTM specifications offer similar *grades* of fasteners. SAE grades are applicable to automotive and other general design fastening whereas ASTM specifications include structural applications.

Standard fastener-head markings have two important meanings: identifying the strength properties of the fastener and showing the original manufacturer of that fastener. The most widely used strength grade marks are illustrated below:

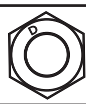
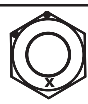
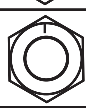
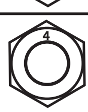
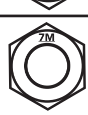
Externally Threaded Inch Fasteners

Grade Identification Marking	Specifications	Material	Nominal Size (In)	Proof Load Stress (ksi)	Tensile Strength Min. (ksi)
 No Mark	SAE J429 – Grade 1	Low or Medium Carbon Steel	1/4 thru 1-1/2	33	60
	SAE J429 – Grade 2		1/4 thru 3/4 over 3/4 thru 1-1/2	55 33	74 60
	ASTM A307 – Grade A		1/4 thru 4	–	60
	ASTM A307 – Grade B		1/4 thru 4	– –	60 min 100 max
	SAE J429 – Grade 5 ASTM A449 – Type 1	Medium Carbon Steel, Quenched and Tempered	1/4 thru 1 over 1 thru 1-1/2	85 74	120 105
	ASTM A449 – Type 1		over 1-1/2 thru 3	55	90
	SAE J429 – Grade 5.1	Low or Medium Carbon Steel, Quenched and Tempered	No. 6 thru 1/2	85	120
	SAE J429 – Grade 5.2 ASTM A449 – Type 2	Low Carbon Martensite Steel, Quenched and Tempered	1/4 thru 1/2	85	120
	ASTM A325 – Type 1	Medium Carbon Steel, Quenched and Tempered	1/2 thru 1 over 1 to 1-1/2	85 74	120 105
	ASTM A325 – Type 3	Atmospheric Corrosion Resistant Steel, Quenched and Tempered			
	ASTM A354 – Grade BC	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 3/4 over 3/4 thru 1-1/2	55 33	74 60
	SAE J429 – Grade 8	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 2-1/2	120	150
	ASTM A354 – Grade BD		1/4 thru 2-1/2 over 2-1/2 thru 4	120 105	150 140
	SAE J429 – Grade 8.2	Low Carbon Martensite Steel, Quenched and Tempered	1/4 thru 1	120	150
	ASTM A490 – Type 1	Medium Carbon Alloy Steel, Quenched and Tempered	1/2 thru 1-1/2	120	150 min 170 max
	ASTM A490 – Type 2	Low Carbon Martensite Steel, Quenched and Tempered	1/2 thru 1	120	150 min 170 max
	ASTM A490 – Type 3	Atmospheric Corrosion Resistant Steel, Quenched and Tempered	1/2 thru 1-1/2	120	150 min 170 max

Externally Threaded Metric Fasteners

Property Class Designation Marking	Nominal Product Dia. (mm)	Material and Treatment	Proof Load Stress (MPa) Length Measurement	Yield Strength (MPa)	Tensile Strength (MPa)
4.6	M5 – M100	Low or Medium Carbon Steel	225	240	400
4.8	M1.6 – M16	Low or Medium Carbon Steel, Partially or Fully Annealed as Required	310	340	420
5.8	M5 – M24	Low or Medium Carbon Steel, Cold Worked	380	420	520
8.8	M16 – M72	Medium Carbon Steel, Quenched and Tempered	600	660	830
8.8	M16 – M36	Low Carbon Martensite, Quenched and Tempered	600	660	830
8S3	M16 – M36	Atmospheric Corrosion Resistant Steel, Quenched and Tempered	600	660	830
9.8	M1.6 – M16	Medium Carbon Steel, Quenched and Tempered	650	720	900
9.8	M1.6 – M16	Low Carbon Martensite Steel, Quenched and Tempered	650	720	900
10.9	M5 – M20	Medium Carbon Steel, Quenched and Tempered	830	940	1040
10.9	M5 – M100	Medium Carbon Alloy Steel, Quenched and Tempered	830	940	1040
10.9	M5 – M36	Low Carbon Martensite Steel, Quenched and Tempered	830	940	1040
10S3	M16 – M36	Atmospheric Corrosion Resistant Steel, Quenched and Tempered	830	940	1040
12.9	M1.6 – M100	Alloy Steel, Quenched and Tempered	970	1100	1220

Internally Threaded Inch Fasteners

Grade Identification Marking	Alternate Identification Marking	Specifications	Material	Nominal Size (In)	Proof Load Stress (ksi)
 No Mark		ASTM A563 – Grade O	Carbon Steel	1/4 thru 1-1/2	69
		ASTM A563 – Grade A	Carbon Steel	1/4 thru 1-1/2	90
		ASTM A563 – Grade B	Carbon Steel	1/4 thru 1 Over 1 thru 1-1/2	120 105
		SAE J995 Grade 2	Carbon Steel	1/4 thru 1-1/2 UNC 1/4 thru 1-1/2 UNF	90 80
		SAE J995 Grade 5	Medium Carbon Steel	1/4 thru 1 UNC 1/4 thru 1 UNF Over 1 thru 1-1/2 UNC Over 1 thru 1-1/2 UNF	120 109 105 94
		ASTM A563 – Grade C	Carbon Steel May Be Quenched and Tempered	1/4 thru 4	144
		ASTM A563 – Grade C3	Atmospheric Corrosion Resistant Steel May Be Quenched and Tempered	1/4 thru 4	144
		ASTM A563 – Grade D	Carbon Steel May Be Quenched and Tempered	1/4 thru 4	150
		SAE J995 Grade 8	Medium Carbon Steel, Quenched and Tempered	1/4 thru 1-1/2	150
		ASTM A563 – Grade DH	Carbon Steel, Quenched and Tempered	1/4 thru 4	175
		ASTM A563 – Grade DH3	Atmospheric Corrosion Resistant Steel, Quenched and Tempered	1/4 thru 4	175
		ASTM A194 – Grade 1	Carbon Steel	1/4 thru 4	130
		ASTM A194 – Grade 2	Medium Carbon Steel	1/4 thru 4	150
		ASTM A194 – Grade 2H	Medium Carbon Steel, Quenched and Tempered	1/4 thru 4	175
		ASTM A194 – Grade 2HM	Medium Carbon Steel, Quenched and Tempered	1/4 thru 4	150
		ASTM A194 – Grade 4	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 4	175
		ASTM A194 – Grade 7	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 4	175
		ASTM A194 – Grade 7M	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 4	150

Internally Threaded Metric Fasteners

Property Class Identification Marking & Property Class of Nut	Style of Nut	Nominal Nut Dia. Range	Proof Load Stress (MPa)
5	Hex Style 1	M1.6 - M4 M5 & M6 M8 & M10	520 580 590
	Hex Style 1 Heavy Hex	M12 - M16 M20 - M36	610 630
	Heavy Hex	M42 - M100	630
8	Hex Style 1 Hex Flange	M1.6 - M10 M5 - M10	800
	Hex Style 1 Hex Flange	M12 - M16 M12 - M16	820
	Hex Style 1 Hex Flange	M20 - M36 only M20	830
9	Heavy Style 2	M3 & M4	900
	Heavy Style 2 Hex Flange	M5 & M6	915
	Hex Style 2 Hex Flange	M8 & M10	940
	Hex Style 2 Hex Flange Heavy Hex	M12 - M16	950
	Hex Style 2 Hex Flange Heavy Hex	M20 - M36 only M20 M20 - M36	920
	Heavy Hex	M42 - M100	920
10	Hex Style 1 Hex Flange	M1.6 - M10 M5 - M10	1040
	Hex Style 1 Hex Flange	M12 - M16 M12 - M16	1050
	Hex Style 1 Hex Flange	M20 - M36 only M20	1060
12	Hex Style 2 Hex Flange	M5 & M6	1150
	Hex Style 2 Hex Flange	M8 & M10	1160
	Hex Style 2 Hex Flange Heavy Hex	M12 - M16	1190
	Hex Style 2 Hex Flange Heavy Hex	M20 - M36 only M20 M20 - M36	1200
8S 8S3	Heavy Hex	M12 - M36	1075
10S 10S3	Heavy Hex	M12 - M36	1245

DESIGN HINTS

Which Assembly Fails?

Two assemblies with identical parts – bolt, stiff spring, and rigid plates are shown.

A cyclic load (P), equal to initial bolt tension (F), is applied to the plates. One assembly fails, the other doesn't. Can you tell which?

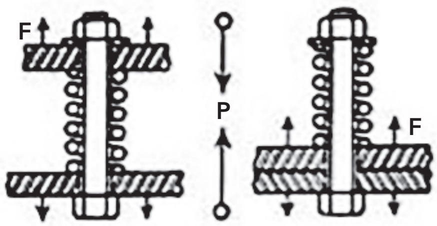


Figure A Figure B

In Fig. A the total bolt load becomes twice the initial bolt tension because the spring's expansion force (equal and opposite to the bolt's initial clamping tension) adds to the load P.

In Fig. B the spring is in effect part of the bolt instead of part of the bolted assembly. With F applied, the load between the plates reduces to zero. The bolt undergoes virtually no basic change in load because it isn't stretched (or the spring compressed).

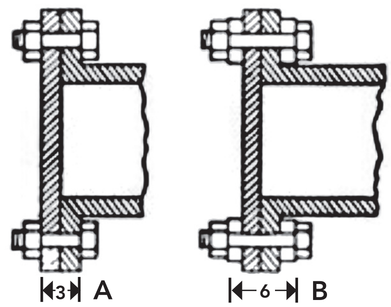
Hence the loading of the bolt in Fig. B is essentially static. This assembly is strong, remains unaffected by the cyclic external stress, and won't fail.

Conversely, the bolt loading in Fig. A, being dynamic, fluctuates between P. Subject to fatigue, it fails.

The hypothetical assemblies shown here have analogs in actual practice for the assembly of flexible, gasketed, and rigid joints.

Role of Fastener Length in Joint Reliability

All steel bolts elongate approximately 0.001-in. per inch of length for each 30,000 psi of tensile stress. Consequently, a long bolt (or screw) can prove to be more reliable than a shorter one for dynamic loading. Here's why:



Hypothetical Example

In Fig. A, the bolt *grip length* is 3 in.; in Fig. B, it's 6 in. Both bolts are SAE Grade 5, coarse threaded, ⅝ in. diameter.

The load is the same on both joints, with each bolt tightened to 60,000 psi, 4,650 pounds of residual tension. Therefore, in Fig. A, the bolt stretches 0.006 in.; in Fig. B, 0.012 in.

Now assume that burrs or scale exist under nut and bolt head *bearing surfaces*. These flatten in service because of cold flow, causing some grip relaxation. Assume a 0.002 in. relaxation in both examples.

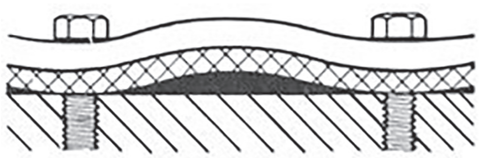
The shorter bolt in Fig. A loses 1,550 pounds of residual tension, retaining only 67% of its initial clamping force. The longer bolt in Fig. B loses 775 pounds of tension, retaining 83% of its initial clamping force.

Thus, the longer bolt, with a higher remaining tension, would be more reliable in joints subjected to cyclic loads, including vibration. In addition, the longer bolt would better resist both loosening and fatigue.

Note: Coarse threads refer to Unified National Coarse (UNC) and fine threads refer to Unified National Fine (UNF).

Fastening Gasketed Joints

Selecting the right fastener for a "flexible" joint depends on the type of gasket material and its compressibility. Total *preload* on all of the fasteners in the connection must be just enough to compress the gasket and provide sufficient clamping force to withstand the hydrostatic test pressure. Excessive fastener *preload* can cause leakage through "bowing" of the clamping plate or simply by gasket creep from overloading the gasket material.



