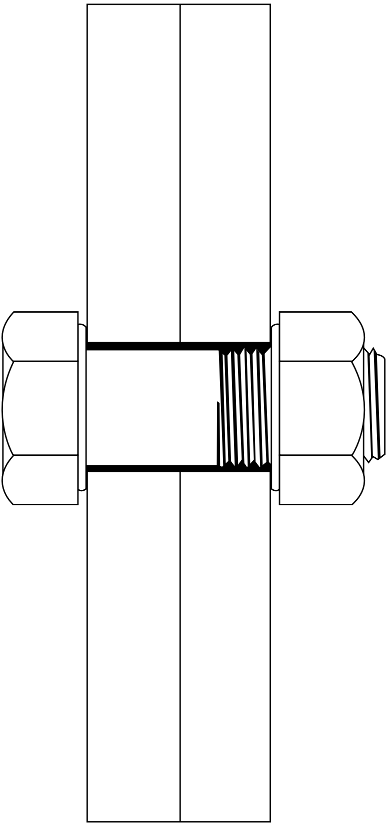
# Helpful Hints



## For Fastener Design and Application

RB&W®
A Park-Ohio Company

# Contents

## Helpful Hints

8th Edition

People talk about "getting down to 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 Eighth Edition of "Helpful Hints" is intended to assist users in determining their needs. Material in this edition has been updated, rewritten, and regrouped for easier access to the subject matter. A new section devoted to metric fasteners outlines recently published standards and terminology.

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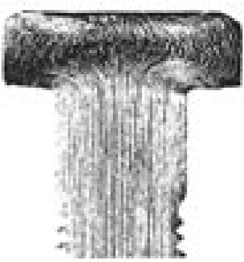
# Cold Heading

## An Introduction

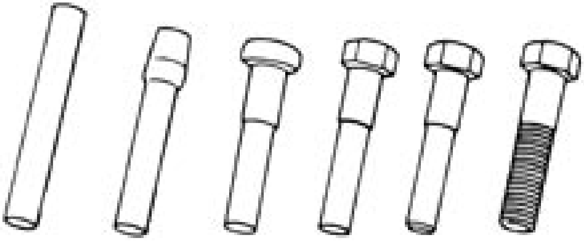
Cold heading and cold forming, the basic methods of manufacturing most bolts and nuts, is the process of forcing unheated metal to flow into dies to change its shape. The machines used are called headers and formers, which are actually high-speed multi-blow presses.



*Metal flow lines of a cold-headed product.*

As illustrated below, 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 screw**



*Cut-off*

*Extrude & bulb*

*Head & extrude*

*Trim*

*Point*

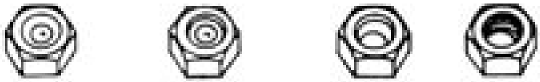
*Thread*

**Stages in Cold Nut Manufacturing**

*Raw material*

*Cut-off*

*Flatten*



*1st upset*

*2nd upset*

*Finish Blank*

*Tap*

# Cold Heading

## 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 production run quantities as follows…

| Diameter/Length | Approx. Pieces |
|---|---|
| • 1/4 thru 3/8 in. x 3 in. long | 100,000 |
| • 7/16 thru 1/2 in. x 5 in. long | 60,000 |
| • 5/8 thru 3/4 in. x 6 in. long | 40,000 |
| • 7/8 thru 1 in. x 6 in. long | 25,000 |

2. Maximum volume of upset is equivalent to length of stock 4 1/2 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. But keep carbon content in steel under 0.45.

4. Concentric pieces are easier to form though eccentric and serrated shapes are practical.

5. Avoid sharp corners. Allow generous radii.

6. Because 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.

# Cold Heading

## Materials

Materials commonly used in cold heading and cold forming commercial grade product 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 screws (made cold) |
| C1030-38 | Grade 5 screws (made cold), A 325 Bolts |
| C4140 | Grade 8 screws, A 490 bolts |

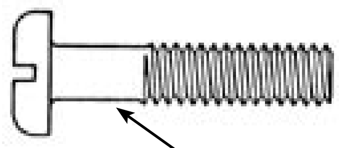
## Rolled Threads

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

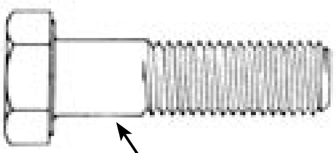
Rolled threads date back to 1851. At that time, rolled threading produced threads with a larger diameter than the shank. As a result, it was necessary to cut threads to have the same thread and shank diameter. Rolled or cut threads were used to identify screws with undersize or full-size body shanks,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

# Cold Heading

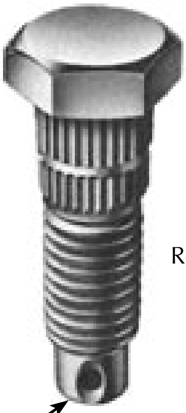
## Custom Cold Formed Parts

Today's cold heading and cold forming equipment offer users a source for volume procurement of close-tolerance complex parts. The reasons are many:

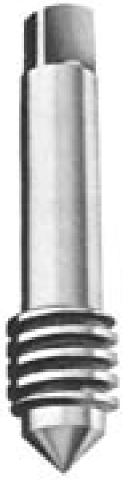
- Design considerations may rule out use of a standard fastener.
- A special part may prove more efficient in assembly.
- A single special may replace two or more separate parts.
- A cold headed or cold formed part may be less expensive than a machined part.

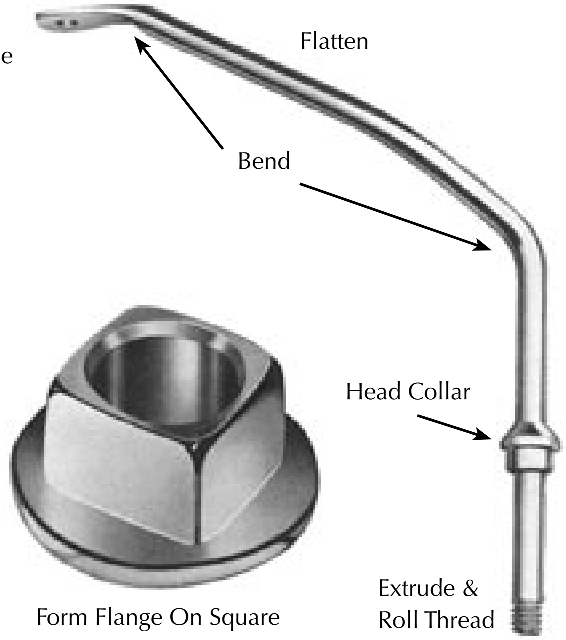
When designing a non-standard part, try to use a listed mechanical performance standard. This will give your supplier easier access to the raw material, or it may be his inventory.

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

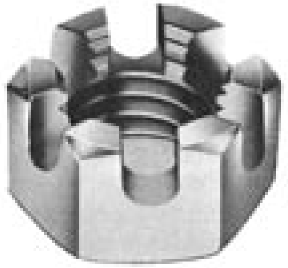


## Cold Heading





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 in the United States include:

**ANSI** **American National Standards Institute, Inc.**
25 West 43rd Street
4th Floor
New York, NY 10036
ANSI standards usually spell out dimensional requirements and refer to other specifications for mechanical performance standards.

**ASTM** **American Society for Testing and Materials**
100 Barr Harbor Drive
West Conshohocken, PA 19428-2959
ASTM standards usually spell out the minimum material and mechanical performance requirements.