Exaggerated sketch showing how too much torque tends to distort clamping plate and leads to leakage.

Hypothetical Case
Suppose leakage develops when a joint is tightened with Grade 5 cap screws to their *yield strength* and that a switch to alloy screws and further tightening does not solve the problem. The solution may be as simple as using Grade 5 or possibly Grade 2 cap screws, all torqued evenly to a lower clamping load.

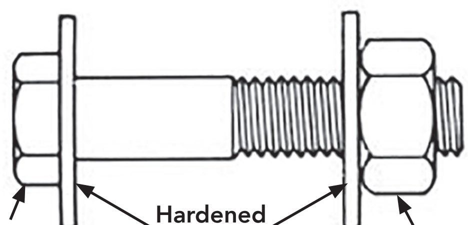
Actual Case
The fasteners on one product's flange had to withstand a 4,000 pound hydrostatic pressure. The hard-asbestos gasket selected required a bolt load of 28,000 pounds for sufficient compression to seal. By substituting a rubber and fibre gasket, in this case, bolt load could be reduced. Bolt size could also be reduced, thereby saving 73% on fastener costs.

Fastening Rigid Joints

Theoretically, a rigid joint is impossible because there is always some elasticity in the fastened metals. For practical purposes, you can consider a joint rigid when the bearing areas of the metal-to-metal fastened members will not crush or yield before the full load-carrying capacity of the screw or bolt is developed.

User Benefits
Rigid joints offer a definite product advantage. They can take high strength cap screws or bolts tightened up to or beyond the *yield strength* of the fastener. Under such tension, fasteners will stay tight despite limited low frequency vibration. In addition, rigid joints are resistant to fatigue from constant load reversals.

Installation Hints
There is no problem achieving a rigid joint when fastening heavy section steel members. Just tighten the fastener to its full fastener capacity. Thin sections can be reinforced and similarly fastened. In joining milder steels or softer metals, use of a *hardened* washer will distribute bolt load, prevent crushing, and give the desired effect of rigidity.



Fastening "Blind Hole" Joints

Holes which don't go all the way through a solid member must be tapped. For small, shallow holes and in fastening soft materials, thread-cutting or thread-rolling screws work well and save time.

Thread Effects
Tapped holes should be coarse threaded because coarse threads are stronger than fine threads and they take fewer turns in assembly. Studs typically go into tapped holes with an interference thread fit, cap screws with a simpler clearance fit.

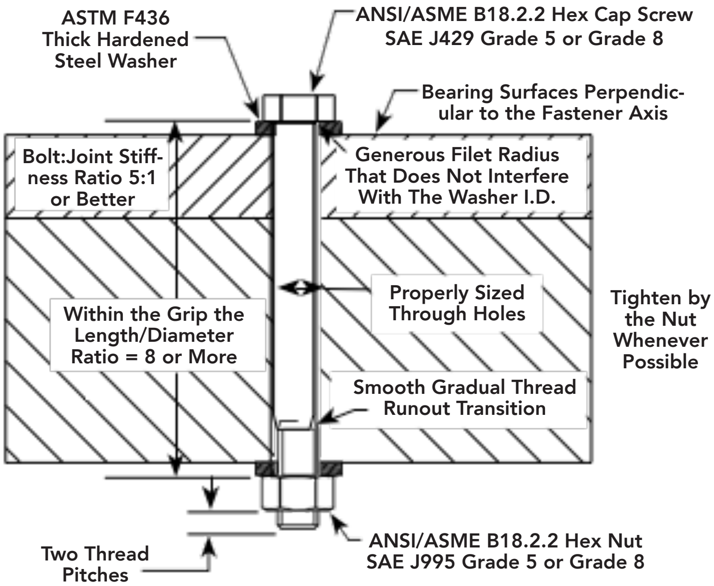
Desirably, studs should all be driven to the same hole depth and bottomed. However, because of normal manufacturing differences, mismatched high and low tolerances will cause some studs to project unevenly. Assemblers must be careful with studs to ensure uniform height projection. This problem does not exist with hex screws that can be tightened to a specified *preload*. When using hex screws, there is no double-driving operation as required with a stud plus nut.

Strength of Blind Joints

Tapped holes behave like nuts. Their threads adjust elastically and plastically to distribute stress and develop high thread tension. Is this harmful in repeated disassembly? Not at all. Cap screws have been installed 50 times in cast iron test blocks, then tightened to failure without damage to tapped holes.

However, "blind hole" fastening of large flanges, pressure plates, heads, etc., requires greater "tapped hole" precision, when using ¼ in. and larger diameter fasteners.

Noteworthy Joint Features



BOLT/SCREW SELECTION

Fastener Strength vs. Fastener Weight

Weight and dimensions of the same size high-strength cap screw (SAE Grade 5) and standard cap screw (SAE Grade 2) are identical as the same machine makes both. They only differ in the clamping force that can be developed and usually in cost.

Higher fastener strength comes from more carbon in the steel and *heat treatment*.

Heat treatment and carbon content represent only a fraction of the total cost of material and manufacturing that go into a fastener. Yet the heat-treated fastener, depending on size, may develop two times the strength of its standard (SAE Grade 2) counterpart. A given load requirement can be met with larger Grade 2 screws, or smaller Grade 5 screws. Weighing less, Grade 5 fasteners cost less, making them the best choice for economy both in terms of purchasing and production.

Grade 5 Most Versatile

There are fastener grades stronger than SAE Grade 5. These are heat-treated alloy-steel fasteners that cost more and unless their strength is needed and utilized in an assembly, money is wasted.

Of the many SAE strength grades, Grade 5 is the most versatile. High-strength Hex Screws suit most design and rapid assembly operations.

Selecting the Right Fastener Grade

With few exceptions, the true function of a fastener is to clamp members together, not to act as an axle or fulcrum. The residual tension set up in the fastener keeps joints tight. Three physical *grades* of steel can satisfy most "clamping" applications:

S.A.E. Grade	Diameter Range (in.)	Tensile Strength (psi)	Proof Load (psi)	Grade Ident. Marking
2	1/4 - 3/4	74,000	55,000	none
	7/8 - 1-1/2	60,000	33,000	none
5	1/4 - 1	120,000	85,000	
	1-1/8 - 1-1/2	105,000	74,000	
8	1/4 - 1-1/2	150,000	120,000	

Metric Property Class	Diameter Range (mm)	Tensile Strength (min., MPa)	Proof Load (MPa)	Grade Ident. Marking
5.8	M5 - M24	520	380	5.8
8.8	M16 - M72	830	600	8.8
10.9	M5 - M100	1040	830	10.9

Grade 2 or PC 5.8 material is low or medium carbon steel.

Grade 5 or PC 8.8 material is a medium carbon steel, quenched and tempered.

Grade 8 or PC 10.9 material is medium carbon alloy steel, quenched and tempered.
Special circumstances allow material and processing modifications. The graph below shows the relative clamping strength of the three S.A.E. grades.

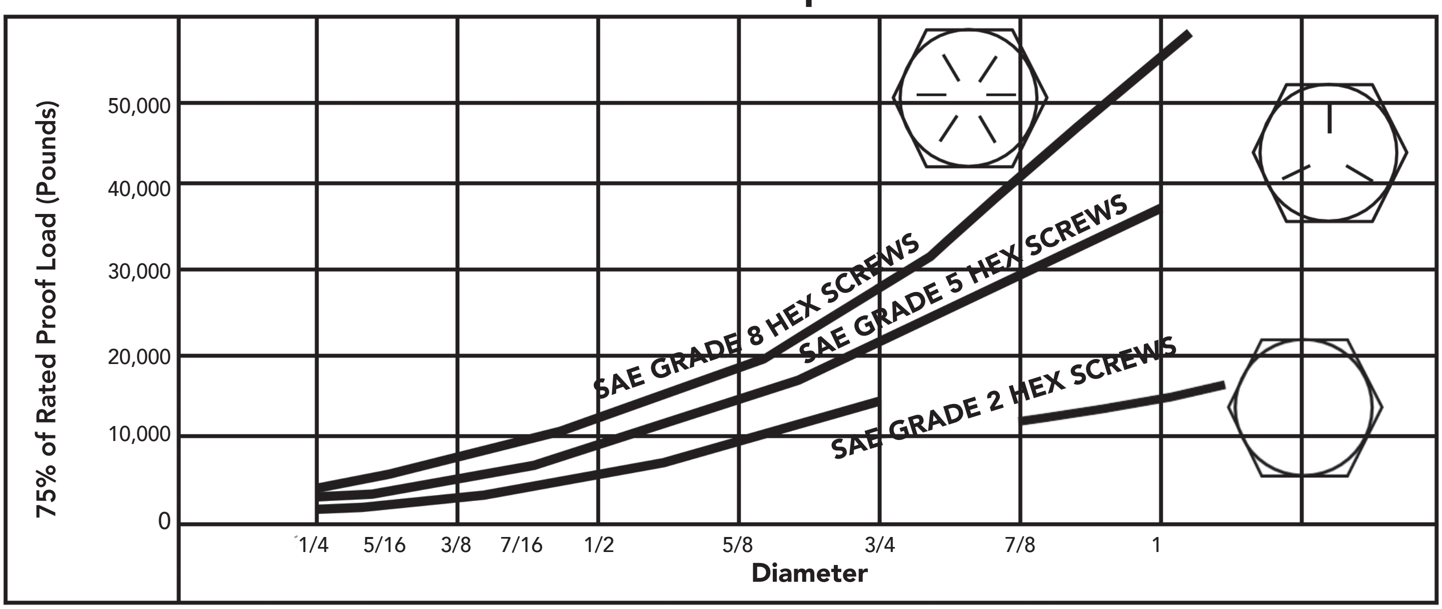
Some Suggestions

In terms of holding power, stronger fasteners may be less costly than the least expensive Grade 2. For example, a ⅝ in. diameter Grade 5 fastener has a *proof load* of 19,200 lb. and a ¾ in. diameter Grade 2 fastener has a *proof load* of 18,400 lb. The ⅝ in. diameter Grade 5 fastener, being smaller than the ¾ in. Grade 2, costs less.

By using higher-strength fasteners, users can affect economies through diameter reduction. If diameter reduction is not desirable, fewer higher-strength fasteners are needed to achieve the same *proof load*.

In short, for more pounds of clamping effort per dollar, use high-strength fasteners; for more pieces per dollar, use lower grade fasteners.

Proof Load Comparison



Hex Screws vs. Studs

Many products once fastened with studs are now assembled with hex screws. Why the change in preference?

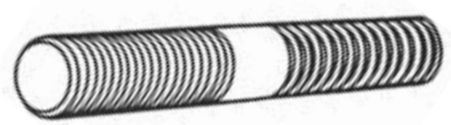
Studs exhibit certain advantages in large diameters and in high-temperature applications. However, when used in smaller sizes and tapped holes, the hex screw has a number of design and production advantages.

Assembly Considerations
In fastener selection, production and assembly considerations are as important as joint strength.

Studs require two wrenchings (first stud, then nut). Also, expensive close-tolerance *tapping* is required since a stud takes an interference fit to stay tight and not withdraw when the nut is backed off.

Hex screws only require a clearance fit. Used in a tapped casting, they can be repeatedly inserted and removed without thread damage.

Stud Application
Studs should not be used as dowels to locate and line up for fastening. To line up numerous studs and bring two pieces together raises assembly cost. Use dowel pins for part alignment cost. Use hex screws to achieve greater fastening economy.



Hex vs. Socket Head Screws

If head diameter is limited, consider using an *internally wrenched* fastener. For more holding power per dollar, use the hex screw.