**SAE** **Society of Automotive Engineers, Inc.**
400 Commonwealth Drive
Warrendale, PA 15096
Both dimensional and mechanical standards are usually detailed in SAE specifications.

**IFI** **Industrial Fasteners Institute**
1717 East 9th Street
Suite 1105
Cleveland, OH 44114
The IFI frequently issues industry standards that are not included by other standards writing bodies. Acceptable quality levels, engineering recommendations and locking fastener performance standards are included in the IFI series.

While the above groups issue the majority of total industrial fastener standards, the U.S. Government, The Association of American Railroads, The American Petroleum Institute, The Edison Electric Institute and a number of industrial firms also issue fastener standards.
Fastener Standards

## Quality Assurance

RB&W manufactures products to nationally and internationally recognized standards. Quality assurance and inspection procedures to ensure compliance with these specifications are used throughout the manufacturing process.

The past several decades have seen the adoption of the Federal Consumer Protection Act and the Federal Motor Vehicle Safety Standards. These regulations directly or indirectly require documentation of product manufacturing ranging from raw material verification to product application performance testing.

Such levels of quality assurance may be wasteful and costly if they are not required by the application. Accordingly, the user should keep in mind the needs of the application when specifying levels and degrees of quality assurance and inspection.

RB&W is prepared to provide documentation and to assist in determining desirable levels and degrees of quality assurance and inspection documentation.

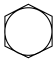
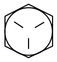
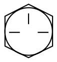
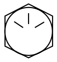
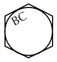
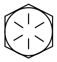
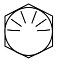
# Fastener Standards

## Product Identification – Fastener Markings

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

Standard fastener head markings have two important meanings: identifying the strength capabilities of the fastener and show 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<br>over 3/4 thru 1-1/2 | 55<br>33 | 74<br>60 |
|  | ASTM A307 – Grade A | | 1/4 thru 4 | – | 60 |
|  | ASTM A307 – Grade B | | 1/4 thru 4 | –<br>– | 60 min<br>100 max |
|  | SAE J429 – Grade 5<br>ASTM A449 – Type 1 | Medium Carbon Steel, Quenched and Tempered | 1/4 thru 1<br>over 1 thru 1-1/2 | 85<br>74 | 120<br>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 | Low Carbon Martensite Steel, Quenched and Tempered | 1/4 thru 1 | 85 | 120 |
| | ASTM A449 – Type 2 | | | | |
|  | ASTM A325 – Type 1 | Medium Carbon Steel, Quenched and Tempered | 1/2 thru 1<br>over 1 to 1-1/2 | 85<br>74 | 120<br>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 2-1/2<br>over 2-1/2 thru 4 | 105<br>95 | 125<br>115 |
|  | 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<br>over 2-1/2 thru 4 | 120<br>105 | 150<br>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<br>170 max |
|  | ASTM A 490 – Type 2 | Low Carbon Martensite Steel, Quenched and Tempered | 1/2 thru 1 | 120 | 150 min<br>170 max |
|  | ASTM A490 – Type 3 | Atmospheric Corrosion Resistant Steel, Quenched and Tempered | 1/2 thru 1-1/2 | 120 | 150 min<br>170 max |

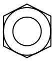
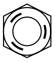
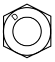
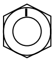
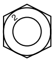
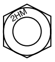
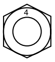
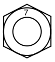
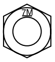
## Product Identification – Fastener Markings

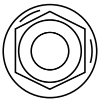
### Externally Threaded Metric Fasteners

| Property Class Designation | Nominal Product Dia, mm | Material and Treatment | Proof Load Stress MPa Length Measurement | Yield Strength MPa | Tensile Strength MPa | Property Class Identification Marking |
|---|---|---|---|---|---|---|
| 4.6 | M5 – M100 | Low or Medium Carbon Steel | 225 | 240 | 400 | 4.6 |
| 4.8 | M1.6 – M16 | Low or Medium Carbon Steel: Partially or Fully Annealed As Required | 310 | 340 | 420 | 4.8 |
| 5.8 | M5 – M24 | Low or Medium Carbon Steel: Cold Worked | 380 | 420 | 520 | 5.8 |
| 8.8 | M16 – M72 | Medium Carbon Steel: Product Quenched and Tempered | 600 | 660 | 830 | 8.8 or 8S |
| 8.8 | M16 – M36 | Low Carbon Martensite : Product Quenched and Tempered | 600 | 660 | 830 | 8.8 |
| 8.8.3 | M16 – M36 | Atmospheric Corrosion Resistant Steel: Product Quenched and Tempered | 600 | 660 | 830 | 8S3 |
| 9.8 | M1.6 – M16 | Medium Carbon Steel: Product Quenched and Tempered | 650 | 720 | 900 | 9.8 |
| 9.8 | M1.6 – M16 | Low Carbon Martensite Steel: Product Quenched and Tempered | 650 | 720 | 900 | 9.8 |
| 10.9 | M5 – M20 | Medium Carbon Steel: Product Quenched and Tempered | 830 | 940 | 1040 | 10.9 |
| 10.9 | M5 – M100 | Medium Carbon Alloy Steel: Product Quenched and Tempered | 830 | 940 | 1040 | 10.9 or 10S |
| 10.9 | M5 – M36 | Low Carbon Martensite Steel: Product Quenched and Tempered | 830 | 940 | 1040 | 10.9 |
| 10.9.3 | M16 – M36 | Atmospheric Corrosion Resistant Steel: Product Quenched and Tempered | 830 | 940 | 1040 | 10S3 |
| 12.9 | M1.6 – M100 | Alloy Steel: Product Quenched and Tempered | 970 | 1100 | 1220 | 12.9 |

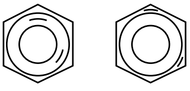
## Product Identification – Fastener Markings

### Internally Threaded Inch Fasteners

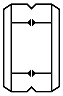
| Grade Identification Marking | Specification | Material | Nominal Size In. | Proof Load Stress ksi |
|---|---|---|---|---|
|  No Mark | ASTM A563 – Grade O | Carbon Steel | 1/4 thru 1-1/2 | 69 |
| | ASTM A563 – Grade A | Carbon Steel | 1/4 thru 1-1/2 | 90 |
| | ASTM A563 – Grade B | Carbon Steel | 1/4 thru 1 | 120 |
| | | | Over 1 thru 1-1/2 | 105 |
|  | 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 |
|  | 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 194 – 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 |



Flange Nut



Formed Hex Nut

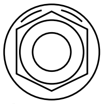


Turned Hex Nut

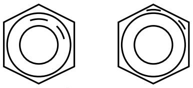
SAE J995 Grade 5
Medium Carbon Steel

Nominal Sizes: 1/4 thru 1" UNC and 8UN
Proof Load Stress: 120,000 psi

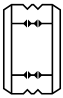
Nominal Size: 1/4 thru 1" UNF, 12UN and Finer
Proof Load Stress: 109,000 psi



Flange Nut



Formed Hex Nut



Turned Hex Nut

SAE J995 Grade 8
Medium Carbon Steel
Quenched and Tempered

Nominal Sizes: 1/4 thru 1-1/2"
Proof Load Stress: 150,000 psi

## Product Identification – Fastener Markings

### Internally Threaded Metric Fasteners

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

## Which Assembly Fails?

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

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

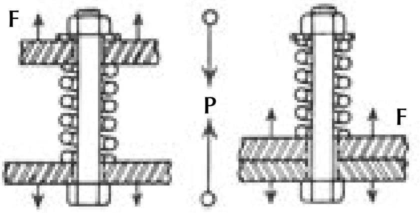


**Figure A** **Figure B**

In Fig. A the total bolt load – i.e., force tending to separate the plates – 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).

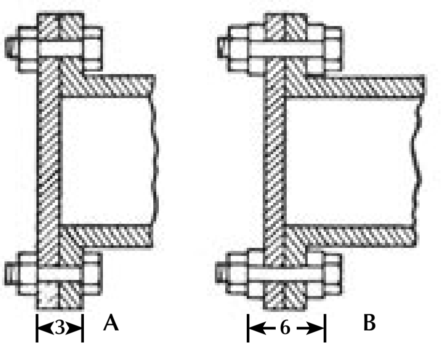
Hence the loading of the bolt in Fig. B is essentially static. This assembly is strong, remains unaffected by the cyclic external stress and won't fail.

Conversely, the bolt loading in Fig. A, being dynamic, fluctuates between P and 2P. 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, 3/8-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.

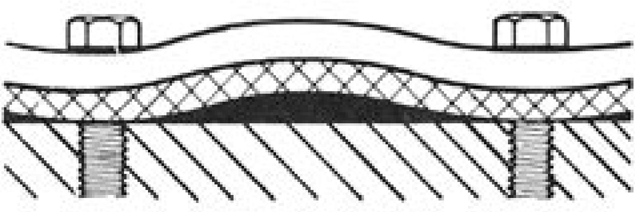
The shorter bolt in Fig. A loses 1,550 pounds of residual tension, retaining only 67% of its initial clamping force. The longer bolt in Fig. B loses 775 pounds of tension, retaining 83% of its initial clamping force.

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 Hex 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 Hex 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. But the hard asbestos gasket used took 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 fasteners.

## Fastening Rigid Joints

Theoretically, a rigid joint is impossible because there's always some elasticity in the fastened metals. But, 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.

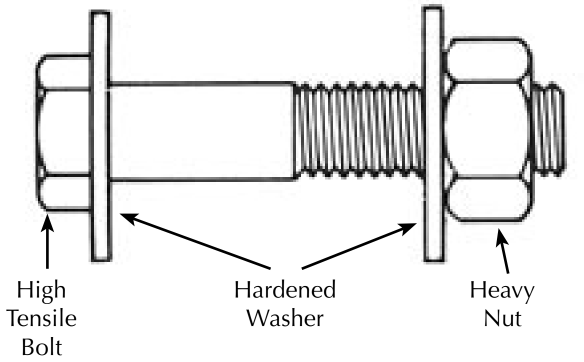
### User Benefits

Rigid joints offer a definite product advantage. They can take high strength hex 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.

Grade 5 Hex Screws offer the additional benefits of lower cost and faster assembly, since you can reduce size of the fasteners, or their number, while actually improving joint strength.

### Installation Hints

There's no problem achieving a rigid joint when connecting heavy section steel members. Just clamp them to the full fastener capacity. Thin sections can be reinforced and similarly fastened. In joining milder steels or softer metals, use of a plate 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, in noncritical fastening applications 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 go into tapped holes with an interference thread fit, hex screws with a simpler clearance fit.

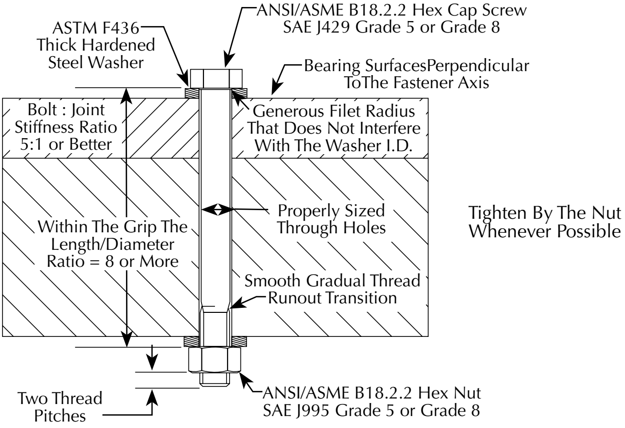
Desirably, studs should all be driven to the same hole depth and bottomed. But, because of normal manufacturing differences, mismatched high and low tolerances will cause some studs to project unevenly. Assemblers must be careful with studs to ensure uniform height projection. This problem does not exist with hex screws that can be tightened to a specified preload. And with hex screws, there's 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 disassemblies? Not at all. Hex screws have been installed 50 times in cast iron test blocks, then tightened to failure without damage to tapped holes.

But, "blind hole" fastening of large flanges, pressure plates, heads, etc., requires greater "tapped hole" precision, when using 1/4-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 hex screw (SAE Grade 5) and standard hex screw (SAE Grade 2) are identical – the same machine makes both. They only differ in cost and the clamping force that can be developed.

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

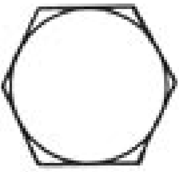
Heat treatment and carbon content represent only a fraction of the total cost of material and manufacturing that go into a fastener. Yet the heat-treated fastener, depending on size, may develop two times the strength of its standard (SAE Grade 2) counterpart. A given load requirement can be met with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

### Grade 5 Most Versatile

There are fastener grades stronger than the high-strength ex (SAE Grade 5). These are heat-treated alloy-steel fasteners. But, they cost more, and unless their super-strength is needed and utilized in 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.

In all cases, nuts used with any particular fastener grade must have a proof load equal to the ultimate tensile of the screw.



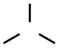
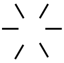
Grade 2

Grade 5

# Bolt/Screw Selection

## 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 PC5.8 Material is low or medium carbon steel.
Grade 5 or PC8.8 Material is a medium carbon steel, quenched and tempered.
Grade 8 or PC10.9 Material is medium carbon alloy steel, quenched and tempered.

Special circumstances do allow material and processing modifications.

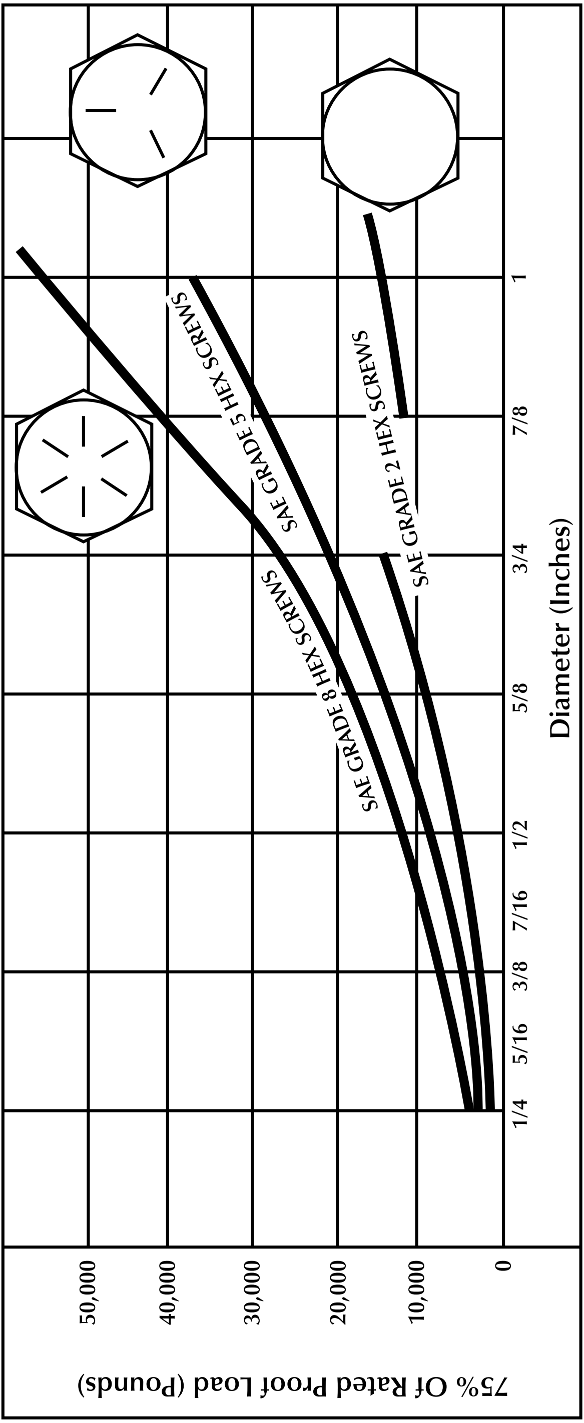
The graph on Page 19 shows the relative clamping strength of the three S.A.E. grades.