There is no question that small-head socket screws are best utilized in counterbored holes to clear tight spaces. It is not possible to utilize the full strength of the alloy used for the socket head. Remember that the strength of a connection depends on fastener *preload* and not on the strength of the fastener material alone.

Socket Head Screws
Internal wrenching rarely develops the torque needed for proper *preloading*. If high torque is developed, the smaller bearing area of the socket head tends to crush the *bearing surface* rather than increase tension or *preload*.

Hex Head Screws
Design that takes advantage of heat treated hex screws of SAE Grade 5 quality will yield a stronger connection at a lower cost. These standard fasteners have ample bearing and wrenching surfaces and can be torqued right up to *yield strength*. More importantly, they cost less than alloy fasteners.

NUT DESIGN

The standard nut is not as simple as it looks. Considerable design expertise goes into nut manufacturing to avoid these pitfalls:

Thread Shear: Several factors can cause thread shear in nuts (Fig. A). The nut may not be high enough to contain sufficient threads. The material may lack the plasticity necessary to deform under tightening so enough threads engage and distribute the load to avoid progressive shear.

Wall Dilation: The inclined plane effect of the thread angle (Fig. B) divides the contact stress on a nut into two components: vertical (shear) and horizontal (dilation). Dilation enlarges nut diameter, shifting joint load towards the weaker "tips" of the nut threads.

Crushing: Nuts that do not have sufficient width across flats, (Fig. C) bear down on too small a surface. High unit stress may crush the *bearing surface*, with relaxation of bolt tension as the undesirable end result.

Reputable fastener manufacturers supply ANSI standard Hex Nuts having sufficient height to sustain high thread tension, sufficient wall thickness to control nut dilation under load, and the right amount of bearing area.

Figure A Figure B Figure C

Selecting the Right Type & Grade of Hex Nuts

The standard Hex Nut series is the "workhorse" among nuts. Its products provide adequate bearing area, sufficient height to sustain high thread tension and enough wall thickness to control elastic nut dilation under load.

Heavy Hex Nuts are wider than Hex Nuts in all sizes by ⅛ in. across flats. Thus, their value diminishes as size increases. Most effective in ½ to 1½ in. size ranges, Heavy Hex Nuts satisfy applications using structural bolts or involving excessive clearance holes, unusual loads, and certain boiler codes.

High carbon heat-treated nuts are used with bolts of 150,000 psi (Grade 8) and greater *ultimate tensile strengths*.

Hex Thick Nuts are used where additional thread engagement is required for severe fastening applications and where fine threaded bolts are required.

The need for Hex Thick Nuts is limited since the nut material of the regular carbon steel nut (non-heat treated) is strong enough to pull bolts (through Grade 5) beyond their yield point and lets threads distribute the load to avoid stripping.

Jam Nuts are special function nuts and are used for position locking. The use of two nuts together can be used to form a locking device. When used to lock a standard nut, the Jam Nut should be used between the Hex Nut and the joint surface so the Hex Nut, rather than the Jam Nut, will take the full bolt load.

2H Nuts are used for stability in high-temperature applications and with high-strength structural bolts.

High Nuts are used for shackle, U, and tractor pad bolts. They are furnished only with fine threads and *hardened*. However, the better choice for these applications is coarse thread Heavy Nuts.

Lock Nuts are used where dynamic environmental loading (vibration), differential thermal expansion, or soft joint fastening is present.

Nut Strength

It is difficult to plastically adjust and distribute load over many threads with hard heat-treated nuts. That is why non-heat-treated nuts are strong enough for most applications.

A good rule of thumb is to use a heat-treated nut where bolts with a minimum 150,000 psi tensile strength are used to achieve maximum bolt stress. Conversely, bolts with minimum tensile strengths less than 150,000 psi should use a non-heat-treated nut.

Nut Stripping
On tightening, a nut both compresses and dilates. Dilation can only be overcome by wall thickness, not by added height.

Nut dilation is important since a reduction in the thread flank contact area of the mating bolt under tension also occurs. Threads pull away from each other from their stronger base to weaker thread tips. For fine threads, the shallowness of the threads will also cause progressive shear.

That is why Heavy Hex Nuts (with coarse threads) are a better choice than High Nuts (with fine threads).

Wrenching Stress
Nut rotation places both tension and torsion in the mating bolt. The wrenching force applying this combined stress is about 20% greater than the load, which must be sustained when nut rotation stops.

Thus, if a nut has not failed in wrenching, it can still withstand at least 20% more direct pull than it sustained during tightening.

Should the Nut be able to Break the Bolt?

Practically, the nut should be the strongest member of a bolted assembly. If suitably mated, nut *proof load* will be rated equal to the minimum *ultimate tensile strength* of the bolt.

Thus, with alloy bolts, such as those meeting ASTM A-490 specifications, a 2H Nut is used.

With Grade 5 Bolts, standard Hex Nuts are adequate to develop at least 100% of the bolt's *ultimate tensile load*.

Material strength is not the only factor governing whether the bolt or nut fails first. Threads-per-inch complicate the picture as further explained in the How Threads Affect Nut Strength section.

When applying this to a possibility on the production line, for example, when power wrenching to excessively high loads, a fine thread nut will generally strip before the bolt breaks. In this instance, the nut locks and the wrench "kicks out" with the result that the installer can't tell if the nut is stripped.

With a coarse thread nut at the same load, the bolt would break, and need to be replaced. To gain the benefits of this automatic "inspection" technique, use coarse threads. Actual *proof loads* will be higher with better reliability for critical joints.

Nuts: Their Use & Abuse

Nut performance is critical when bolts are tightened to high load levels.

Bolt tension is produced by a nut rotating and advancing on the bolt threads. To do this properly, there must be a mating condition of threads, which is influenced by *thread lead*.

Thread lead is a matter of tolerance only before the bolt is stressed. When tightened, the nut is then under compression and its threads tend to contract. Conversely, the bolt is in tension and its threads tend to stretch. *Thread lead* is affected - elastically before the yield point is reached, permanently beyond it.

This shortening of one lead thread and lengthening of the other has two effects:
1. Load is distributed unequally along the threads
2. Bolt torsion increases

Deformation must occur, especially for high-tension bolts. In this case, it is better for the nut to do so. As a result, a nut should be soft enough to *deform plastically* and compensate for off-lead threads. If it does, it distributes the load and can advance to increase tension.

Soft Nuts Do Most Jobs

Soft Nuts adjust more readily than hard ones under severe loading conditions. While soft nuts may not be as strong in shear as heat-treated nuts, they can pull the bolt well into its plastic range. However, nuts should be matched according to the appropriate specification.

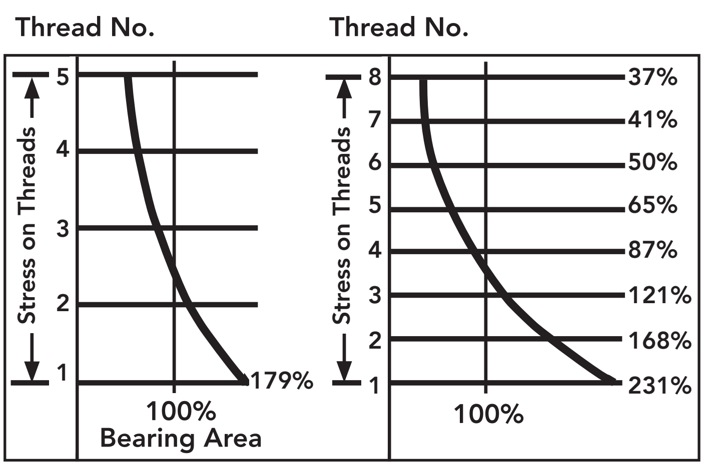
THREAD SELECTION

How Threads Affect Nut Strength

Any nut should be able to withstand stripping loads equal to the minimum *ultimate tensile strength* of the bolt with which it is used.

Theoretically, the stress areas of fine thread bolts should be stronger than coarse thread bolts, however they are not. Failure by thread stripping on a fine thread is a greater possibility for two reasons:

1. On a ½ in. diameter nut, 231% of the average stress on all threads is concentrated on the first fine thread, compared with 179% for the first coarse thread as shown in the following graph.



1/2-13 Coarse

1/2-20 Fine

2. Nuts will dilate under load, shifting stress toward the weaker tips.

Both reasons invite progressive shear.

(0.1419 sq. in.) (0.1599 sq. in.) (0.1799 sq. in.)

The point is shown in the three sketches. Assume a stress area of 0.1419 sq. in. for the coarse thread ½ in. diameter bolt. For a fine thread of equal diameter, the stress area would be 0.1599 sq. in. and, theoretically, a ½ in. diameter fastener could be made with a still shallower thread with a stress area of 0.1799 sq. in. Although this third fastener would be the strongest, the threads would quickly strip.

Therefore, it is not a stress area, but the volume of thread metal remaining in actual engagement under load and the ability to distribute that load over all threads, will then determine the stripping strength.

It is for this reason that SAE specifications for ¼ in. through 1 in. diameter Hex Nuts have reduced the *proof load* of Grade 5 fine thread nuts to 109,000 psi, while their coarse thread counterparts remain at 120,000 psi.

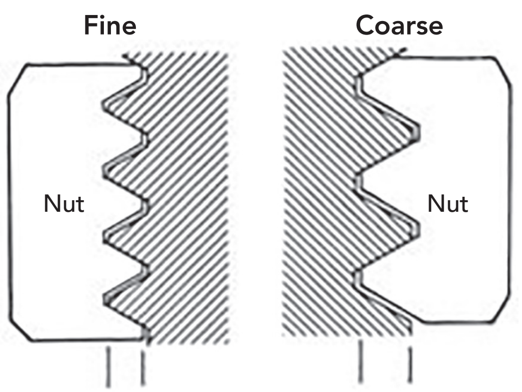
Engaged Thread Material Governs Strength

The thread profile is extremely important in the ability of mating threads (nut and bolt) to take high thread tension loads without stripping.

The thread can be considered as a triangle of material with an apex and a root. There is less material in the thread apex to sustain a load.

Therefore, it is better to have more flank engagement of threads than less (as long as the threads don't bind). Flank engagement is especially important since nuts tend to dilate under high loads.

Coarse threads offer more latitude and reliability because greater flank engagement can be achieved as the following illustrates.



Thread Flank Engagement

Another consideration is to look at threads as a helical inclined plane. The longer the "inclined plane" thread contact area of mating thread members, the greater the distribution of load and the less stress concentration on any individual thread section.

For this reason, nuts should be made from material that enables threads to "give" and thereby engage more thread material to carry the load.

Coarse Threads vs. Fine Threads

Thread load and stress concentrations are lower in standard coarse thread fasteners than in fine thread fasteners. Flank engagement is also greater because coarse threads are deeper. As a result, coarse threads are preferable to fine threads except for applications where fine adjustments are required. Coarse threads have greater resistance to striping and consequently a greater portion of their strength goes into making a stronger assembly.

Faster Assembly Time

Coarse thread fasteners tighten with only two-thirds the revolutions needed for fine threads. This helps speed assembly time. Also, coarse thread bolts enter nuts or mating holes with less tendency to cross thread when not accurately positioned. In hard-to-reach areas, this ease of thread starting can be helpful. Another consideration is that coarse threads need less handling care since they are less apt to be damaged.

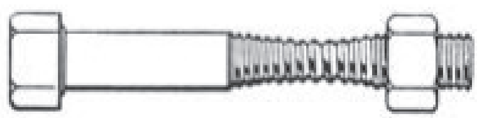
All in all, coarse threaded standard fasteners are usually the best choice for an assembly because of their additional clamping strength and the production savings they bring in assembly.

Stripping Strength vs. Tensile Strength

If a long, threaded anchor or nut made from *Delrin®* is mated to a steel screw and tensile tested, the screw will break before the plastic nut strips.