### Some Suggestions

In terms of holding power, stronger fasteners may be less costly than the least expensive Grade 2. For example, a 5/8-in. diameter Grade 5 fastener has a proof load of 19,200-lb., and a 3/4-in. diameter Grade 2 fastener has a proof load of 18,400-lb. The 5/8-in. diameter Grade 5 fastener, being smaller than the 3/4-in. Grade 2, costs less.

By using higher strength fasteners, users can effect 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.


75% Of Rated Proof Load (Pounds)
0
10,000
20,000
30,000
40,000
50,000
Diameter (Inches)
1/4
5/16
3/8
7/16
1/2
5/8
3/4
7/8
1
SAE GRADE 8 HEX SCREWS
SAE GRADE 5 HEX SCREWS
SAE GRADE 2 HEX SCREWS

## 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 application. But, when used in smaller sizes and tapped holes, the Hex Screw has a number of design and production advantages.

### Assembly Considerations

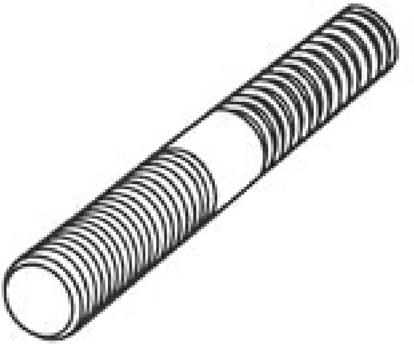
In fastener selection, production and assembly considerations are as important as joint strength.

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.



# Bolt/Screw Selection

## Hex vs. Socket Head Screws

If head diameter is limited, consider using an internally wrenched fastener. But, for more holding power per dollar, use the Hex Screw.

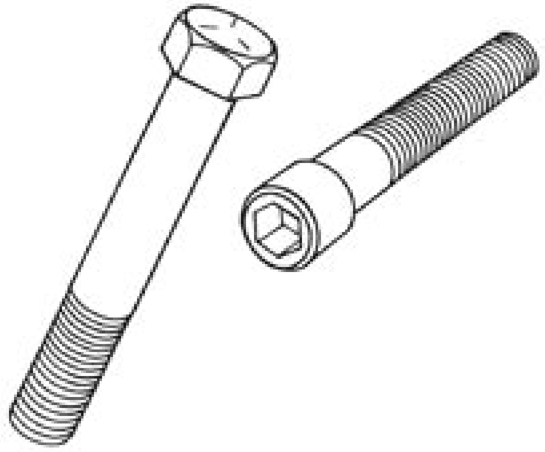
There's no question that small head socket screws are best utilized in counter bored holes to clear tight spaces. But, it's not possible to utilize the full strength of the alloy used for the socket head. Remember that the strength of a connection depends on fastener preload – not on the strength of the fastener material alone.

### Socket Head Screws

Internal wrenching rarely develops the torque needed for proper preloading. And, 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 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 Selection

## Nut Design

The standard nut isn't as simple as it looks. Considerable design know-how goes into nut manufacture 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 that 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. Nuts manufactured for ANSI's "Heavy" series are higher and 1/8-in. wider across flats for added bearing area and wall thickness. They are generally used with structural bolts or when bolt through holes are oversized.

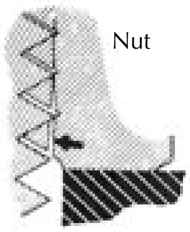


Figure A

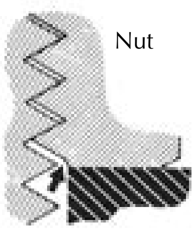


Figure B

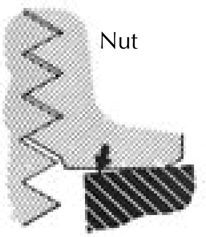


Figure C

# Nut Selection

## 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.

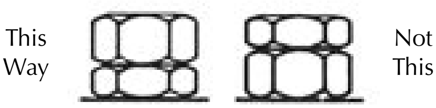
HEAVY HEX nuts are wider than “Hex” nuts in all sizes by 1/8-in. across flats. Thus, their value diminishes as size increases. Most effective in 1/2 – to 1 1/2-in. size ranges, Heavy Hex nuts satisfy applications involving excessive clearance holes, unusual loads and certain boiler codes.

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.

NUT MATERIALS. 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.

“SPECIAL” FUNCTION NUTS. Jam Nuts are used for position locking. The use of two nuts together forms a superior and economical locking device. When used to lock a standard nut, the Jam Nut should be used between the Hex Nut and the joint surface so that 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’re furnished only with fine threads and hardened. But the better choice for these applications is coarse thread Heavy Nuts.

LOCK NUTS are used where dynamic environmental loading (vibration) is present.

# Nut Selection

## Nut Strength

It is difficult to plastically adjust and distribute load over many threads with hard heat-treated nuts. That's 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 also cause progressive shear.

That's 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.

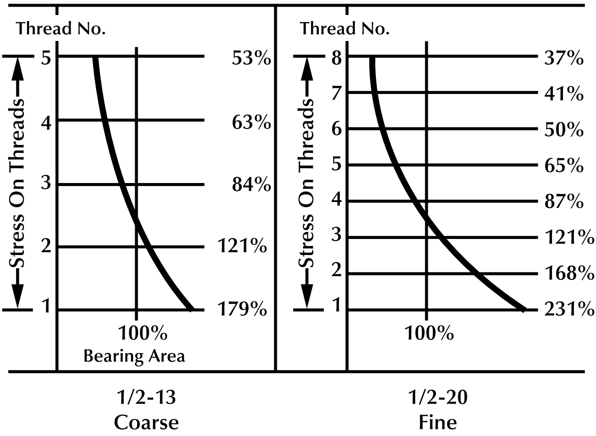
Thus if a nut hasn't failed in wrenching, it can still withstand at least 20% more direct pull than it sustained during tightening.

## 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 specs., 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.

But material strength isn't the only factor governing whether the bolt or nut fails first. Threads-per-inch complicate the picture. Stress variation differs for coarse and fine threads as shown in this graph.



The nut with higher stress on the first and shallower thread has a higher risk of failure through progressive shear.

Now let's apply this to a possibility on the production line. 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 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.