This seems incredible when you consider that Grade 2 steel has seven times the tensile strength of *Delrin®*. A practical explanation is found in a study of thread geometry and behavior under load.

Break Location
Bolts almost always break at an unengaged thread root because this is the *minor diameter* of the external thread and, therefore, the smallest cross section. It is the weakest link.



When equally loaded, a bolt experiences the same tension along its entire *grip length*. Tension is the same at any cross-sectional plane. Overloaded, the weakest area is the first to strain beyond its *ultimate strength* with the result that the bolt "necks" down and finally snaps.

Thread tension causes a plastic adjustment of the nut threads. Under tension, the bolt load distributes itself along the entire height of the nut. Thus, shearing stress on an individual internal thread is far less than on a plane across the *minor diameter* of an external thread.

That's why, paradoxically, a nut of weaker material can match the *proof load* of a stronger screw. A *Delrin®* nut cannot be torqued on a steel bolt to any degree of tightness without failing. This example shows the importance of the plastic thread adjustment available with the use of a softer nut.

LOCKING FASTENERS

Broad principles of locking fasteners are covered in this section. For more complete information on the use of locking fasteners, please check with a Supply Technologies technical representative.

Locking Power

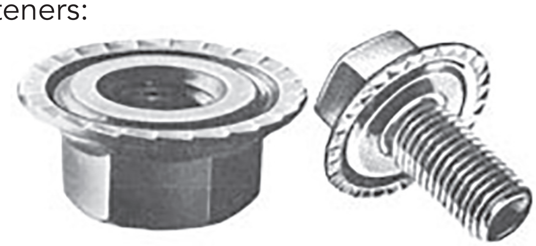
The locking action of self-locking fasteners depends on an increase in thread friction, an integrated anchoring device, or both.

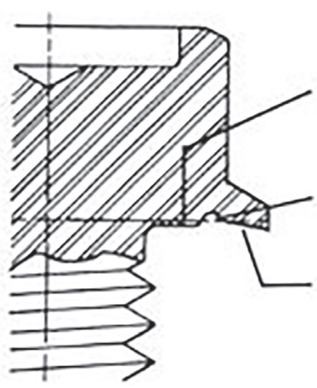
There are three basic types of Locking Fasteners:
1) Free Spinning
2) *Prevailing Torque* Resistance
3) *Prevailing Torque* Chemical Reaction

Free Spinning Locking Fasteners turn easily until they begin to seat. Further tightening brings the locking action into play. The locking action is caused by pressure on the *bearing surface* of the fastener and the work being held.

Since these elements must be tight to lock, they get extra resistance to back-off from the friction caused by thread tension.

Below is an example of Free Spinning Locking Fasteners:

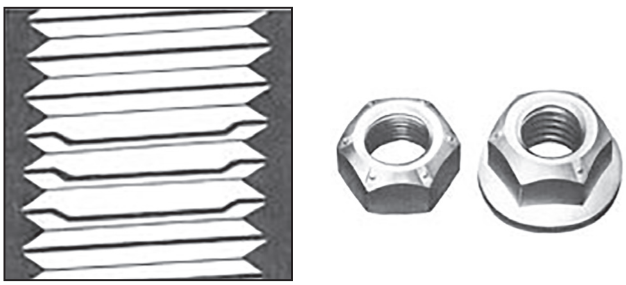




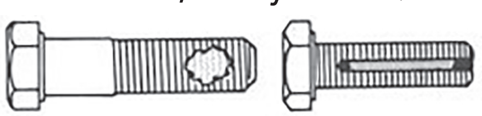
Prevailing Torque (PT) Resistance Fasteners develop resistance to back-off as soon as the friction device reaches the mating thread. The most common *prevailing torque* fasteners are all-metal deflected thread, nylon insert, and nylon patch. All rely on the spring action of the locking device to develop *prevailing torque*.

Prevailing torque screws utilize a deflected external thread to create a friction lock.

The pictures below reflect the various locking mechanisms.



External Thread Deflection, Internal Thread Deflection, and Nylon Patch/Inserts



Prevailing Torque Chemical Reaction Locking Fasteners use epoxy, glues, or resins to hold mating fasteners together with an adhesive action. The chemicals may have separate bands of adhesives and hardeners or the hardener may be microencapsulated with the adhesive. *Prevailing torque* chemical reaction locking fasteners offer high *breakaway torques* and minimal prevailing-on torque, but once dissembled, their locking device has limited reuse.

Use and Reuse of Locking Fasteners

Importance of Locking Fastener Reuse
The frictional reaction between two loaded and sliding surfaces will cause wear and a given amount of permanent material deformation. Because of this, mechanical *prevailing torque* elements will lose some of their holding power during and after initial assembly and tightening. The amount of performance drop-off from first assembly to subsequent reuse is a good measure of locking element reusability. Performance drop-off signifies the degree of *prevailing torque* element wear and spring-back potential.

Specifications such as ASME B18.16.6, IFI 100/107, and IFI 124/125 have been established to measure performance capabilities of competitive *prevailing torque* fasteners.

Where to use Locking Fasteners
Prevailing torque fasteners should be used in assemblies where effective thread tension may be overcome by a sustained vibratory force or momentary shock overload. Experience has shown that conventional fasteners perform adequately in static joints and in some instances may be suitable under limited dynamic conditions.

Selecting the Right Prevailing Torque (PT) Nut Grade

There are three basic inch and metric *prevailing torque* nut grades manufactured in the United States. Each grade has been developed for use with a specific grade of screw:

PT Nut Grade	Screw Grade (Inch)
A	2
B	5
C	8

PT Nut Grade	Screw Grade (Metric)
PC5	PC5.8
PC8	PC8.8
PC10	PC10.9

Commercially available Grade A *prevailing torque* nuts are not heat treated. Grade B *prevailing torque* nuts may or may not be heat treated (manufacturer's option). Grade C *prevailing torque* nuts are heat treated to achieve desired hardness and strength.

Prevailing Torque Nut Design

Prevailing torque nuts are designed and manufactured so their stress levels are best suited to the mating screw grade. Thread hardess, nut body dilation, and friction element deflection are key design factors in matching *prevailing torque* nut and screw.

Prevailing Torque (PT) Nut Selection Critical

An example of improper application would be to use a Grade A or B *prevailing torque* nut on a Grade 8 cap screw. Conversely, using a Grade B or C *prevailing torque* nut on a Grade 2 cap screw is a waste of money. In either example, improper stressing of one member (bolt or nut) will occur. In the first example, the Grade A or Grade B *prevailing torque* nut will be overstressed if tightened with Grade 8 cap screw torques. In the second example, the Grade B or Grade C *prevailing torque* nut would overstress the Grade 2 cap screw if tightened to its full potential.

Of equal importance is the possible mismatch of material hardness between the screw member and friction element. Such a mismatch can cause exceptional wear to softer threaded elements with a corresponding loss in nut performance.

Strength Identification

It is necessary to know the strength of the fastener prior to its application to ensure that the friction producing element will work properly when the fastener is loaded. From a cost standpoint, why pay for strength that is not being used? From a strength standpoint, an inferior grade fastener could be subject to possible premature failure.

Manufacturer Identification

All domestic manufacturers identify their product by type and strength grade and by manufacturer.

Prevailing Torque Screws

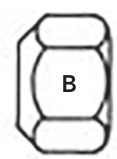
Strength grades and the producer's identification appear on the heads of domestically made *prevailing torque* screws. See **pages 7 and 8** for strength grade markings.

Prevailing Torque Nuts

On *prevailing torque* nuts, the grade of nut must be identified by one of three different sets of markings that denote the strength level and manufacturer. Comparable screw grades are shown below.

Nut	Screw
Grade A	Grade 2
Grade B	Grade 5
Grade C	Grade 8

The three common methods of identifying locknuts are as follows:



60°- 90° Included Angle

Grade Identification	**Grade Identification**	**Grade Identification**
Grade A no marks Grade B three marks Grade C six marks Marks need not be located at corners	Grade A no marks Grade B letter B Grade C letter C	Grade A no notches Grade B one circumferential notch Grade C two circumferential notches

Importance of Breakloose Torque

The amount of torque required to start disassembly of an axially loaded fastener is called the *breakloose torque*. Resistance to *breakloose torque* is usually thought to be synonymous with the *preload* in a bolted assembly. Under normal conditions, bolt *preload* is sufficient to maintain joint integrity. Normally the torque required to loosen an assembly is less than the torque required to tighten. The primary need then for any locking device is not only to resist coming apart, but to increase the joint's resistance to initial loosening when joint integrity has been jeopardized.

In a good *prevailing torque* fastener, the effective *breakloose torque* of the assembly is increased by the added locking feature.

Breakaway Torque – For Joints Not Under Load
Any bolted assembly that is not under initial *preload*, or has lost its *preload*, may start to disassemble when subjected to external dynamic forces. The static torsional resistance to disassembly in a non-axially loaded locking fastener is called *breakaway torque*.

Breakaway torque is normally less than *breakloose torque*, but greater than *prevailing torque*. The merits of *breakaway torque* are best exemplified in assemblies where *preload* is not allowed. Examples include most spring-loaded assemblies, gasketed joints, and the adjustment of positioning assemblies.

Prevailing Off Torque – Last Line of Defense
Conventional bolt and nut assemblies are free spinning. This means that the fastener will assemble and disassemble without applying torque. *Prevailing torque* is the amount of rotational force required to keep the fastener in motion during assembly or disassembly. The importance of *prevailing off torque* is resistance to disassembly. It is the last line of defense against separation of an assembled joint.

TAPPING SCREWS

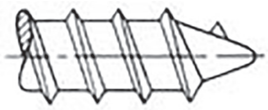
Tapping screws are *hardened* screws that form mating threads when driven into the material being jointed. They are available in a wide variety of *tapping* styles, head styles, drives and locking features.

Tapping styles can be grouped into three categories:

- Thread-Forming
- Thread-Cutting
- Thread-Rolling

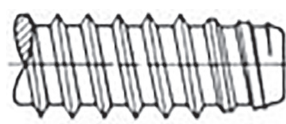
Thread-Forming Screws

Thread-Forming tapping screws are generally used in materials where large internal stresses are permissible or desirable to increase resistance to loosening.



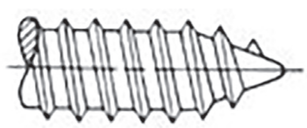
(Type A)

Type A – a spaced thread screw with a gimlet point. Primarily used in light sheet metal, resin-impregnated plywood, or compressed composition materials. Often used in place of wood screws because of its quicker driving time, full-length thread, and larger thread profile. Type AB screws usually are recommended over Type A.



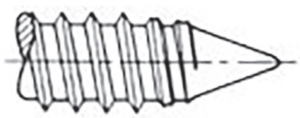
(Type B)

Type B – has a finer thread pitch than Type A and a blunt point. Primarily used for light and heavy sheet metal, nonferrous castings, plastics, resin-impregnated plywood, and compressed compositions. Recommended for heavier material thickness than Type AB because its gradual point taper starts more easily than Type AB, which starts with a full thread diameter.



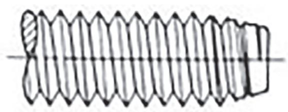
(Type AB)

Type AB – has the thread pitch of Type B and the gimlet point of Type A. Primarily used for thin sheet metal, resin-impregnated plywood, compressed compositions, and nonferrous castings. Recommended over Type A especially for use in brittle materials such as plastics and zinc *die* castings.



(Type BP)

Type BP – has a conical point. Primarily used for piercing fabrics or in assemblies where holes may be misaligned. Type AB screws are recommended over Type BP.



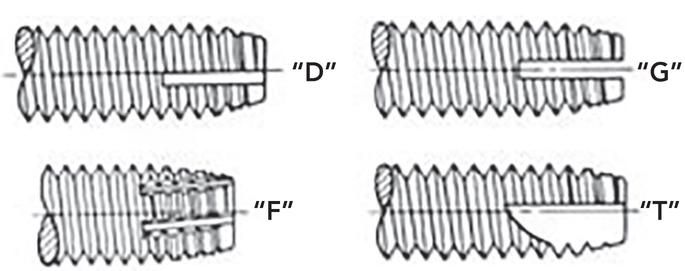
(Type C)

Type C – has machine screw threads with a blunt point and tapered entering threads. Primarily used to *tap* into thicker metallic sections than the Type AB thread series and where chips from thread-cutting screws are objectionable. Extreme driving torques may be required where long thread engagement is involved. As a result, thread-rolling screws have frequently replaced Type C screws for difficult applications.

Thread-Cutting Screws

Thread-cutting screws are used in materials where disruptive internal stresses are undesirable or where excessive driving torques would be encountered if thread-forming screws were used.

Types D, F, G and T – have machine screw threads and diameter pitch combinations, blunt point and tapered entering threads. The entering threads have one or more cutting edges and chip cavities. Primarily used for aluminum, zinc, or lead *die* castings, steel sheets and shapes, cast iron, brass and plastics.



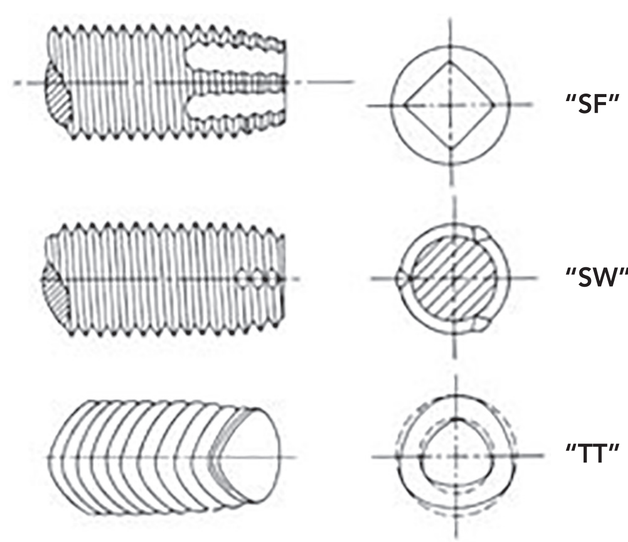
Types BF and BT – are Type B tapping screws with the addition of cutting edges and chip cavities. Primarily used for materials such as plastics and similar compositions.

Thread-Rolling Screws

High-performance thread-rolling screws are a more sophisticated version of thread-forming tapping screws. Available in coarse or fine threads, thread-rolling screws overcome most problems encountered with other tapping screws. Types SF, SW, and TT are described in published *standards*.

Compared with thread-forming and thread-cutting tapping screws, thread-rolling screws have three advantages:

1. Easier starting due to their unique points and body configuration
2. Less driving torque required
3. The cold working of material during thread forming improves joint strength

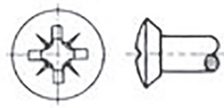


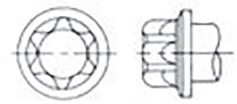
Head Styles and Drives

Pan Head tapping screws are recommended for most applications. Severe torque conditions may dictate use of a drive such as Cross or Six-Lobe or a Hex Head.


Slotted Pan Head


Hex Head


Cross-Recess


Six-Lobe

BOLT INSTALLATION

Proper Loading of Bolts

The *preload*, or residual tension, in a tightened bolt means more to assembly strength than actual strength of the bolt itself.

In a joint, a bolt torqued to its proper load level can resist a maximum amount of external load without loosening. Designers can take advantage of this fact to assure correct bolt loading and at the same time cut costs.

Example: One designer calculated that a particular truck frame needed high-strength bolts with a minimum ½ in. diameter bolts. He specified ⅝ in. diameter bolts. On the assembly line, these bolts were torqued to 100 lb. ft., but a minimum 200 lb. ft. was required for proper residual tension. Use of a ½ in. diameter bolt at 100 lb. ft. would have provided a stronger assembly at less cost.

Example: The bucket on earth moving equipment was always coming loose. To correct the problem the design engineer specified a 1¼ in. diameter bolt, but to no avail. The impact wrench used to assemble the joint was supplying far too little torque for this size fastener. A return to the original ¾ in. bolt assembled at 350 lb. ft. torque solved the problem.

Assembled bolts are tightest when stressed as near as possible to their *elastic limit*.

Bolt Load Levels

Tightening a bolt beyond its proportional limit will deform the bolt plastically. The stress level that causes permanent set is called the bolt's *yield strength*.

Overtightening

In a static (stationary or nonmoving) joint, a bolt can be tightened up to *yield strength* and beyond. This is recommended for permanent connections.

When a fastener is tightened up to and beyond its *yield strength*, *pitch* of the external threads increases as the screw stretches. Conversely, *pitch* of the internal nut threads decreases.

This results in a cumulative off-lead condition. Normally it's not practical to reuse a bolt and/or nut stressed beyond its *elastic limit*.

Recommended stress levels for non-permanent connections are as follows:

Inch	Metric
SAE Grade 2 – 40,000 psi	PC5.8 – 285 MPa
Grade 5 – 60,000 psi	PC8.8 – 450 MPa
Grade 8 – 90,000 psi	PC10.9 – 620 MPa

Although below *yield strength*, these levels are approximately 75% of the bolt's *proof load* and provide a high *preload* for good assurance against loosening and fatigue.

For structural steel connections, A325 bolts (equivalent to SAE Grade 5) should be tightened to *proof load* or beyond (85,000 psi up to 1 in. diameter 74,000 psi for 1⅛ to 1½ in. diameter sizes). A490 bolts (equivalent to SAE Grade 8) should be tightened beyond 120,000 psi.

Always specify that the assembler torque a fastener to achieve the recommended tension (clamping load), not the torque level. Tension is the usable "muscle." Not used, money is being wasted and the joint isn't as strong as it should be.

Tighten Bolts to Yield Strength

Bolts should be tightened to *yield strength*. Here's why:

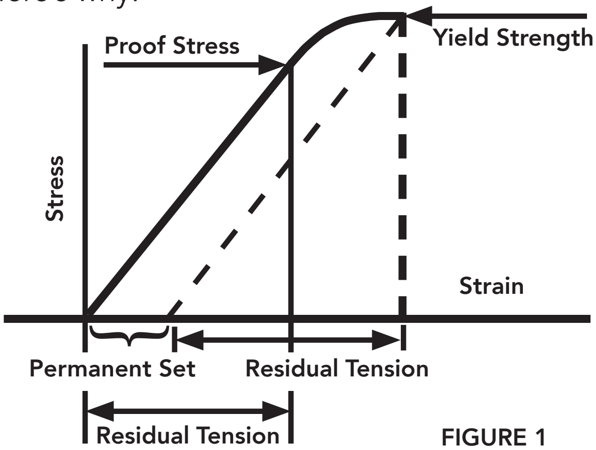


FIGURE 1

Up to proof stress, the strain in a bolt is proportional to stress. Beyond the proof stress or *elastic limit*, the bolt goes into its "plastic range" and some permanent stretch takes place (Fig.1). Although the bolt will not return to original length, the residual tension is almost fully maintained. This is the force that keeps a bolt tight and determines joint strength.

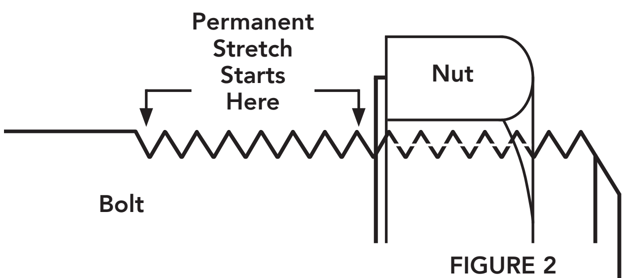


FIGURE 2

Permanent set starts at the section with the highest unit stress, the unengaged threads (Fig. 2). Ultimately, this set throws thread pitch off, locking the nut, and subjecting the bolt to torsion (rather than further tightening). This force disappears with wrench removal. Bolts can be torqued well into their plastic range, provided they won't be reused or need adjustment.

Safety Factors

A bolt with a calculated *yield strength* of four times the working load does not automatically mean a safety factor of 4. The bolt must actually be tightened to X 4 working load to get a safety factor of 4.

The reason is that rigidly fastened members can be loaded externally to the full value of residual tension in bolts without any separation or significant extra bolt stress.

Example: A bolt has been selected for a 5,000 lb. working load. In order to obtain X 4 safety factor, it is necessary to use a 20,000 lb. capacity bolt and tighten it to 20,000 lb. tension. If this bolt is tightened to 10,000 lb., a larger external load will cause loosening and progressive bolt failure from fatigue. In reality, the safety factor is only X 2.

Assembly Important

Safety factors then are not established on the drawing board. They can only be put into the product by the person with the wrench. A bolt is no better than the supervision of its tightening.

Flexible Joints

Flexible joints should just be tightened to their working load. Select a bolt capable of meeting the working load plus added stress multiplied by the safety factor.

Avoiding Bolt Failure under Dynamic Loads

A dynamic loading in a tightened bolt may vary from no stress at all to that exceeding bolt *preload*.

The connecting rod in a reciprocating engine is a classic example of dynamic loading. However, such cyclic stress is encountered wherever fastened members move or vibrate.

It has been shown that when the fluctuating stress approaches or exceeds actual bolt tension, early fatigue failure can be expected.

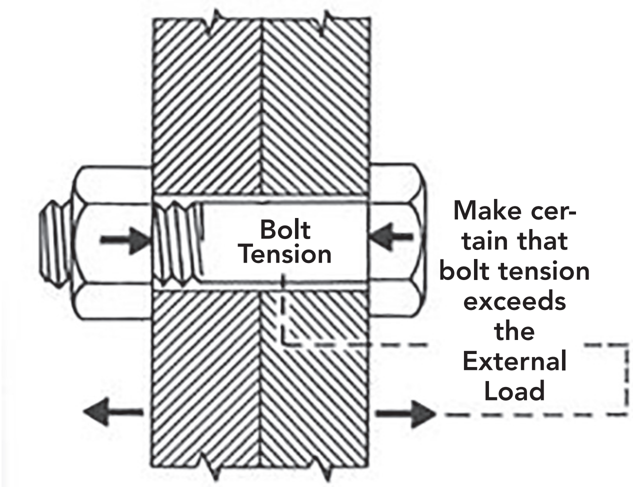
Demonstration

In tests, bolts tightened to a 1,420 lb. tension and stressed cyclically to 9,215 lb. failed after 5,960 cycles. Identical bolts tightened to 8,420 lb. and cyclically stressed to the same 9,215 lb. went 4.65 million cycles before failure.

If the bolts had been tightened beyond 9,215 lb., it would have been impractical to cycle them to failure.

To avoid fatigue failure, make sure bolt tension exceeds the known or estimated maximum dynamic load. This way, the bolt's life under dynamic loading will approach its life under static loading.

To avoid fastener failure, take advantage of the high residual tension available in today's high-strength bolts and screws.



Role of Friction in Fastening

Friction is both an asset and a detriment. Friction helps make a better joint but friction also complicates and varies the torque/tension relationship.

Fastener friction is no problem until high thread tension is developed. Then friction becomes the major resistance to further rotation. Friction occurs between head and *bearing surfaces* and between mating threads.

Most of the torque required for fastening goes to overcoming this torsional resistance; only a minor part to increase tension.

Lubrication, either from plating or oil, changes this ratio. That's why the same torque values can't be used for dry, oiled, and plated fasteners. The best approach is to test for the proper torque with a pilot assembly to get full usable strength from high-strength fasteners.

Joint Friction
In structural engineering, a friction-type joint is one where bolts have been tightened to transfer the load from one connected member to the other by friction, rather than by bearing on the bolts.