# Nut Selection

## 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. Here 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

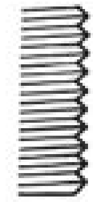
## How Threads Affect Nut Strength

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

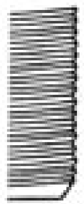
Theoretically, the stress areas of fine thread bolts should be stronger than coarse thread bolts. But they're not. Failure by thread stripping on a fine thread is a greater possibility for two reasons:

1. On a 1/2-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.
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 1/2-in. diameter bolt. For a fine thread of equal diameter, the stress area would be 0.1599 sq. in., and, theoretically, a 1/2-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's not stress area, but the volume of thread metal remaining in actual engagement under load, and the ability to distribute that load over all threads, that determines stripping strength.

It is for this reason that SAE specifications have, for 1/4-in. through 1-in. diameter Hex Nuts, reduced the proof load of Grade 5 fine thread nuts to 109,000 psi, while their coarse thread counterparts remain at 120,000 psi.

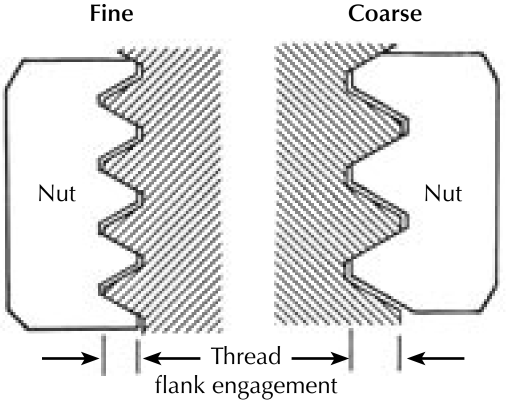
## Engaged Thread Metal Governs Strength

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 metal — with an apex and a root. Obviously, there is less metal 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.



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 metal that enables threads to "give" and thereby engage more thread metal 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 “babying” in handling since they’re 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.

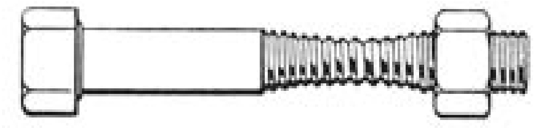
## Stripping Strength vs. Tensile Strength

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. But a study of thread geometry and behavior under load brings a practical explanation.

### 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’s 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. Obviously, a Delrin nut cannot be torqued on a steel bolt to any degree of tightness without failing. But this example does show 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, ask for RB&W's "A Quick Look at Locking Fasteners."

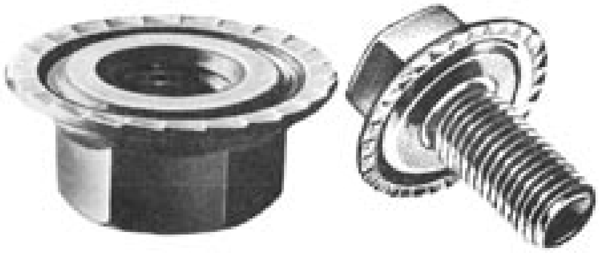
### Locking Power

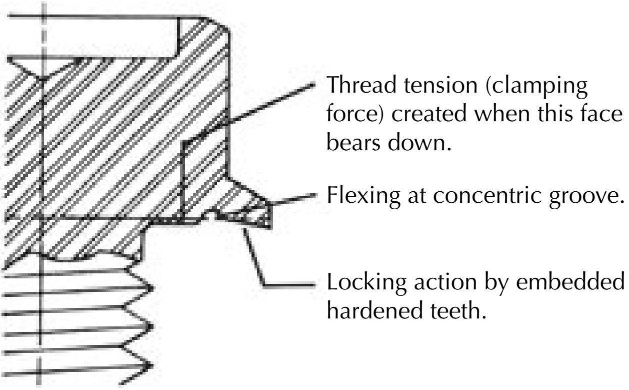
The locking action of self-locking fasteners depends on: an increase in thread friction; an integrated anchoring device; or both.

There are three basic types of locking fasteners: 1) free spinning; 2) prevailing torque reusable type; and 3) prevailing torque chemical reaction type.

**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.

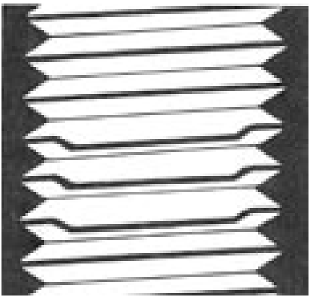


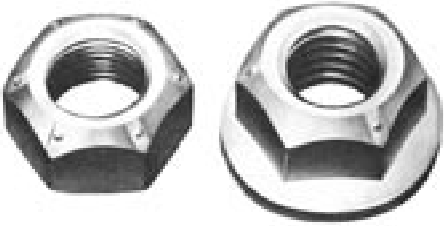


## Locking Fasteners

**Reusable Prevailing Torque (PT)** fasteners develop resistance to back-off as soon as the friction device reaches the mating thread. The most prevalent prevailing torque fasteners are all-metal deflected thread, plastic insert and plastic patch. All rely on the spring action of the locking device to develop prevailing torque. The spring-back action of the locking element is the key to fastener reuse.

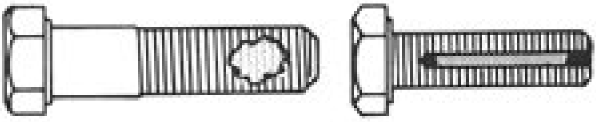
Prevailing torque screw deflected external threads create a friction lock with spring-back for reuse.





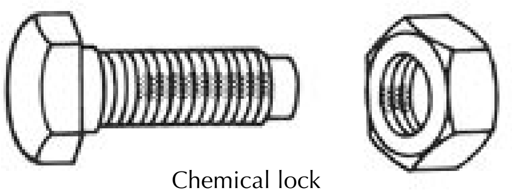
The Hex prevailing torque nut (left) with six marks signifying Grade C; and Hex Flange prevailing torque nut (right) with three marks indicating Grade F (Grade B if a Hex Nut with no flange.)

Plastic Patch and Plastic Insert Lock Screws



# Locking Fasteners

**Chemical Reaction** locking fasteners use epoxys, 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. Break-loose (off) torque levels are similar to the reusable types, but they offer little or no prevailing torque once dissembled. Chemical reaction locking fasteners offer high break-loose (off) torques and minimal prevailing-on torque, but once dissembled, their locking device has limited reuse.



Chemical lock

# 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 performance. Performance drop-off signifies the degree of prevailing torque element wear and spring-back potential.

Specifications such as 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 nut are not heat treated. Grade B prevailing torque nut may or may not be heat treated (manufacturers' 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 that their stress levels are best suited to the mating screw grade. Thread harness, 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 screw. Conversely, using a Grade B or C prevailing torque nut on a Grade 2 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 screw torques. In the second example, the Grade B or Grade C prevailing torque nut would overstress the Grade 2 screw 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.

# Locking Fasteners

## 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

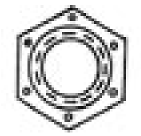
Strength grades and the producer's identification appear on the heads of domestically made prevailing torque screws. See pages 10 and 11 for strength grade markings.

### 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.

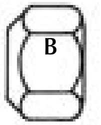
| Nut | Screw |
|---|---|
| Grade A | Grade 2 |
| Grade B | Grade 5 |
| Grade C | Grade 8 |

The three common methods of identifying locknuts are as follows:



**Grade Identification**
Grade A no marks
Grade B three marks
Grade C six marks

Marks need not be located at corners



**Grade Identification**
Grade A no marks
Grade B letter B
Grade C letter C

60˚ -90˚ Included Angle

**Grade Identification**
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.

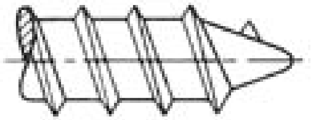
## Introduction

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 in three categories:
- Thread Forming
- Thread Cutting
- Thread Rolling (or Swaging)

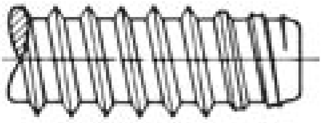
### Thread Forming Screws

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



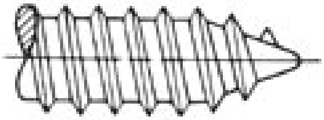
(Type A)

TYPE A – a spaced thread screw with a gimlet point. Primarily used in light sheet metal, resin-impregnated plywoods or asbestos 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.



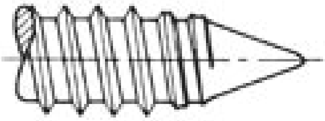
(Type B)

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 asbestos 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.



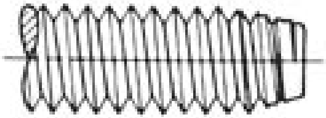
(Type AB)

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, asbestos 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)

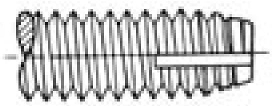
TYPE C – has machine screw threads with a blunt point and tapered entering threads. Primarily used to tap into thicker metallic sections than the Type AB thread series and where chips from thread cutting screws are objectionable. Extreme driving torques may be required where long thread engagement is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

# Tapping Screws

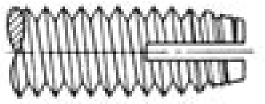
## Thread Cutting Screws

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.

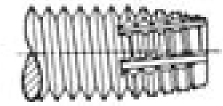
TYPES D, F, G and T – have machine screw threads and diameter pitch combinations, blunt point and tapered entering thread. The entering threads have one or more cutting edges and chip cavities. Primarily used for aluminum, zinc, or lead die castings, steel sheets and shapes, cast iron, brass and plastics.



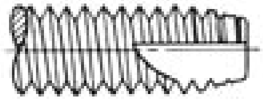
“D”



“G”

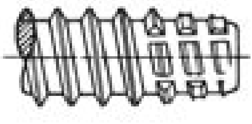


“F”

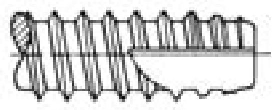


“T”

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, asbestos and similar compositions.



“BF”



“BT”

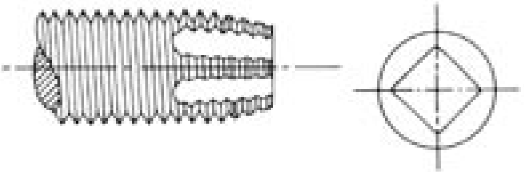
### Thread Rolling Screws

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.

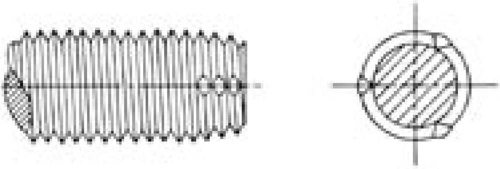
# Tapping Screws

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

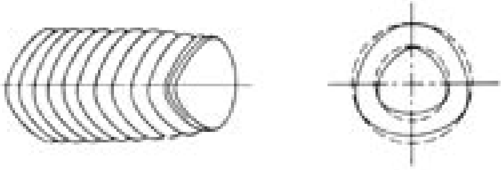
1. Easier starting due to their unique points and body configuration.
2. Less driving torque required.
3. The cold working of material during thread forming improves joint strength.



"SF"



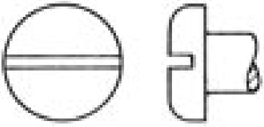
"SW"



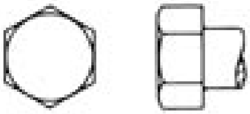
"TT"

**Head Styles and Drives**

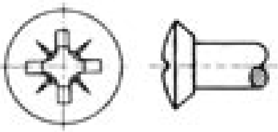
Pan head tapping screws are recommended for most applications. Severe torque conditions may dictate use of a recess drive such as CROSS or Six-Lobe or a hex head.



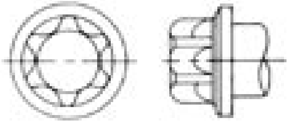
Slotted Pan Head



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 1/2–in. diameter bolts. He specified 5/8-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 1/2–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 1 1/4–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 3/4-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 proofload and provide a high preload – 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 $1^{1}/_{8}$ to $1^{1}/_{2}$ 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:

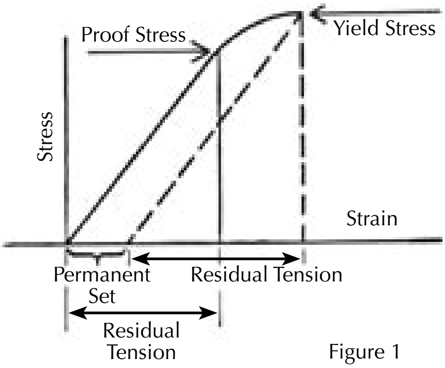


Figure 1

Up to proof stress, the strain in a bolt is proportional to stress. Beyond the 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.

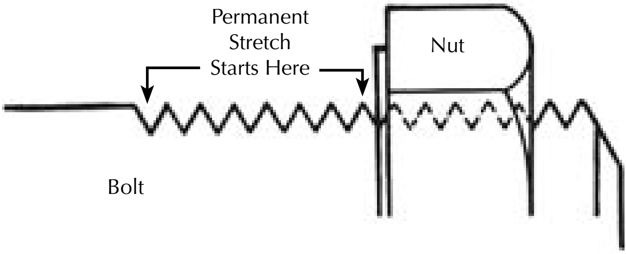


Figure 2

Permanent set starts at the section with 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 four times 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. But to obtain a 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 man with the wrench. A bolt is no better than the supervision of its tightening.

### Flexible Joints Different

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

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. But 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.

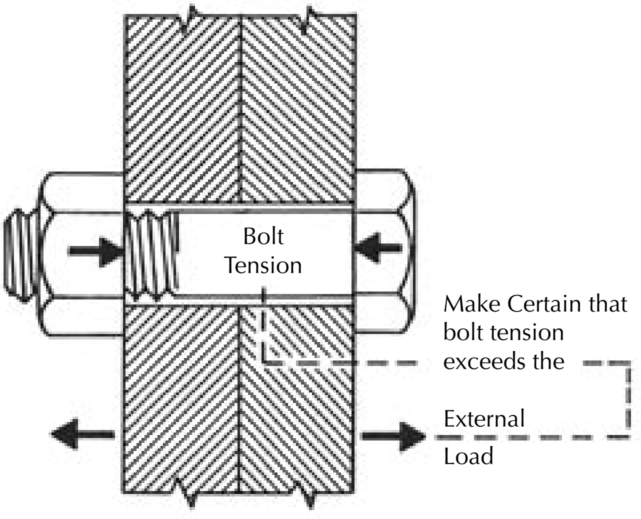
### Demonstration

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's no slippage, no loosening. The full net section of the structural member becomes available to support the shear load.

Tightening high-strength bolts to their proper load level will result in a stronger joint.

# 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. But 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?

Obviously, 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's the right torque for bolts?" is a tough question. There are too many variable conditions. But 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.

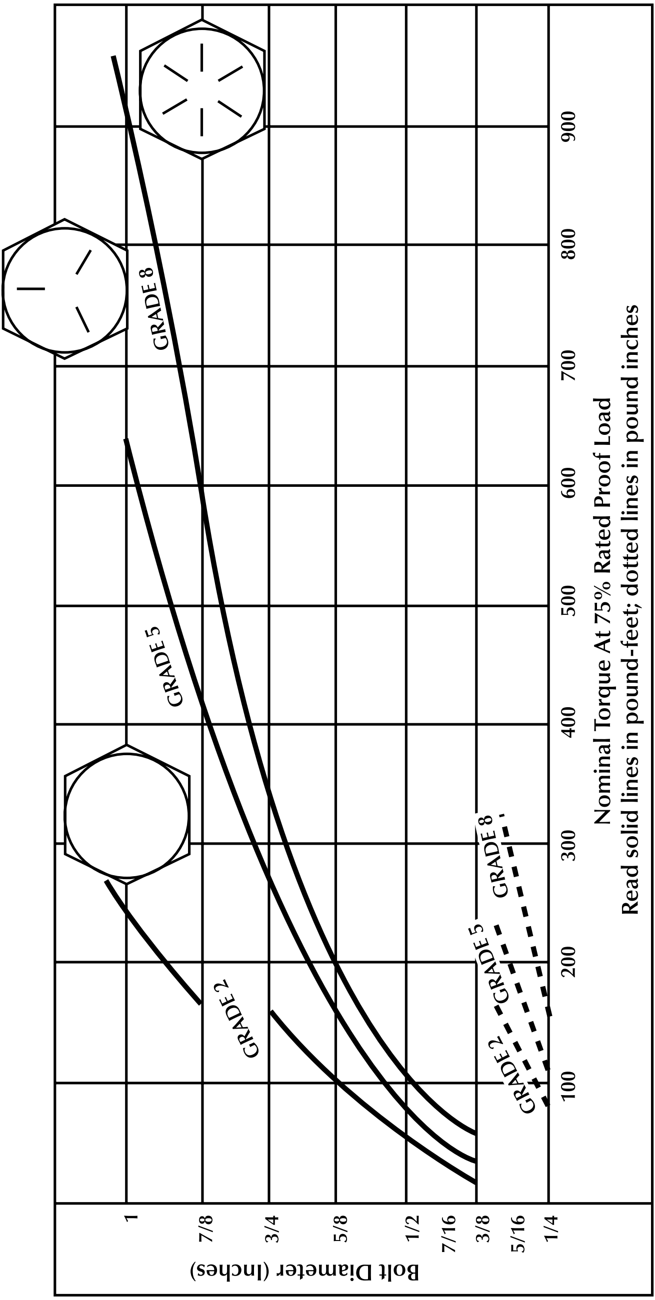
**Lb.-In. Torque =**
**0.2 x bolt diameter x bolt tension**

Tests show that the 0.2 torque coefficient is approximately constant for normal friction conditions, all fastener diameters, and for coarse and fine threads. The average deviation is about 7%. But when are conditions "normal"? The only sure way to check torque is to set up a pilot assembly and try it out.

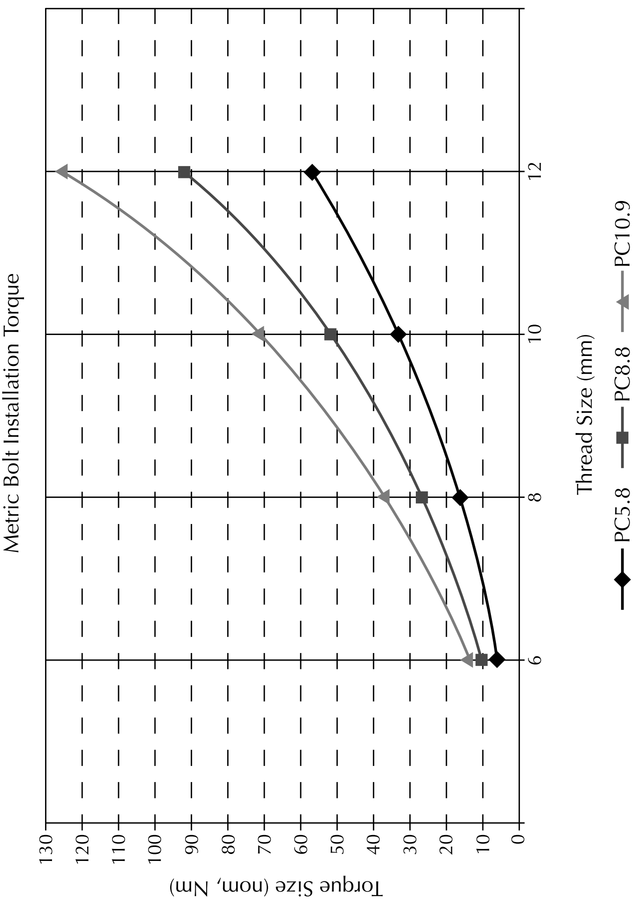
In pilot testing for rigid joints, first tighten a few bolts with the torque wrench to failure. Then set the torque at 60% of that load; or even at yield strength, since the torsion component vanishes when tightening ceases, leaving the bolt under tension only (which is well below ultimate strength).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

The graphs on pages 51, 52 and 53 suggest torque for various size bolt. Remember, these curves are only starting points in determining proper torque-tension relationships. (Note: The break in Grade 2 torque between the $^3/_4$ in. and $^7/_8$ in. diameters reflects the drop in the proof load specification from 55,000 psi to 33,000 psi.)

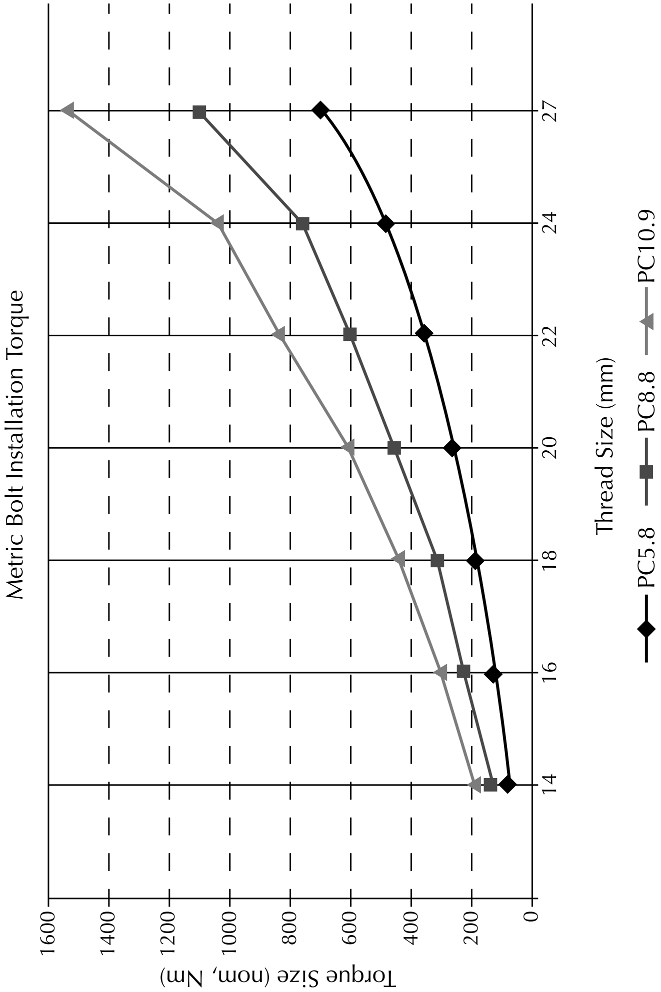
# Bolt Installation – Torque



# Bolt Installation – Torque



# Bolt Installation – Torque



## 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.

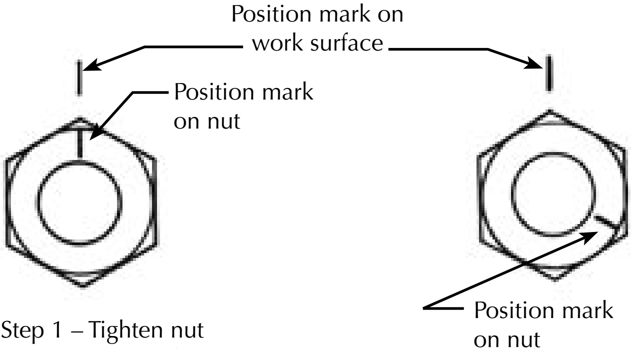
As the nut advances, the bolt is stretched and loaded in tension. Since thread pitch is fixed, an increment of nut turn from the snug tight position produces a fixed increment in tension. Coarse thread nuts obviously need less turning than fine thread nuts since they have fewer threads per inch.

In steel structures, nuts are coarse thread exclusively. They are wrenched from 1/3 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 1/3 turn (for bolts up to 4 diameters long – bearing faces normal to bolt axes). Use 1/2 turn for bolts 4 to 8 diameters, and 2/3 turn for bolts 8 to 12 diameters.

# Corrosion Resistance

## An Introduction

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 cadmium, copper, nickel or other platings.

### Organic Coatings

Corrosion resistance also can 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 also 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.

Experience has developed a suitability "scale" of various coatings for fastener protection:

**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.

**Cadmium plate** stands out where salt atmospheres predominates. As cadmium will react with some foods, its use in many appliances is therefore ruled out.

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

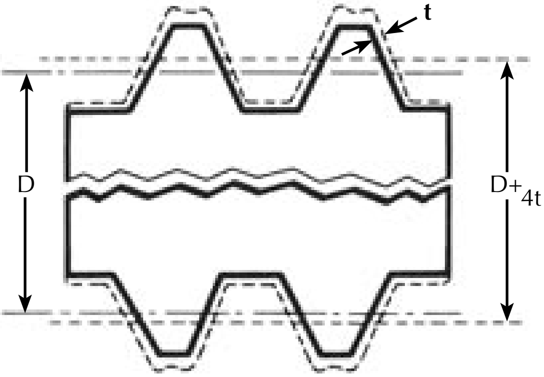
Various **conversion coatings** are available which tend to extend the life of zinc, cadmium and phosphate finishes.

**Chromium**, plated over copper, should be considered more for appearance on fasteners rather than protection.

## Effect Of Plating On Thread Fits

Because of thread geometry, one unit of plating thickness on screw thread flanks will add four times 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 bath 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

Plated threads should go into a Class 2 basic GO gage to assure ease of assembly with mating parts. However, if the plated or unplated screw will enter a Class 2A LO gage, it should be rejected. Its entry indicates that 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. Obviously, all stainless steels are not alike. Some will corrode where others won't.

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 hardenable, 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're the most economical in the stainless steel group, but are generally used where corrosion-resistance requirements are not severe.

### 300 Series

Formerly called 18-8 stainless (18% chromium-8% nickel), the 300 Series of alloys offers the highest degree of corrosion resistance. This series is also the most costly commercial fastener due to the expensive raw material and the difficulty of cold working it. 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."

# Metric Fasteners

## An Introduction

International accord has been reached on a complete system of metric fastener standards. The International Organization for Standardization (ISO) is reviewing these recommendations as the basis for developing new ISO standards. The ultimate benefit will be an international mass interchangeability of fastener products.

A complete set of metric fastener standards for use in North America has been published by the Industrial Fasteners Institute (IFI). The IFI believes their "Metric Fastener Standards" will be substantially accepted as the final international standards.

The IFI metric fastener system is based on four "building blocks":
- Language
- Sizes
- Screw threads
- Materials

# Metric Fasteners

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

**Fastener Quantities And Their Metric Units Of Measurement**

| Quantity | Unit Of Measurement | Symbol |
|---|---|---|
| Area | square millimetre | $mm^2$ |
| Density | kilograms per cubic metre | $kg/m^3$ |
| Force | kilonewton (1) | kN |
| Length | millimetre | mm |
| Mass (weight) | kilogram (2) | kg |
| Plane angle | degree | ° |
| Plating thickness | micrometre | µm |
| Stress (pressure) | megapascal (3) (4) | MPa |
| Surface finish | micrometre | µm |
| Temperature | degree Celsius | °C |
| Time | second | s |
| Torque | newton-metre | N•m |
| Volume (6) | cubic millimetre | $mm^3$ |

1. A newton (N) is the unit of force required to accelerate a mass of one kilogram (kg) one metre per second per second. kN = (kg) $(m/s^2)/1000$.
2. Grams (g) and megragrams (Mg) are also acceptable.
3. Newton per square millimeter ($N/mm^2$) is numerically equal to MPa and consequently, may be seen on customer orders or inquires.
4. The Pascal unit of stress is a Newton per metre squared, $N{\cdot}m^2$.
5. Minute (min.), hour (h), day, week, month and year are acceptable.
6. Litre for the unit of fluid volume is acceptable. Litre is abbreviated ℓ.

# Metric Fasteners

Sizes … 25 diameter-pitch combinations are recommended in IFI's "Metric Fastener Standards" book in the 1.6 mm through 100 mm diameter range. This compares with 57 diameter-pitch combinations for inch system fasteners in the same diameter range. The following table compares the Unified-Inch and IFI Metric Diameter-pitch combinations:

### Unified-Inch And IFI Metric Compared

| Unified inch | | | | IFI-500 Screw Thread Standard |
|---|---|---|---|---|
| Coarse | | Fine | | |
| Nom. Size | Threads/ (in.) | Nom. Size | Threads/ (in.) | Major Dia. (mm) Thread 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 | M |
| 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 Fasteners

**Screw Threads** … A single thread series is offered in IFI's "Metric Fastener Standards" with pitches that fall between the Coarse and Fine Thread series of the inch system. Two classes of fits are offered on external threads – General Purpose, which equates to the inch Class 2A; and Close Tolerance, which equates to the inch Class 3A. Only one General Purpose fit is offered for internal threads.

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

Below are shown seven metric material property classes for bolt products and four for nuts.

### IFI METRIC PROPERTY CLASSES

**Bolts, Screws and Studs**

| Property class | SAE grade(1) | Nominal diam (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 diam (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 | 855 | 8.8, 5.8, 4.8 | B92, | C31 |
| | | M8 thru M10 | 870 | | | |
| | | M11 thru M16 | 880 | | | |
| | | M17 thru M39 | 920 | | B98, | C37 |
| 9 | | M1.6 thru M14 | 900 | 9.8, 8.8, 5.8, 4.8 | B89, | C30 |
| | | M5 thru M24 | 990 | | | |
| | | M16 thru M36 | 910 | | | |
| 10 | 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 |

*Indicates exact conversion

# 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 |
| | | .5512 | **14** |
| | 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

## Helpful Hints

You can realize efficient assemblies and control costs with correctly designed fasteners. Customers are encouraged to review their fastener applications with RB&W engineers.

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