In a friction joint, there is no slippage, no loosening. The full net section of the structural member becomes available to support the shear load.

BOLT INSTALLATION – TORQUE

Fasteners take Greatest Stress during Wrenching

Two forces put stress on fasteners as they tighten: tension due to bolt stretch; torsion due to friction. However, only tension remains after wrenching. In a rigid joint, if this tension exceeds external forces, the fastener will not experience further strain, and will therefore not loosen or fail.

Why Some Failures?
Unusual and unforeseen loads can cause trouble.

The instant the load exceeds residual tension, additional stress is placed on the fastener that can cause joint separation. These loads can cause loosening, leading to stress change, which in turn causes delayed or fatigue failure. This is why it is important to torque bolts tight and the tighter the better.

A flexible joint is an exception. With high cyclic loading, loosening and fatigue cause trouble in a flexible joint. Since a metal gasket should not be tightened too much, sometimes the only remedy is to remove the flexible element and install a rigid joint. (An example would be a metal-to-metal flange connection instead of a gasketed one.)

Determining the Right Torque for Bolts

"What is the right torque for bolts?" is a tough question. There are too many variable conditions. The following may help:

Bolts take two stresses during wrenching: torsion and tension. Correct tension is the goal. Torsion is a necessary evil due to friction. Probably 90% of applied torque goes to overcome friction.

With the friction factor changed by lubrication, plating, etc., it is hard to determine the torque needed to produce a given tension. However, a useful empirical formula exists for normal friction conditions.

Lb.-In. Torque =
0.2 x Bolt Diameter x Bolt Tension

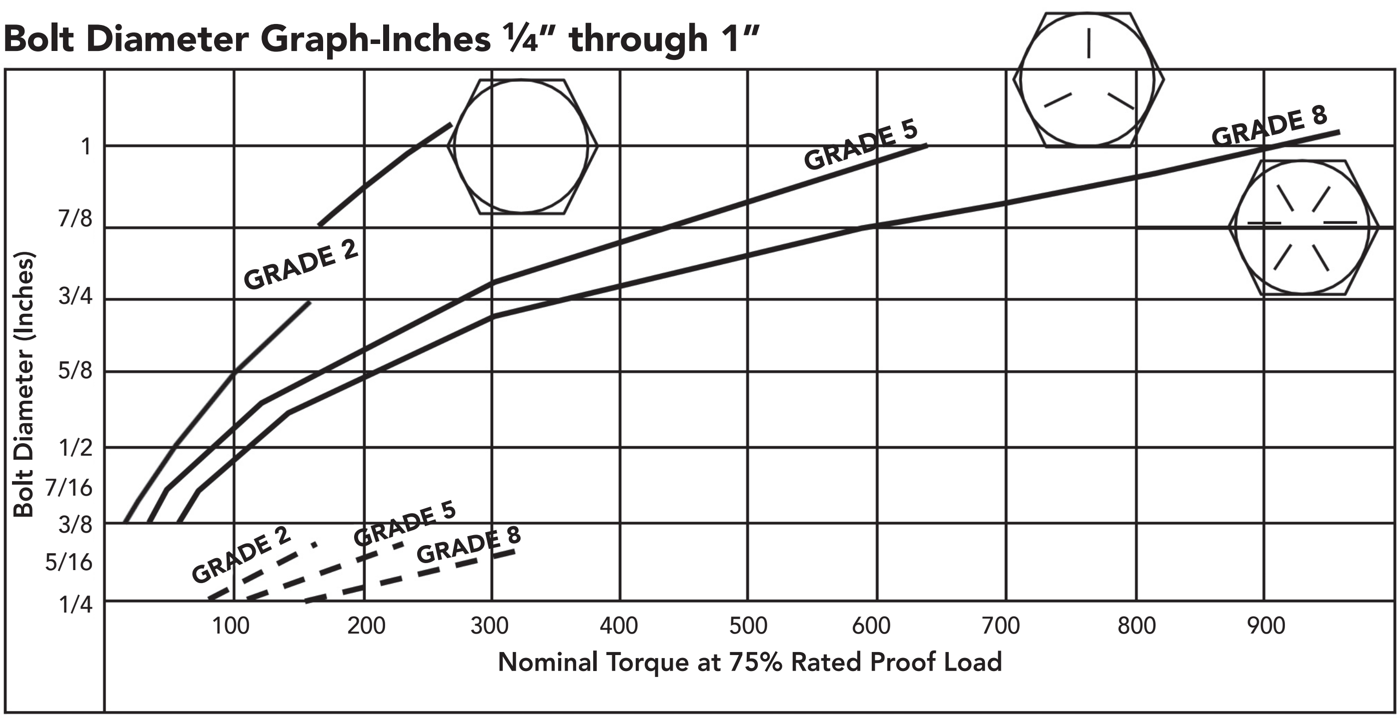
Tests show that the 0.2 torque coefficient is approximately constant for normal friction conditions, all fastener diameters, and for coarse or fine threads.

The average deviation is about 7%. When conditions are "normal" the only sure way to check torque is to set up a pilot assembly and try it out.

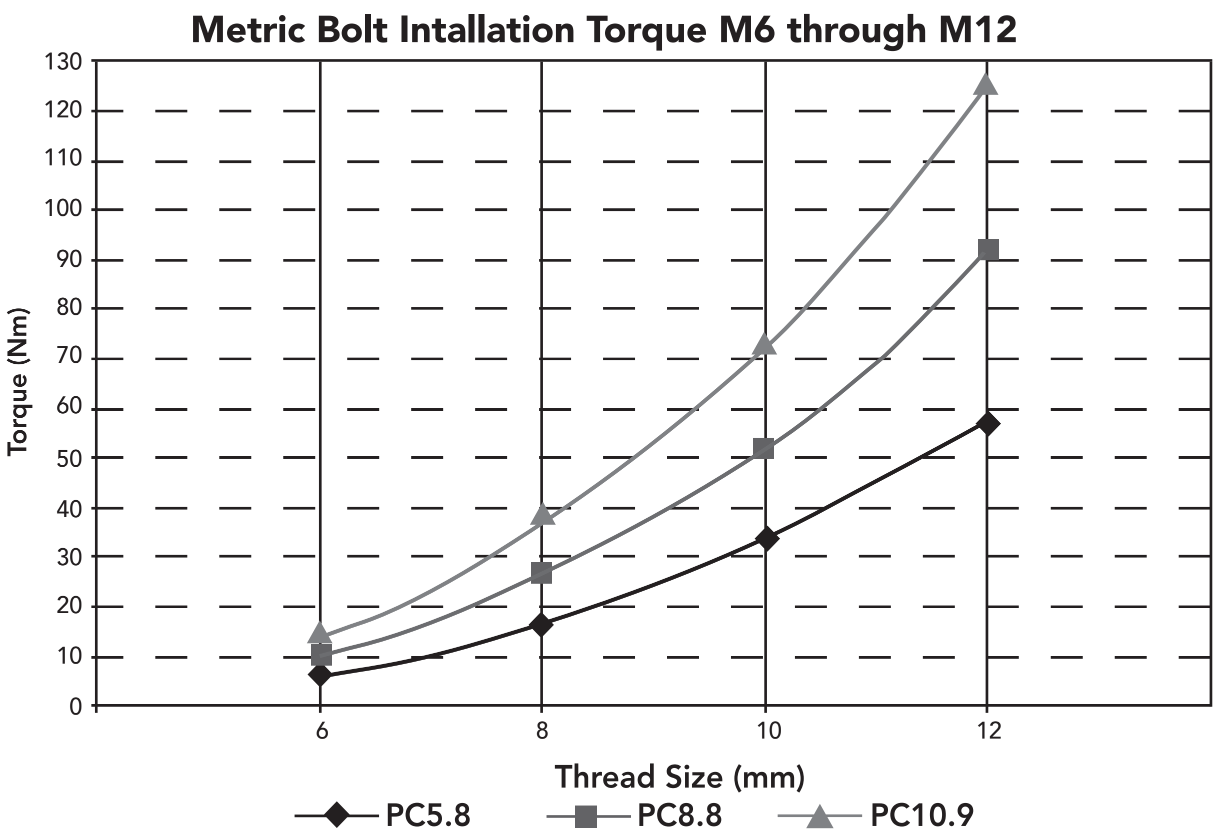
In pilot testing for rigid joints, first tighten a few bolts with the torque wrench to failure. Then set the torque at 60% of that load, or even at *yield strength*, since the torsion component vanishes when tightening ceases leaving the bolt under tension only (which is well below ultimate strength). Torque is the vehicle to create bolt tension.

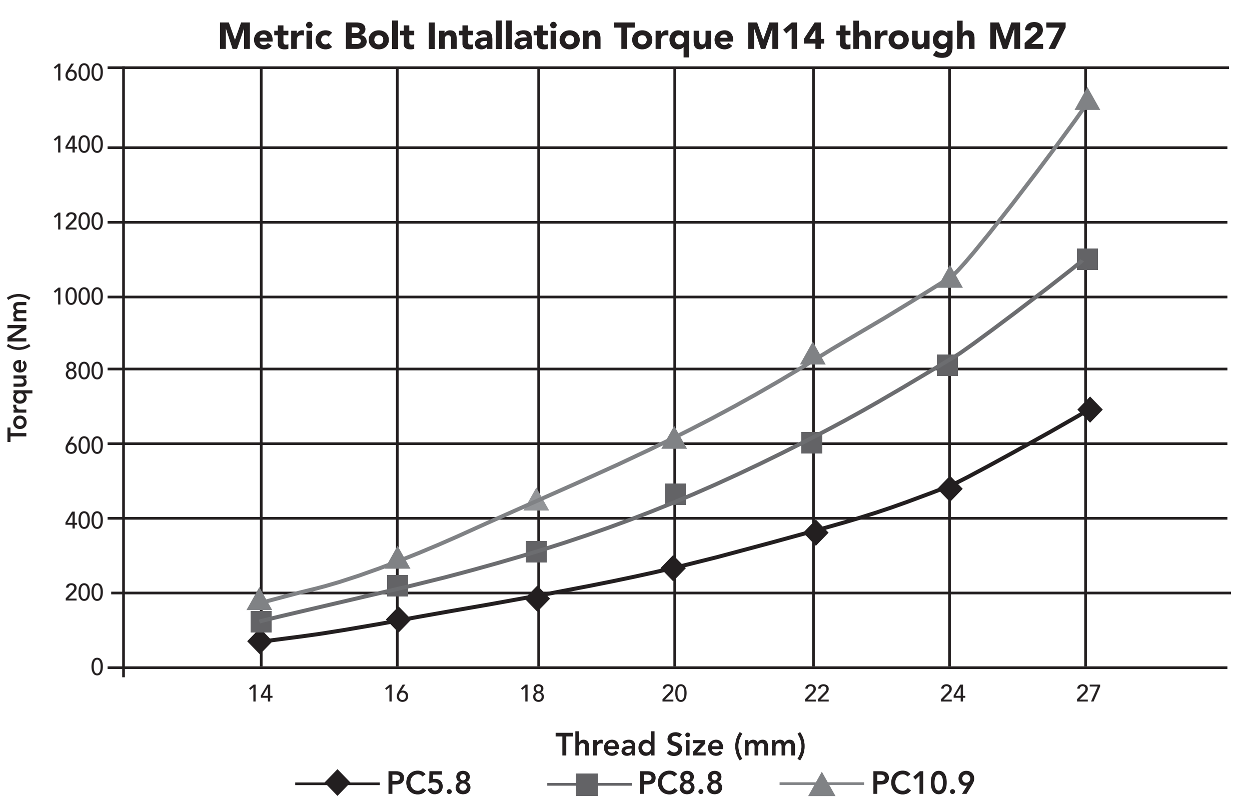
The graphs on pages **26 and 27** suggest torque for various size bolts. Remember, these curves are only starting points in determining proper torque-tension relationships. (Note: The break in Grade 2 torque between the ¾ in. and ⅞ in. diameters reflects the drop in the *proof load* specification from 55,000 psi to 33,000 psi.)

Bolt Diameter Graph-Inches ¼" through 1"



Solid lines are in Ft-Lbs — — Dashed lines are in In-Lbs


Metric Bolt Intallation Torque M6 through M12
130
120
110
100
90
80
70
60
50
40
30
20
10
0
Torque (Nm)
6
8
10
12
Thread Size (mm)
PC5.8
PC8.8
PC10.9


Metric Bolt Intallation Torque M14 through M27
1600
1400
1200
1000
800
600
400
200
0
Torque (Nm)
14
16
18
20
22
24
27
Thread Size (mm)
PC5.8
PC8.8
PC10.9

Bolt Tension from "Turn-of-Nut" Method

Product designers think in terms of torque when specifying how much to *preload* a bolt.

Structural steel fabricators customarily use the "Turn-of-Nut" method.

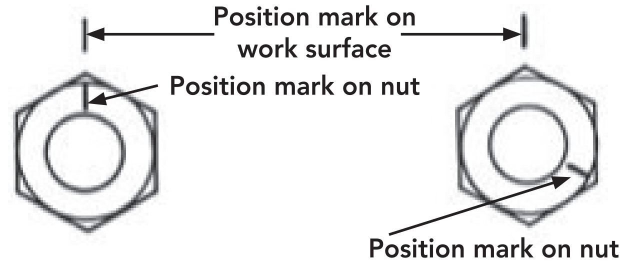
To understand the "Turn-of-Nut" method, first consider that within its *elastic limit*, a steel bolt stretched 0.001 in. for each inch of *grip length* is loaded to 30,000 psi.

As the nut advances, the bolt is stretched and loaded in tension. Since thread *pitch* is fixed, an increment of nut turn from the snug tight position produces a fixed increment in tension. Coarse thread nuts need less turning than fine thread nuts since they have fewer threads per inch.

In steel structures, nuts are coarse thread exclusively. They are wrenched from ⅓ to 1 full turn (depending on length of bolt) to load the bolt to 100% of its *proof load*, or beyond, into the bolt's plastic range. The significance?

The "Turn-of-Nut" method results in a safe, strong joint because it eliminates the effect of variation in coefficient of friction due to presence or absence of lubrication on threads or *bearing surfaces*.

"Turn-of-Nut" Method



Step 1: Tighten nut to snug fit

Step 2: Turn nut additional ⅓ turn (for bolts up to 4 diameters long, bearing faces normal to bolt axes). Use ½ turn for bolts 4 to 8 diameters and ⅔ turn for bolts 8 to 12 diameters.

CORROSION RESISTANCE

Fasteners must be coated or made from a corrosion-resistant metal to stand up in a corrosive environment.

Metallic Coatings

Outdoor corrosion resistance can be improved by *electroplating* or galvanizing the fastener. Mechanical or hot dip galvanizing processes offer much thicker zinc *coatings* than commercial electroplated zinc, hence, they have greater resistance to atmospheric corrosion. They usually cost more than the *electroplate*. Specific corrosion conditions may require other metallic or bi-metallic, *coatings* such as zinc nickel.

Organic Coatings

Corrosion resistance can also be increased through the application of organic *coatings*, usually phosphate and oil. Results can match those of electroplated metallic *finishes* for many service conditions. Recent developments of conversion *finishes* promise increased protection from organic *coatings*.

Corrosion Resistant Metals

Stainless steel, aluminum, and silicon bronze offer distinct advantages in given corrosive environments.

Silicon bronze is popular for electrical uses due to its unusual strength and resistance to stress corrosion.

Aluminum fasteners offer lightweight as well as excellent conductivity. Anodizing improves aluminum's corrosion resistance and permits the use of various colors.

Widely used for fasteners, the 300 Series stainless steels assure good strength and excellent corrosion resistance in most atmospheres.

Fastener Coatings

Salt spray tests of the various metallic *coatings* used on fasteners do not always give a true picture. In actual service, accelerated test results are not always borne out. The reason is that tests can only approximate, but not duplicate, atmospheric or service conditions.

Organic and inorganic coating can be low chrome or no chrome, while providing excellent corrosion resistance.

Mechanical galvanizing offers great endurance under most conditions, followed by hot-dip galvanizing.

Electrodeposited zinc is the next most practical, providing good appearance, controlled tolerance at threads, and ability to take high bolt tensions.

Where appearance is secondary, **phosphate and oil coating** offer protection under relatively severe conditions. For general applications, the rust prevention of **black oxide coatings** generally proves satisfactory.

Various ***conversion coatings*** are available which tend to extend the life of zinc and phosphate *finishes.*

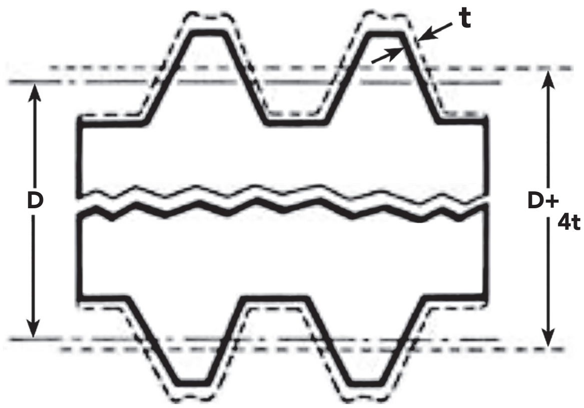
Chromium should be considered more for appearance but does offer extended protection.

RoHS, REACH, and other environmental regulations restricting the use of certain heavy metals, carcinogens, and other chemicals persistent in the environment (i.e. hexavalent chromium, cadmium, etc.) need to be considered before selecting a surface *finish*.

Effect of Plating on Thread Fits

With respect to thread geometry, one unit of plating thickness on screw thread flanks will add X 4 that 'thickness' dimension to the screw's *pitch diameter*.

Illustration: The standard and minimum 0.00015 in. of plating "t" adds 0.0006 in. or more to the screw *pitch diameter* "D."



Moreover, because of plating both characteristics, the average deposit varies from thread tips to roots. On long fasteners, plate build-up is thicker at ends and thinner towards the mid-length, adversely affecting the screw's starting threads.

Specifying heavier-than-normal plating for threaded fasteners can make assembly difficult, even impossible, unless special undercut threads are specified.

Gaging Procedures

Thread limits for standard external thread Class 2A, apply prior to *coating*. Threads after plating or *coating* should go into a Class 3A GO thread gage and must not go into a class 2A minimum material or NO GO gage to assure ease of assembly with mating parts. However, if the plated or unplated screw will enter a Class 2A NO GO gage, it should be rejected. Its entry indicates excessive *coating* or plating or there is insufficient thread depth to develop full strength of the fastener.

Stainless Steel Fasteners

Stainless steels are iron alloys containing from 12% to 30% chromium and from 0% to 20% nickel. All stainless steels are not alike, some will corrode where others will not.

They also vary in strength, in temperature resistance, and in workability. In turn, fasteners made from them vary, depending on the type of stainless steel used.

400 Series
Steels in this group are straight chromium types. All are magnetic. Some are heat treatable, others are not. They can be cold formed and cold worked with good results.

Fasteners made from these steels are easier to head and thread. They are the most economical in the stainless steel group but are generally used where corrosion-resistance requirements are not severe.

300 Series
Sometimes called 18-8 stainless (18% chromium 8% nickel) the 300 Series of alloys offers the highest degree of corrosion resistance. 300 Series stainless steels work-harden rapidly, which greatly reduces tool and *die* life.

The choice of a stainless material should be based on the application. Be sure to specify the grade or the service requirement, not just "stainless."

Due to their low carbon content, 300 Series stainless steel cannot be heat treated to improve strength.

METRIC FASTENERS

The International Organization for Standardization (ISO) has created the most commonly used metric fastener *standards* worldwide.

Metric Fasteners
Language is based on the existing comprehensive SI (International System of Units) Metric System. Metric units for measurement of fasteners are shown in the table below.

Fastener Quantities and their Metric Units of Measurement

Quantity	Unit of Measurement	Symbol
Area	square millimeter	mm2
Density	kilograms per cubic meter	kg/m3
Force	kilonewton (1)	kN
Length	millimeter	mm
Mass (weight)	kilogram (2)	kg
Plane angle	degree	°
Plating thickness	micrometer	µm
Stress (pressure)	megapascal (3) (4)	MPa
Surface finish	micrometer	µm
Temperature	degree Celsius	˚C
Time	second	s
Torque	newton-meter	Nm
Volume (6)	cubic millimeter	mm3

Metric Fasteners
The table below compares the Unified-Inch and Metric Diameter-pitch combinations:

Unified-Inch and Metric Compared

Unified-Inch				Screw Thread Standard
Coarse		Fine		
Nom. Size	Threads/ (in.)	Nom. Size	Threads/ (in.)	Major Thread Dia. (mm) Pitch (mm)
—	—	0	-80	M1.6x0.35
1	-64	1	-72	—
2	-56	2	-64	M2x0.4
3	-48	3	-56	M2.5x0.45
4	-40	4	-48	—
5	-40	5	-44	M3x0.5
6	-32	6	-40	M3.5x0.6
8	-32	8	-36	M4x0.7
10	-24	10	-32	M5x0.8
12	-24	12	-28	—
1/4	-20	1/4	-28	M6.3x1
5/16	-18	5/16	-24	M8x1.26
3/8	-16	3/8	-24	M10x1.5
7/16	-14	7/16	-20	—
1/2	-13	1/2	-20	M12x1.75
9/16	-12	9/16	-18	M14x2
5/8	-11	5/8	-18	M16x2
3/4	-10	3/4	-16	—
7/8	-9	7/8	-14	M20x2.5
1	-8	1	-12	M24x3
1-1/8	-7	1-1/8	-12	—
1-1/4	-7	1-1/4	-12	M30x2.5
1-3/8	-6	1-3/8	-12	M33x3.5
1-1/2	-6	1-1/2	-12	M36x4
—	—	—		—
1-3/4	-5	—		M42x4.5
—	—	—		—
2	-4-1/2	—		M48x5
2-1/4	-4-1/2	—		M56x5.5
2-1/2	-4	—		M64x6
2-3/4	-4	—		—
3	-4	—		M72x6
3-1/4	-4	—		M80x6
3-1/2	-4	—		—
3-3/4	-4	—		M90x6
4	-4	—		M100x6

Metric screw thread pitch sizes fall in between the coarse and fine thread series of the Inch System.

All metric thread dimensions are referenced from the *major diameter* instead of the *pitch diameter* used in the inch series. The "thread limits of size" concept for inch fasteners has been replaced in metric by "boundary profiles for gaging."

Materials and Mechanical Specifications

Shown below are seven metric material *property classes* for bolt products and four for nuts.

Metric Property Classes

BOLTS, SCREWS, AND STUDS					
Property Class	**SAE Grade[1]**	**Nominal dia. (mm)**	**Proof load (MPa)**	**Tensile strength (min. MPa)**	**Rockwell hardness min. max.**
4.6	1	M5 thru M36	225	400	B67 B100
4.8	–	M1.6 thru M16	310	420	B71 B100
5.8	2	M5 thru M24	380	520	B82 B100
8.8	5	M16 thru M36	600	830	C24 C34
9.8[2]	–	M1.6 thru M16	650	900	C27 C36
10.9	8	M5 thru M36	830	1040	C33 C39
12.9	–	M1.6 thru M36	970	1220	C39 C44

HEX NUTS					
Property Class[3]	**SAE Grade[1]**	**Nominal dia. (mm)**	**Proof load (MPa)**	**Mating bolt and screw class**	**Rockwell hardness min. max.**
5	2	M5 thru M36	225	5.8, 4.8, 4.6	C30 max
8	5	M5 thru M7 M8 thru M10 M11 thru M16 M17 thru M39	855 870 880 920	8.8, 5.8, 4.8	B92, C31 B98, C37
9	–	M1.6 thru M14 M5 thru M24 M16 thru M36	900 990 910	9.8, 8.8, 5.8, 4.8	B89, C30
10[2]	8	M5 thru M36	1040	10.9, 9.8, 8.8	C26, C36

1) To be used for guidance purposes only in selecting metric property classes.
2) This class is actually 9% stronger than SAE Grade 5 and ASTM A449.
3) Property Classes 5 & 9 are low carbon steel, not heat-treated. Property Class 10 is medium carbon steel, heat-treated.

English/Metric Conversions

To convert English and Metric units use the following factors:

Quantity	English Unit	Abbre-viation		Conversion Factor		Metric Unit	Symbol		Conversion Factor		English Unit
Length	Inch	in	x	25.4*	=	millimetre	mm	x	0.039370	=	in
Area	Square Inch	in^2	x	645.16*	=	square millimetre	mm^2	x	0.001550	=	in^2
Volume	Cubic Inch	in^3	x	16387.064*	=	cubic millimetre	mm^3	x	0.000061	=	in^3
Mass (weight)	Pounds	lb	x	0.453592	=	kilogram	kg	x	2.20462	=	lb
Force	Pounds (force)	lbf	x	4.44822	=	Newton	N	x	0.224809	=	lbf
Torque	Pound-feet	lb-ft	x	1.35582	=	Newton-metre	Nm	x	0.737562	=	lb-ft
Stress	Pounds per sq. in.	psi	x	0.006895	=	megapascal	MPa	x	145.038	=	psi
Temperature	°Fahrenheit	°F	x	5/9 (°F-32)	=	°Celsius	°C	x	1.8°C + 32	=	°F

Conversion Table: Inches-Millimeters

#	Inches	Decimal inches	mm
		.0984	**2.5**
		.0081	**3**
5	1/8	.1250	3.175
		.1378	**3.5**
6		.1380	3.505
		.1575	**4**
		.1640	4.166
		.1771	4.5
	3/16	.1875	4.763
10		.1900	4.826
		.1968	**5**
12		.2160	5.486
		.2362	6
		.2480	**6.3**
	1/4	.2500	6.35
		.2756	7
	5/16	.3125	7.937
		.3149	**8**
		.3543	9
	3/8	.3750	9.525
		.3937	**10**
		.4330	11
	7/16	.4375	11.112
		.4720	**12**
	1/2	.5000	12.7
		.5118	13

#	Inches	Decimal inches	mm
	9/16	.5625	14.287
		.5906	15
	5/8	.6250	15.875
		.6299	**16**
		.6693	17
		.7087	18
		.7480	19
	3/4	.7500	19.05
		.7874	**20**
		.8268	21
		.8661	22
	7/8	.8750	22.225
		.9055	23
		.9449	**24**
		.9843	25
	1	1.0000	25.4
		1.1811	**30**
		1.4173	**36**
	2	2.0000	50.8
	3	3.0000	76.2
	4	4.0000	101.6
	5	5.0000	127.0
	6	6.0000	152.4
	7	7.0000	177.8
	8	8.0000	203.2
	9	9.0000	228.6

Bold Face = Standard metric diameters

GLOSSARY

Bearing Surface – Refers to the area of a fastener that contacts the mating surface, such as the area under the head on a bolt or screw. Externally threaded and internally threaded fasteners both typically have a bearing surface.

Blank – A section of metal that has been cut off from a larger section or coil of metal. A blank refers to the section of metal that undergoes a cold forming process.

Blow – Refers to the action of a punch applying a force to a blank as a step in the cold forming process. The number of blows refers to the number of times a force is applied to the blank.

Breakaway Torque – The static torsional resistance of a threaded fastener to disassemble in the absence of any amount of clamp load.

Breakloose Torque – The amount of rotational force required to overcome clamp load and reverse the rotation of a threaded fastener that is in the seated, clamped, or tightened position.

Coating – A finishing process of applying a product to the surface of a part by adhesion, such as paint. It may also refer to the actual product that is applied to a surface rather than the process of applying that product.

Coil / Wire – A long, round section of metal that is looped and bent into a coil shape to be used as the raw material for cold forming processes.

Cold Forming – Any process that involves permanently deforming a section of unheated metal. Cold forming includes cold heading, upsetting, extruding, piercing, trimming, point rolling, and thread rolling among other operations.

Cold Heading – A process that involves applying a force to a section of unheated metal that is beyond that metal's elastic limit causing permanent plastic deformation of the metal. Cold heading typically refers to decreasing the length and increasing the diameter of the metal as is done when the head of a bolt is created.

Conversion Coating – A secondary electro-chemical process that creates an additional layer on top of an electroplated finish. They are typically used to extend the life of metallic plating and thus the overall life of the base part. Conversion coatings may also be used for decorative purposes, conductivity, or as an adhesion promoter or to optimize the torque tension relationship.

Deform Plastically – See "Plastic Deformation."

Delrin® – A brand name for a type of low-friction, high-wear acetal plastic.

Die – A high strength metal device used inside cold forming machines to control the shape of the blank as it is deformed. A punch applies a force to the metal blank and the blank expands until it fills a void within a die.

Elastic Deformation / Limit – The movement of a material occurring due to a force being applied to that material in which the material movement completely reverts if the force is removed. Elastic deformation is not permanent, and the material's original shape is unchanged once that force is removed. This is referred to as the Elastic Limit of the material.

Electroplate – A finishing process of using chemicals and an electrical charge to cause a thin metallic layer to bond to the exterior surface of another metallic part. Zinc electroplate and galvanizing are examples of electroplated finishes.

Embossing – The process of forming raised or recessed areas on the surface of a material.

External Wrenching / Drive – Fasteners that require the installation force to be applied to the outermost surface of the fastener, such as a bolt with a hex head or a nut that is hex-shaped.

Extrude – To permanently deform a material by increasing the length and decreasing the diameter of that material by pushing or pulling it through a die.

Finish – A thin chemical layer that is applied to the surface of a part. Finishes are applied for decorative, protective, performance, or other purposes.

Former – Refers to a machine that permanently deforms an unheated section of metal in any manner.

Grade / Property Class – A system of defining the material properties of a fastener. "Grade" refers to imperial fasteners whereas "class" refers to metric fasteners. A grade or class rating defines what type of material is used, any heat treat requirements, and final strength requirements among other criteria.

Grip Length – The maximum length, or thickness, of material that can be successfully held together without compromising the integrity of the joint.

Hardened – A material that has undergone a heat treat process and now has an increased hardness as a result of the heat treat process.

Header – Refers to the machine that performs the cold heading process. This machine uses a punch to apply a force to the end of an unheated metal blank contained within a die.

Heat Treating – A group of industrial, thermal, and metalworking processes used to alter the physical, and sometimes chemical, properties of a material. The most common application is metallurgical. Fastener heat treating is typically done within an open-flame furnace and includes any necessary quenching and tempering post-processes.

Internal Wrenching / Drive – Fasteners that require the installation force to be applied to an internal recess in the end of a fastener, such as a screw with a Torx®, Phillips®, or slotted recess.

Machining – The process of using a machine to cut away material, such as drilling, milling, and lathe cutting among other processes.

Major Diameter – The largest circumscribing circle created by a threaded product, such as the diameter created by the thread crest of an externally threaded part or the thread root of an internally threaded part.

Minor Diameter – The smallest circumscribing circle created by a threaded product, such as the diameter created by the thread root of an externally threaded part or the thread crest of an internally threaded part.

Non-Standard Part / Special Part – A part that does not comply to a standard. The requirements for such a part are defined by the purchaser or end-user.

Pitch Diameter – A circumscribing circle created by calculated points on a threaded product that is halfway between the minor and major diameter of that threaded product.

Plastic Deformation – The movement of a material occurring due to a force being applied to that material in which the material movement becomes permanent after the force is removed. Plastic deformation is permanent, and the original shape of the material is changed once that force is removed.

Preload – The amount of tension within a fastener as a result of being installed or assembled.

Prevailing Off Torque – The amount of rotational force required to keep a fastener in motion during disassembly in the absence of any amount of clamp load.

Prevailing Torque – The frictional resistance of a threaded joint during rotation in either the installation or removal cycle which does not produce or remove clamp load or bolt stretch. Typically utilized with locknuts or lock patches to resist loosening of the joint.

Proof Load – The maximum amount of stress that can be applied to a material without causing that material to experience plastic deformation.

Property Class – See “Grade / Property Class.”

Salt Spray Test – The process of continuous exposure of ferrous metal to saline solution for the purpose of determining how long the finish will last before the conversion coating, base finish or both break down, and red oxidation/rust first starts to visibly appear on the ferrous material. Salt spray testing is specified as a number of hours to white corrosion, which occurs when the zinc base finish begins to oxidize, and/or to a number of hours to red corrosion, which is when the zinc base finish breaks down and the ferrous base metal oxidizes/rusts. White oxidation will always appear prior to red oxidation. The process requirements for salt spray testing are defined within globally accepted standards, ASTM B117 being one of the most common. Salt spray testing is not just for zinc coatings.

Shank – The unthreaded section of a bolt or cap screw.

Special Part – See “Non-Standard Part.”

Standard Part – A part, or fastener, that is compliant to a standard. Most readily available fasteners are compliant to a standard, however not all parts defined by a standard are readily available.

Standards – Documents produced by an organization that define the requirements of a part and/or process. Standards are used to define dimensional, material, finish, and inspection requirements of fasteners among many other processes.

Tap / Tapping – The process of cutting threads into a material to form internal threads.

Thread Lead – The amount of overlapping material between two mating internally threaded and externally threaded fasteners.

Trim / Trimming – The process of creating a desired shape on a section of material after that material is headed (diameter is increased, length is decreased). Typically, a trimming operation is performed on a section of material that was not contained within a die and has an undefined or “free-flow” shape (i.e. the hex on a bolt head).

Ultimate Tensile Strength – The maximum amount of stress a material can withstand without fracture.

Upset – The process of applying an axial force to a material that causes the material to expand in a cold forming process. It may also refer to the section of material that has been expanded.

Wire – See “Coil.”

Yield Strength – The stress required to cause plastic deformation of a material.

NOTES

Interesting Nostalgia We Would Like to Share:

Park Ohio purchased Russell, Burdsall, & Ward (RB&W) Manufacturing and Distribution in 1995. After this purchase, Park Ohio changed the RB&W Logistics name to Integrated Logistics Solutions (ILS) which was later rebranded as Supply Technologies. RB&W's roots go back to 1845.

In 1876, RB&W's fastener manufacturing skills were featured at the Centennial International Exposition of 1876, the official First World's Fair in Philadelphia. This Fastener Display was in the Machinery Hall – the second largest structure at the Exposition. The exhibits in this hall focused on machines and evolving industries. The United States of America took up two-thirds of the exhibit space. The Machinery Hall had 8,000 operating machines and was filled with a wide assortment of hand tools, machine tools, material-handling equipment, and the latest fastener technology.

A replica of this display can be seen at the Corporate Headquarters of Park Ohio and Supply Technologies in Cleveland, Ohio. The original one remains with the Smithsonian.

Supply Technologies™

The efficiency experts.

Enabling manufacturers to build their products better, smarter, and faster than ever before.

6065 Parkland Blvd., Cleveland, OH 44124

A ParkOhio Company
Cleveland, Ohio USA
NASDAQ Symbol PKOH

P: +1 440.947.2100 **www.supplytechnologies.com** **marketing@supplytechnologies.com**

For an electronic version of the Helpful Hints, sign in on our website here: