

## **Fasteners**

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Copyright notice:

- Much of the following material is taken from Carroll Smith's *Nuts, Bolts, Fasteners and Plumbing Handbook*. This book contains a wealth of knowledge concerning the proper selection and use of hardware used to join components. It is available for ~ \$20 and will save you thousands of dollars in mistakes your first few years working industry. ***Translation: you would be foolish to perform mechanical design without reading a copy of this book from cover to cover!*** These notes attempt to establish a solid foundation and provide [references for further study](#).
- Additional material and tables come from K. H. Moltrecht's *Machine Shop Practice* and Joseph E. Shigley's *Mechanical Engineering Design* textbooks.

Why you should care about these notes:

- Fastener failures (due to improper design/selection or installation) are **THE NUMBER ONE** cause of mechanical failures in industry. The topic is rarely given the focus it deserves in the classroom for a variety of reasons, none of which are acceptable. Consequently, this will likely be the most thorough discussion of fasteners you will receive.
- Virtually **every** mechanical assembly designed in industry makes use of fasteners to attach components together to form systems.

## 1. Definitions

Fasteners are defined as hardware that can be easily installed and removed with hand or power tools. Common fasteners include screws, bolts, nuts and rivets. The terms *bolts* and *screws* do not refer to specific types of fasteners, but rather how they are used (i.e. the application). Thus the same fastener may be termed a bolt or a screw. Bolts are defined as headed fasteners having external threads that meet an exacting, uniform thread specification such that they can accept a non-tapered nut. Screws are defined as headed, externally-threaded fasteners that do not mate with a non-tapered nut and are instead threaded into the material they will hold. As shown in figure 1, a *bolt joint* can be defined as that which uses a bolt and nut assembly (inherently requiring two tools to tighten or loosen) whereas a *screw joint* can be defined as one in which a screw is mated into a matching female thread in a workpiece (therefore only requiring one tool to tighten or loosen). As seen in figure 1, studs are a hybrid between a bolt and a screw, since one end of the stud functions as a screw while the other functions as a bolt.

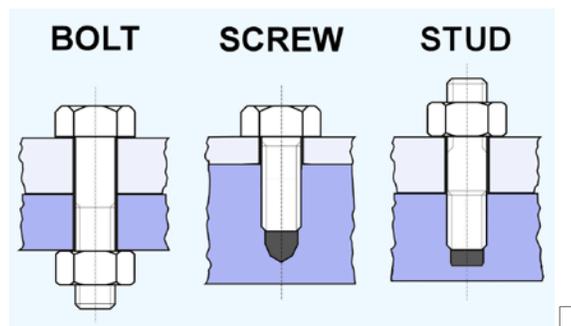


Figure 1. Bolt, screw and stud applications.

## 2. Common Fastener Types

Figure 2 illustrates the variety of male fasteners used in industry; the most common types are hex head, slotted head, flat (or countersunk) head, round head, socket (or “allen”) head, button head and socket set screw. Figure 3 shows different female fasteners (i.e. *nuts*) used in industry; the most common types are regular hexagonal nuts and nylon ring elastic stop nuts (also known as “lock nuts”).

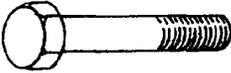
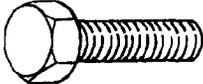
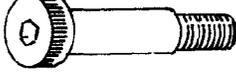
Types of bolts and machine screws		
	American usage	British usage
	Hex head bolt	Bolt
	Set screw Hex head cap screw	Setscrew
	Flat head screw	Countersunk head screw
	Oval head screw	Raised countersunk head screw
	Round head screw	Round head screw
	Pan head screw	Pan head screw
	Cheese head screw	Cheese head screw
	Slotted set screw Grub screw	Grub screw
	Allen bolt Socket head bolt Internal wrenching bolt	Cap head
	Button head screw	Button head screw
	Socket head shoulder screw	Shoulder screw
	Socket set screw	Socket set screw

Figure 2. Male fasteners common in industry.

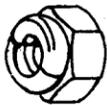
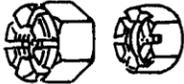
<b>Types of nuts and washers</b>		
	<b>American usage</b>	<b>British usage</b>
	Hexagon nut	Full nut
	Low hex nut Shear nut Jam nut	Lock nut
	Nylon ring elastic stop nut	Self-locking nut with Nylon insert
		Aerotight stiff nut
	Castellated nut Slotted hex nut	Castle nut Slotted nut
	Wing nut	Wing nut
	Acorn nut Dome nut	Dome nut
	Flat washer (plain and chamfered)	Stamped washer (plain and chamfered)
	Lock washer (split)	Spring washer (single coil)
	Lock washer (double split)	Spring washer (double coil)
	Star washer (external teeth)	Shakeproof washer (external teeth)
	Star washer (internal teeth)	Shakeproof washer (internal teeth)

Figure 3. Female fasteners common in industry.

### 3. Fastener Nomenclature

Design engineers are frequently tasked with selecting and specifying fasteners used in their designs. Consequently, understanding basic fastener nomenclature is important. Figure 4 illustrates the different parts of a standard threaded fastener.

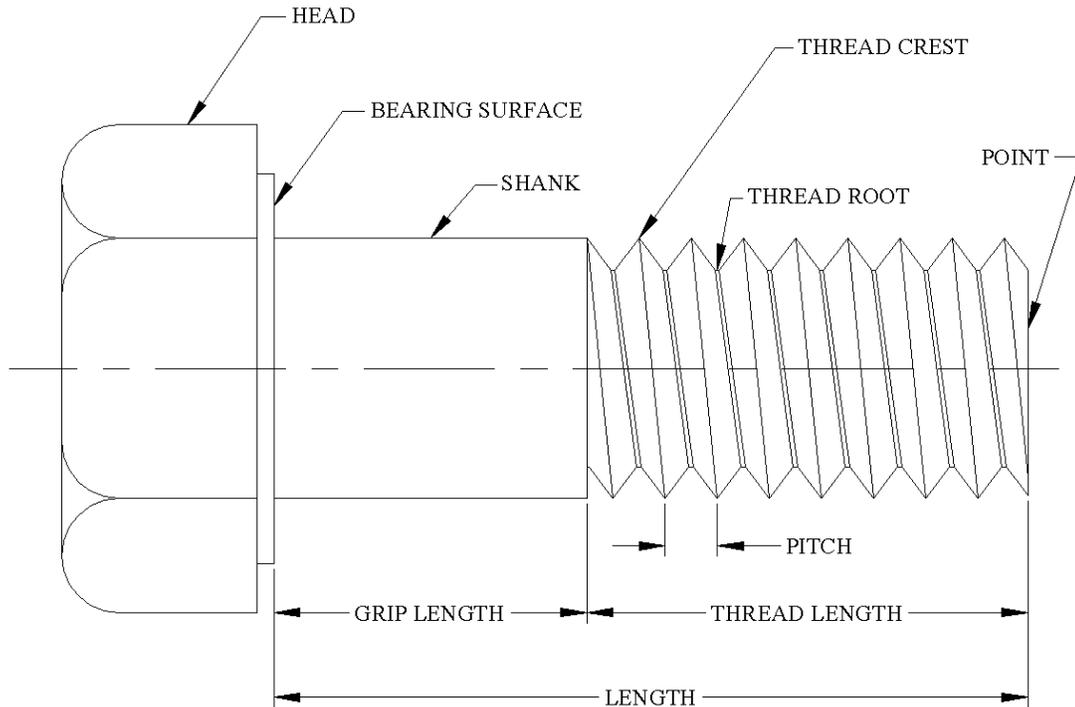


Figure 4. Important male fastener nomenclature.

- A. *major diameter* – the largest diameter of a fastener thread
- B. *minor diameter* – the smallest diameter of a fastener thread
- C. *pitch* – the linear distance from a point on the thread to a corresponding point on the next thread – measured parallel to the axis of the thread
- D. *lead* – the linear distance that a point on a fastener thread will advance axially in one revolution (equal to the pitch of the fastener)
- E. *thread root* – the surface of the thread that joins the flanks of adjacent threads and is immediately adjacent to the cylinder from which the thread projects; in other words, the valley of the thread.
- F. *thread crest* – the surface of the thread that joins the flanks of the thread and is farthest from the cylinder from which the thread projects; in other words, the peak of the thread.
- G. *head* – the enlarged shape that is formed on one end of the fastener to provide a bearing surface and a method of turning (or holding) the fastener
- H. *bearing surface* – the supporting surface of a fastener with respect to the part it fastens

- I. *point* – the extreme end of the threaded portion of a fastener
- J. *shank* – the cylindrical part of a fastener that extends from the underside of the head to the starting thread
- K. *length* – the axial distance between the bearing surface of the head and the extreme point
- L. *grip length* – the length of the unthreaded portion of the fastener (i.e. shank) measured axially from the underside of the bearing surface to the starting thread
- M. *thread length* – the length of the threaded portion of the fastener; NOTE: with all commercial and aerospace fasteners, threaded length is a function of fastener diameter

#### 4. Fastener Thread Types

In the most general sense, there are two classes of fastener threads: English and metric. For each class, regardless of country of origin, there are two types of threads: fine thread and coarse thread. The [drill and tap chart](#) summarizes this information in one convenient location and will be referenced later in these notes.

One of the most common fastener mistakes is using the wrong type of thread in the wrong type of material. The basic rule for fastener selection is: **fine threads are stronger when the female thread is strong relative to the male thread, and coarse threads are stronger when the female thread is weak relative to the male thread.** The reason for this statement is that a smaller minor diameter increases the thread area, resulting in higher static strength and fatigue resistance in female threads. Conversely, a larger minor diameter increases the stress area, resulting in a higher static strength and fatigue resistance in male threads. It is instructive to select a fastener size off the tap chart and prove this statement; when performing the analysis, assume stresses are distributed over only the first five engaged threads.

Figure 5 depicts a 1/4-20 fastener. Due to the elasticity of the fastener, only the first five threads are engaged during loading regardless of the thread type (coarse / fine). Female threads typically fail due to shear along the major diameter and male threads typically fail due to tensile loading along the thread root.

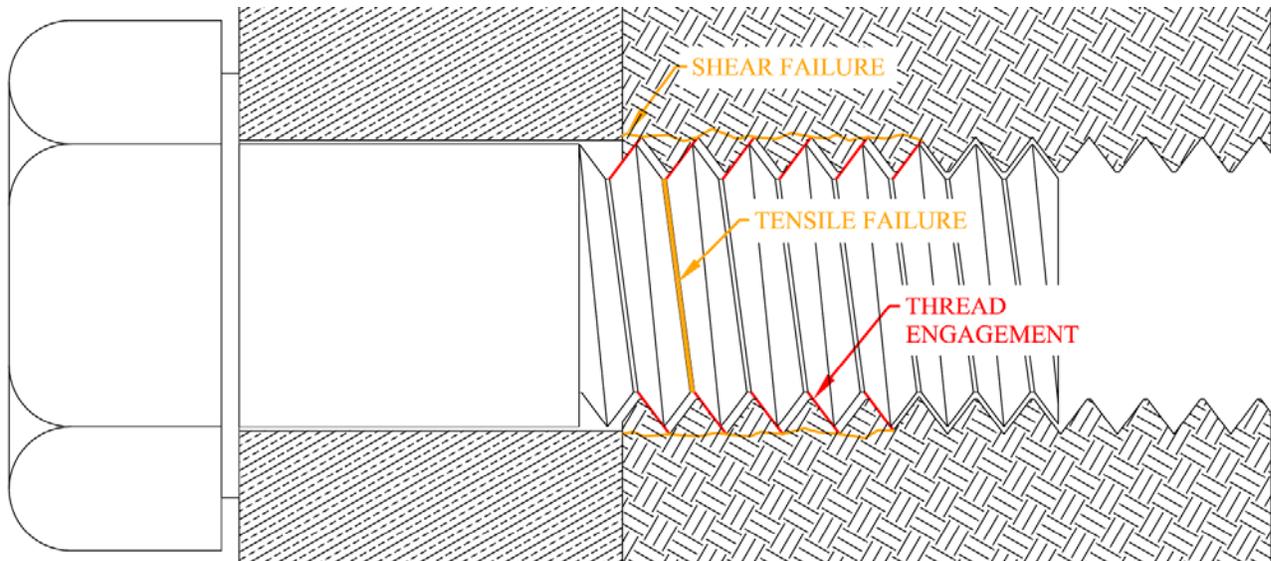


Figure 5. Thread engagement and fastener failure.

Since five threads carry the entire load regardless of thread type, a decrease in the minor diameter increases the shear area and gives an advantage to the female threads while reducing the load carrying capability of the male fastener. Conversely, an increase in the minor diameter increases the male fastener's cross-sectional area and gives an advantage to the male fastener, however, this reduces the shear area and weakens the female threads. Therefore, if the female fastener material is weak compared to the male fastener material, the female fastener should be given the advantage and coarse threads should be chosen. If the female fastener material is strong compared to the male fastener material, the male fastener will always fail first and should consequently be given the advantage by selecting fine threads.

For this reason steel bolts and studs that thread into relatively weak aluminum or cast iron castings such as engine blocks, cylinder heads and gearboxes are always coarse threaded on the end that goes into the casting. Also invariably, the end of the stud that receives the nut is provided with a fine thread. In this way the designer ends up with the best of both worlds.

Because coarse threads are faster to assemble, they are often used in applications where strength and weight are not of utmost concern. Conversely, virtually all aerospace bolted assemblies feature fine threads. **Generally, unless threading into a relatively weak material, avoid coarse threaded fasteners.**

## 5. Rolled Threads Versus Cut Threads

All quality fasteners have rolled threads produced via rolling or sliding dies as seen in figure 6 or [in this video on fastener manufacturing](#). Rolled threads (as opposed to threads [cut on a lathe](#), with a [cutting die](#) or [tap](#)) produce superior surface finish (thus lower stress risers) and improved material properties from cold working the material, resulting in much higher fatigue resistance. Rolled threads increase thread strength by a **minimum** of 30% over well-cut threads.<sup>1</sup>

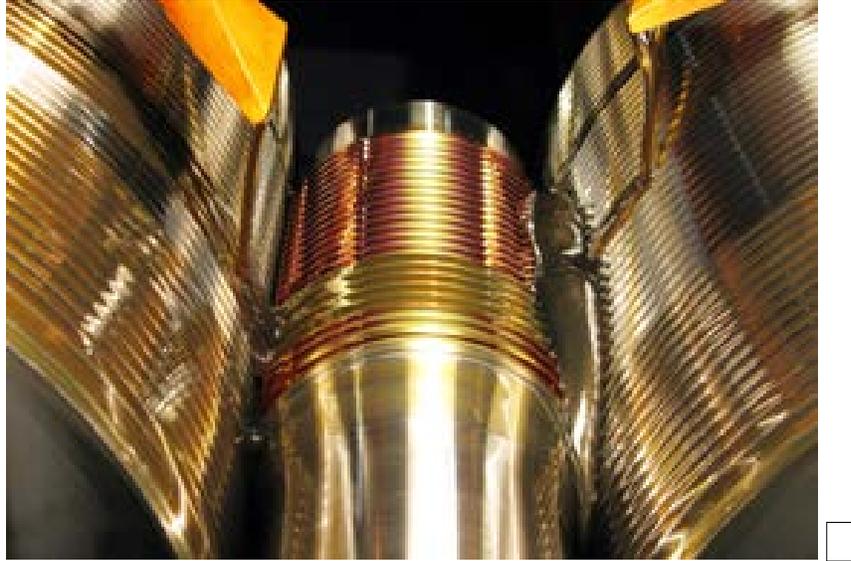


Figure 6. Rolled threads.

As illustrated in figure 7, when a thread is cut into a specimen, the grain flow of the material is severed. When a thread is rolled into a specimen, however, the grain flow of the material remains continuous and follows the contour of the thread. For this reason, rolled threads better resist stripping because shear failures must take place across the material grain rather than with it.

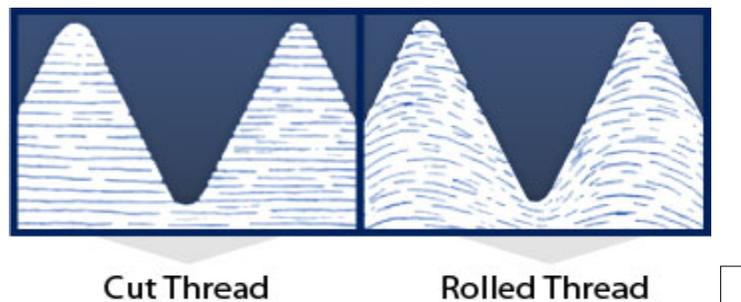


Figure 7. Cut vs. rolled thread grain flow.

<sup>1</sup> EBC Industries

As seen in figure 8, another benefit of thread rolling is it produces a much better surface finish than thread cutting. The surface factor plot presented in figure 9 illustrates the relationship between stress concentrations and surface finish, which clearly shows that on high strength fasteners, rolled threads possess up to twice the fatigue resistance compared to cut threads.

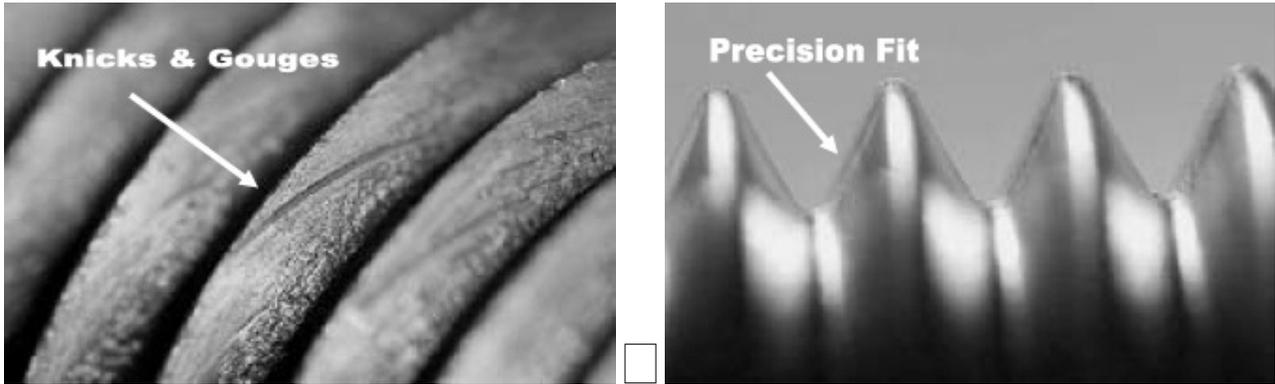


Figure 8. Cut vs. rolled thread surface finish and thread profile.

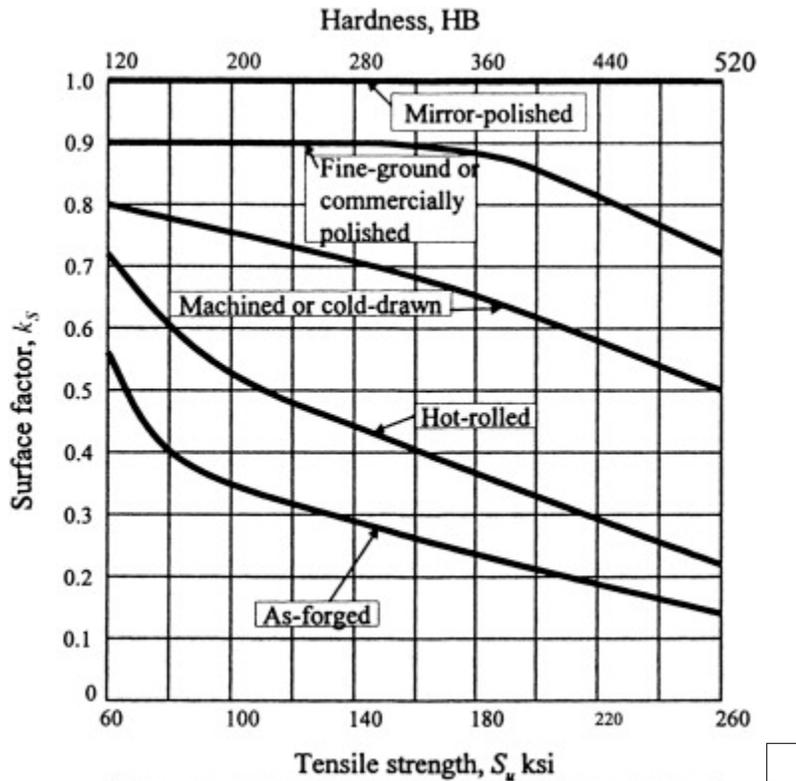


Figure 9. Surface finish modification factor as a function of fastener strength for rolled (*polished*) and cut (*machined*) threads.

Rolling also leaves the surface of the threads, particularly in the roots, stressed in compression. These compressive stresses must be overcome before the tensile stresses can reach a level that will cause fatigue failures. Compressive surface stresses also increase root hardness, further adding to the part's fatigue resistance.

Improved fatigue strength resulting from the above factors is reported to be on the order of 50% - 75%. On heat-treated bolts from Rockwell C36 to 40 hardness that have threads rolled after heat-treatment, tests show increased fatigue strength of 5 to 10 times that of cut threads.

Now consider the effect of heat-treatment on the final thread profile. All quality fasteners must be heat-treated to achieve the desired strength and toughness. The heat-treatment process inevitably results in some physical distortion of the fastener blank. Rolling the thread onto the (already) heat treated blank ensures the thread will be coaxial with the bolt and normal to the bearing surface of the fastener head, which is critical for proper function. Finally, due to [the speed at which fastener threads can be rolled onto a blank with the proper equipment](#), rolled threads can actually be more economical to manufacture in larger quantities.

## 6. Fastener Function

**Fasteners have only ONE intended function: to clamp parts together.** Fasteners are not meant to position parts relative to one another; that is the function of dowel pins (figure 10), locating shoulders and piloting diameters. Additionally, fasteners are not meant to function as pivots, axles and fulcrums; pins appropriately serve this function. (Note: students often get away with using fasteners to locate parts on the designs in this course because of the light duty, short-term use of the project and convenience of doing so; in use these bolted connections will loosen, causing the assembly to fail. This problem is avoided by regularly checking for loose fasteners prior to testing.)

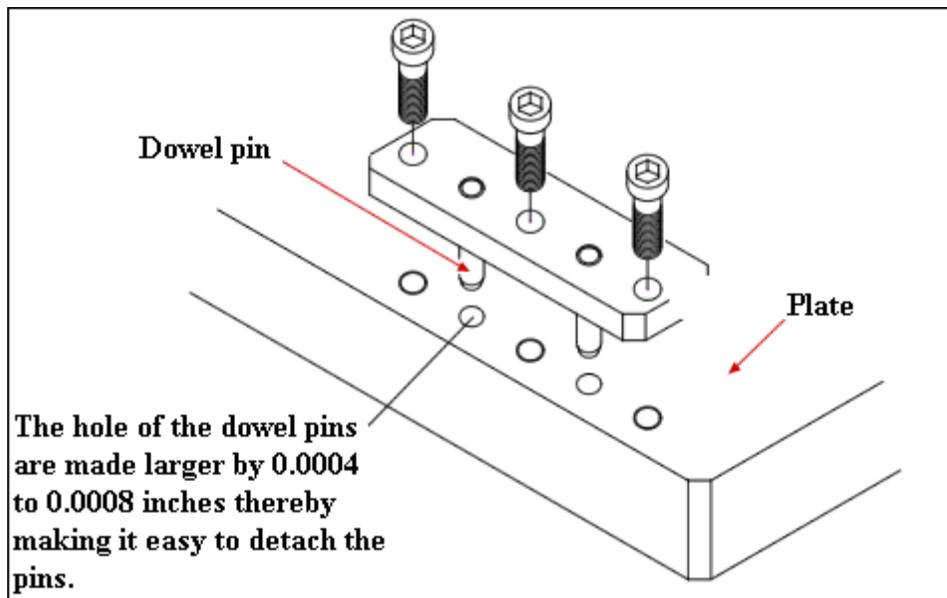


Figure 10. Proper use of dowel pins to position parts and resist shear forces.

More importantly, **the threaded portion of a fastener should NEVER be loaded in shear** for at least three reasons. First, the threaded portion of the fastener is of slightly smaller diameter than the unthreaded shank, allowing the fastener to quickly loosen if transverse loading is applied (and dowel pins are not appropriately used to resist the shear stresses). Second, the threaded portion of the bolt has much less surface area than the shank (figure 11), which means it offers **significantly** less bearing area to the joint; this reduces the load carrying capacity and fatigue resistance of the assembly. Third, when (not if!) the relative motion between the hole and the loose fitting threaded portion of the bolt occurs, the thread will act as a low speed file, removing material from the inside of the hole, exacerbating the problem. **So good design engineers NEVER load fastener threads in shear.**

# 1/4-20 UNC UNDER SHEAR LOADING

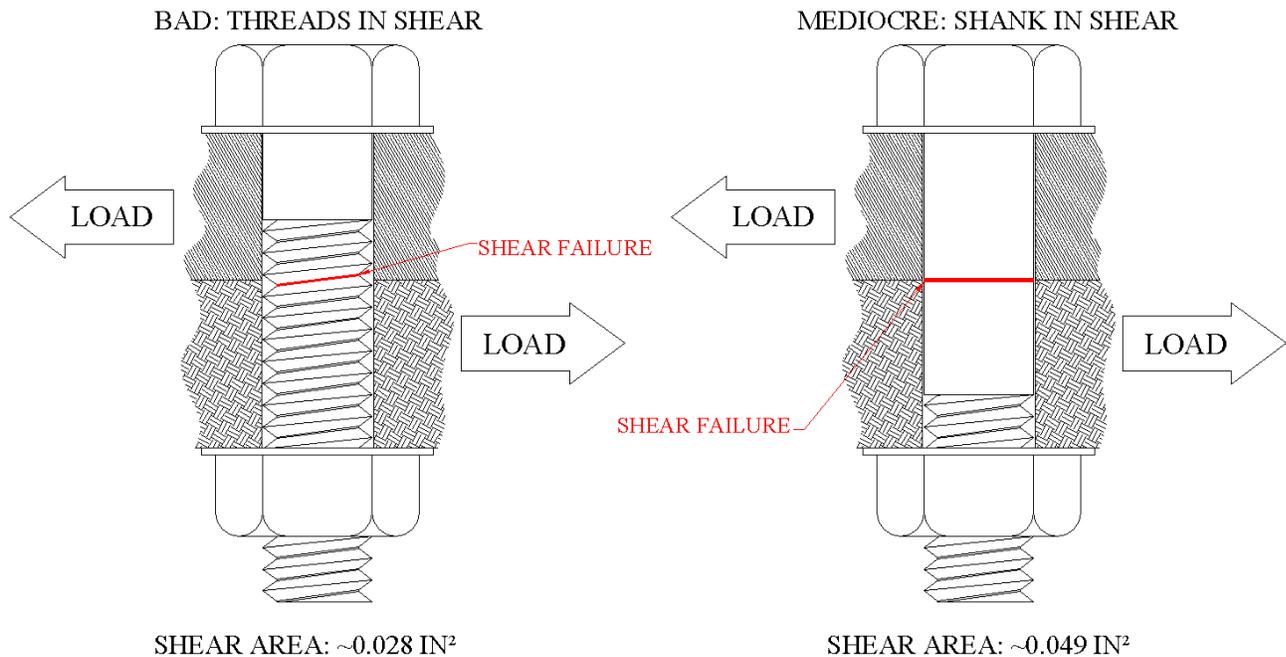


Figure 11. Never place fastener threads in shear; use dowel pins when possible and place the shank in shear when necessary.

## 7. Over-tightening vs. Under-tightening

As design engineers, each of us is ultimately responsible for the success or failure of the components and systems we design. When it comes to fasteners, it is important to understand the consequence of over-tightening versus under-tightening fastener joints. Un-intuitively, it is actually better to over-tighten a bolted joint than to under-tighten it! To explain this statement, it is helpful to review a common stress-strain curve for a typical fastener material, as shown in figure 12.

For those who have not taken a materials course, *stress* is a measure of how much tensile load is placed on the fastener and *strain* is a measure of the fastener's change in length (i.e. stretch). As seen in figure 12, there is a linear (“*elastic*”) region in which the fastener will return to its original length when the load is relaxed. The *yield strength* denotes the point (or magnitude) above which the material yields or permanently deforms. As additional load is placed on the fastener beyond the onset of yield, the strength actually increases as the material undergoes strain hardening, to the maximum (or “*ultimate*”) stress point. Beyond the *ultimate tensile strength*, the material begins to neck (which is a local reduction of cross sectional diameter) and finally fails at the point of *fracture*.

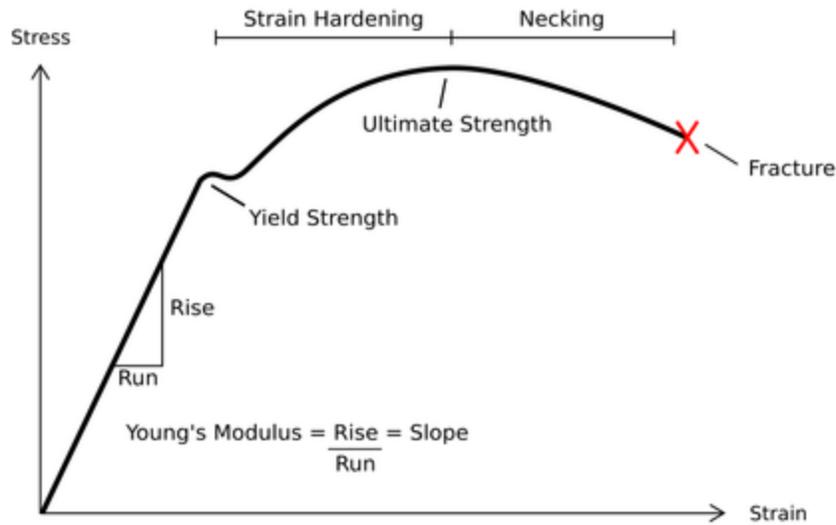


Figure 12. Stress versus strain plot for alloy steel.

The following example is taken from Carroll Smith's *Nuts, Bolts, Fasteners and Plumbing Handbook*. A 3/8" diameter bolt with an ultimate tensile strength (UTS) of 180,000 psi is torqued to 40% of its UTS (72,000 psi) and subjected to a cyclic tension load of 12,000 lb<sub>f</sub> using an Instron testing machine (figure 13). The 3/8" bolt will endure 4,900 force application cycles before failure. Next, an IDENTICAL bolt is torqued to 60% of its UTS (108,000 psi) and subjected to THE SAME cyclic tension load of 12,000 lb<sub>f</sub>. This identical bolt will endure 6,000,000 force application cycles before failure, or roughly **1000 TIMES more stress cycles (or service life)**.

**The previous example demonstrates the necessity for engineers to specify the correct installation torques of all fasteners used in critical assemblies. At the end of the day, PROPER INSTALLATION TORQUE (I.E. FASTENER TENSILE PRELOAD) IS WHAT KEEPS A PROPERLY DESIGNED FASTENER ASSEMBLY TIGHT. Contrary to popular misbelief, so called "lock washers" do not keep fastener joints tight; anaerobic adhesives (such as "Loctite") do not keep fastener joints tight; safety wire does not keep fastener joints tight, elastic stop nuts will not keep fastener joints tight, nor will castellated nuts and cotter pins. Several of these will help prevent a loosened fastener from falling off completely for a limited time, but NONE are a replacement for a properly designed and torqued (i.e. preloaded) fastener joint.**

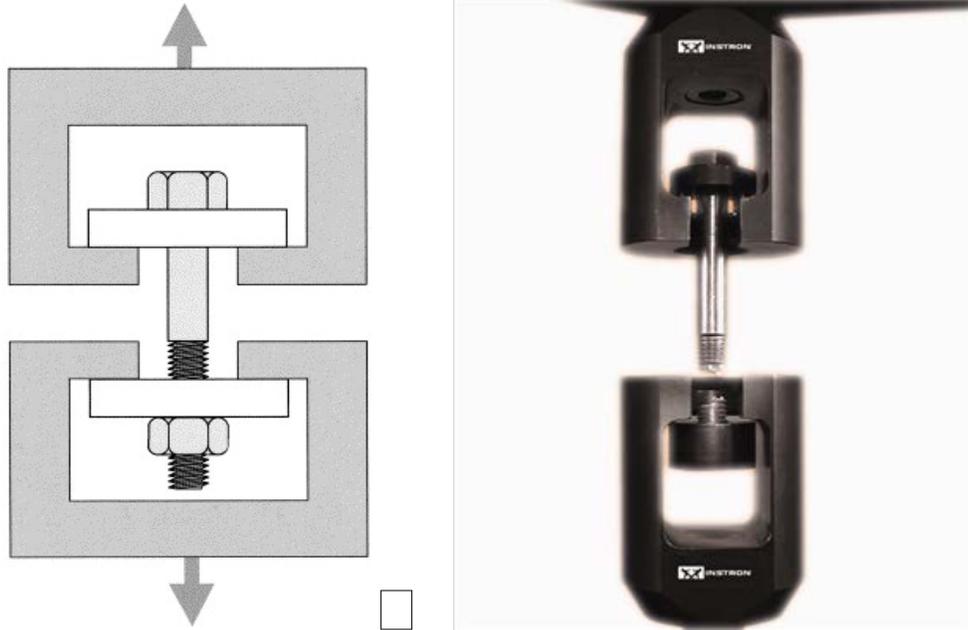


Figure 13. Tensile testing to measure number of cycles before fastener failure.

## 8. Calculating Proper Fastener Torque

The previous section illustrates the importance of specifying the tightening torque for mission critical fasteners; this section explains how to do so and is therefore one of the most useful pieces of information you can take away from this course, so please review it carefully. Equation 1 relates desired fastener preload (or tension) to the installed (or measured) fastener torque:

$$T \approx 0.2 \times F_i \times d \quad (\text{Eq. 1})$$

where  $T$  is the measured installation torque (measured with a torque wrench)  
 $F_i$  is the desired preload (installed tensile force in the bolt)  
 $d$  is the nominal bolt (shank) diameter

Equation 1 assumes a coefficient of friction ( $\mu$ ) of 0.15, which is average and varies based on lubricant and fastener plating. This simple equation results in approximately 80% accuracy, which is within the tolerance of most fastener specifications. On critical fastener installations, the desired preload (tensile stress) for optimum performance is equal to the yield (or *proof*) strength. To allow for a conservative margin for error, **aim for an installed tensile stress ( $\sigma_t$ ) equal to approximately 90% of the ultimate yield strength ( $\sigma_y$ ):**

$$\sigma_t \approx 0.9 \times \sigma_y \quad (\text{Eq. 2})$$

**Example 1:** calculate the proper tightening torque (lb<sub>f</sub>-ft) of a grade 5, 3/8-16 bolt:

from Table 7-5:  $\sigma_y = 85,000$  psi

from equation 2 we desire:  $\sigma_t \approx 0.9 \times \sigma_y = 76,500$  psi

since the installed tensile stress is equal to the bolt preload ( $F_i$ ) divided by the tensile stress area ( $A_t$ ), we can write:

$$\sigma_t = F_i / A_t$$

for a 3/8-16 fastener thread, Table 7-1 gives  $A_t = 0.0775$  in<sup>2</sup>

therefore  $F_i = \sigma_t \times A_t = 76,500$  psi  $\times$   $0.0775$  in<sup>2</sup> =  $5929$  lb<sub>f</sub>

now calculate the tightening torque, T using equation 1:

$$T \approx 0.2 \times F_i \times d = 0.2 \times 5929 \text{ lb}_f \times 0.375 \text{ in} = 444 \text{ lb}_f\text{-in} = \mathbf{37 \text{ lb}_f\text{-ft (answer)}}$$

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For a real design a first approximation of bolt size would be obtained by defining the factor of safety as:

$$n = F_i / P$$

Then the bolt size is determined such that its preload is larger than the external tensile load by the amount of the factor of safety.

Table 7-5 SAE specifications for bolts, cap screws, and studs

Grade	Bolt size, diameter, in.	Proof strength, psi	Tensile strength, min, psi	Hardness		Material—heat-treatment
				Brinell	Rockwell	
0	1/4 to 1 1/2	...	....	....	....	No requirements.
1	1/4 to 1 1/2	...	55,000	207 max	95 B max	Commercial steel.
2	1/4 to 1/2	55,000	69,000	241 max	100 B max	This is intended to be a cold-headed product made from low-carbon steel; C 0.28 max, P 0.04 max, and S 0.05 max. Lengths over 6 in. may be hot-headed from medium carbon steel, C 0.55 max. Deviation from specified chemistry may be made by agreement between producer and consumer.
	Over 1/2 to 3/4	52,000	64,000			Commercial steel.
3	Over 3/4 to 1 1/2	28,000	55,000	207 max	....	Produced by the cold-heading process, up to and including 6 in. in length from medium-carbon steel; C 0.28 to 0.55, P 0.04 max, and S 0.05 max.
	1/4 to 1/2	85,000	110,000	207-269	95-104 B	Medium-carbon steel; C 0.28 to 0.55, P 0.04 max, and S 0.05 max.
	Over 1/2 to 5/8	80,000	100,000			Quenched and tempered at a minimum temperature of 800 F.
5	1/4 to 3/4	85,000	120,000	241-302	23-32 C	Medium carbon steel; C 0.28 to 0.55, P 0.04 max, and S 0.05 max.
	Over 3/4 to 1	78,000	115,000	235-302	22-32 C	Oil quenched and tempered at a minimum temperature of 800 F.
	Over 1 to 1 1/2	74,000	105,000	223-285	19-30 C	Medium-carbon fine-grain alloy steel,* C 0.28 to 0.55, P 0.04 max, and S 0.05 max, providing sufficient hardenability to have a minimum oil-quenched hardness of 47 RC at the center of the threaded section one diameter from the end of the bolt. Oil quenched and tempered at a minimum temperature of 800 F. Roll-threaded after heat-treatment.
6	1/4 to 5/8	110,000	140,000	285-331	30-36 C	
	Over 5/8 to 3/4	105,000	133,000	269-331	28-36 C	
7	1/4 to 1 1/2	105,000	133,000	269-321	28-34 C	
8	1/4 to 1 1/2	120,000	150,000	302-352	32-38 C	Medium-carbon fine-grain alloy steel,* C 0.28 to 0.55, P 0.04 max and S 0.05 max, providing sufficient hardenability to have a minimum oil-quenched hardness of 47 RC at the center of the threaded section one diameter from the end of the bolt. Oil-quenched and tempered at a minimum temperature of 800 F.

\* Carbon steel may be used by agreement between producer and consumer. Note: Carbon range is for check analysis of product.

**Table 7-1 Diameters and areas of unified screw threads, UNC and UNF**

Size designation	Nominal major diameter, in.	Coarse series—UNC			Fine series—UNF		
		Threads per inch, <i>N</i>	Tensile-stress area <i>A<sub>t</sub></i> , in. <sup>2</sup>	Minor-diameter area <i>A<sub>r</sub></i> , in. <sup>2</sup>	Threads per inch, <i>N</i>	Tensile-stress area <i>A<sub>t</sub></i> , in. <sup>2</sup>	Minor-diameter area <i>A<sub>r</sub></i> , in. <sup>2</sup>
0	0.0600				80	0.00180	0.00151
1	0.0730	64	0.00263	0.00218	72	0.00278	0.00237
2	0.0860	56	0.00370	0.00310	64	0.00394	0.00339
3	0.0990	48	0.00487	0.00406	56	0.00523	0.00451
4	0.1120	40	0.00604	0.00496	48	0.00661	0.00566
5	0.1250	40	0.00796	0.00672	44	0.00830	0.00716
6	0.1380	32	0.00909	0.00745	40	0.01015	0.00874
8	0.1640	32	0.0140	0.01196	36	0.01474	0.01285
10	0.1900	24	0.0175	0.01450	32	0.0200	0.0175
12	0.2160	24	0.0242	0.0206	28	0.0258	0.0226
1/4	0.2500	20	0.0318	0.0269	28	0.0364	0.0326
5/16	0.3125	18	0.0524	0.0454	24	0.0580	0.0524
3/8	0.3750	16	0.0775	0.0678	24	0.0878	0.0809
7/16	0.4375	14	0.1063	0.0933	20	0.1187	0.1090
1/2	0.5000	13	0.1419	0.1257	20	0.1599	0.1486
9/16	0.5625	12	0.182	0.162	18	0.203	0.189
5/8	0.6250	11	0.226	0.202	18	0.256	0.240
3/4	0.7500	10	0.334	0.302	16	0.373	0.351
7/8	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
1 1/4	1.2500	7	0.969	0.890	12	1.073	1.024
1 1/2	1.5000	6	1.405	1.294	12	1.315	1.260

Once the required torque is calculated, the final step is ensuring it is achieved, which is performed by either measuring the torque on the fastener during installation (figure 14) or by measuring how much the fastener stretches after installation (figure 15).



Figure 14. Types of wrenches used to measure fastener installation torque/preload.

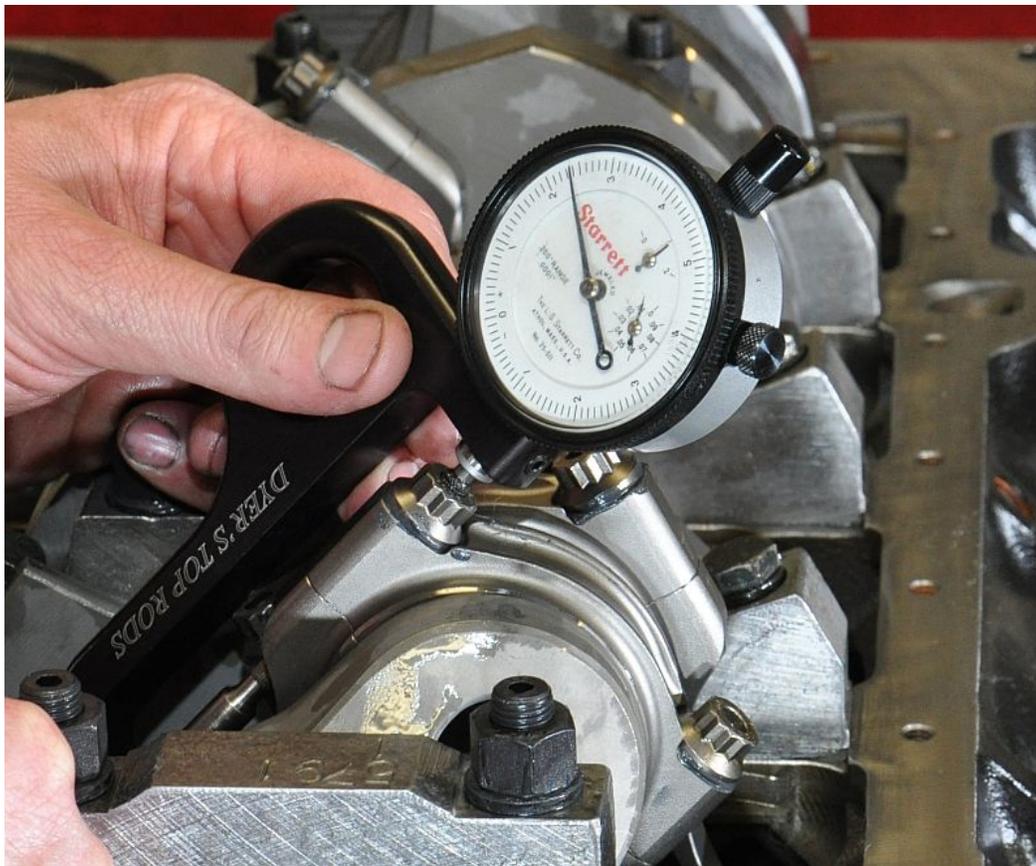


Figure 15. Measuring bolt strain (stretch) to verify proper preload.

## 9. Fastener Choices

The only fastener choices available “off the shelf” (OTS) are those listed on the [drill and tap chart](#). These are THE ONLY options available to design engineers when selecting fasteners, as anything else would require prohibitively expensive custom tooling and fasteners. **So ALWAYS reference a tap chart when selecting fasteners.**

## 10. Fastener Designations

In general, fasteners are referred to by their shank size (i.e. a ½" or a 12mm fastener). Standard (inch) fasteners are referred to by their shank size and the number of threads per inch they possess—for example 3/8"-16 or ½"-20, and are pronounced “three-eighths sixteen” or “one-half twenty.” Standard (inch) fasteners which are ¼" and larger are referred to by their nominal shank size. Fasteners smaller than ¼" are referred to by “screw size” designations, such as “number 10 or number 6”. Metric fasteners are referred to by their shank size and thread pitch—for example M6x1.0 or M10x1.5, and are pronounced “metric six by one” or “M ten by one point five”.

## 11. Tap Drills / Holes

The term *tap drill* refers to the **final drill size** used to make a properly sized hole prior to using a [tap](#) (internal threading tool) to cut threads into the hole. Note every tap drill is a standard drill, but not every standard drill is a tap drill. The size of the tap drilled hole is critical for each fastener and must be obtained from a [tap chart like the one provided in this course](#) or found in the Machinery Handbook. If this hole size is too small the tap will break when trying to create the threads; if the hole size is too large, the threads will be weak and fail (shear) in service. Note the tap drill size listed on the tap chart for a particular thread depends on the material type (weak or strong).

## 12. Clearance Drills / Holes

*Clearance drills* are used to create exactly what the name implies: clearance holes for fasteners, shafts or pins. Industry standards for clearance holes are listed in the last column of the [drill and tap chart](#). Note there are only two options for each size fastener: a close fit or free fit. Close fit clearance holes are used when you want a more accurate bolt pattern (i.e. when using the DRO on the milling machine) and free fit clearance holes are used when you want to save time and make the clearance holes more quickly (i.e. when using a drill press). Clearance hole sizes listed on the [tap chart](#) are industry standards every manufacturing facility will stock and should always be used unless you have a **very** good reason to deviate.

### 13. Fastener Joint Design

When designing fastener joints, **always design the most stable joint that provides the greatest bearing area.** As an example, consider the common shear joint design shown below in figure 16. The joint design is said to be in *single shear* if the fastener would only need to shear in one plane for the joint to fail; whereas the joint design is said to be in *double shear* if the fastener would need to shear in two planes for the joint to fail. Obviously the double shear mount is stronger and more stable than the single shear mount, and should be used at every available opportunity when joint strength and reliability are of concern. In fact, in the great words of the late Carroll Smith: “[when it comes to weight conscious, mission critical designs] the single shear mount in a crime against nature and a perversion of the bad engineer!”

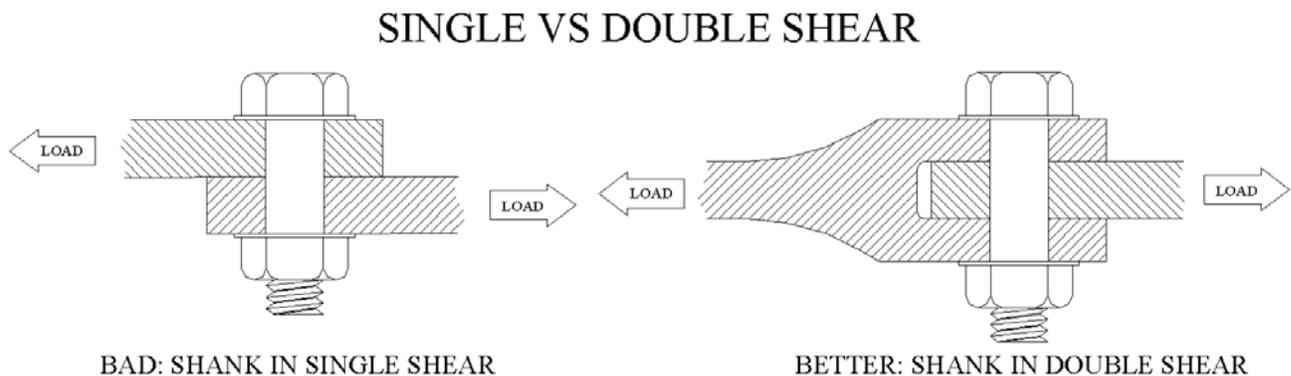


Figure 16. Single shear versus double shear joint design.

Another important rule for fastener joint stability is to **never place fastener holes closer than one major diameter to a workpiece edge**, as doing so does not leave adequate material to resist the bearing stress. As shown in Figure 16 above, the hole locations in each workpiece provide the minimum amount of required material for the fastener joint, resulting in the lightest design which does not compromise strength.

## 14. Application Examples

**Example 1:** You are tasked with designing a motor mount that will attach via 1/4-20 fasteners to 80/20 aluminum extrusion. The tolerances are loose ( $\pm 0.125$ "), the loads are light, and assembly time is important. What hole callout should you use to specify the two thru clearance holes for the 1/4-20 fasteners?

**Answer 1:** Since assembly time is of concern, tolerances are loose, and loads are light, free fit clearance holes are the correct design choice.

The hole callout format is:

*Ø free fit drill diameter + depth;  
quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 0.266" THRU; 2 PLACES
----------------------------

**Example 2:** You are tasked with designing a motor mount that will attach via 10-32 fasteners to a motor with a four hole bolt pattern. Assembly time is at a premium but accuracy is necessary for part function. What hole callout should you use to specify the thru clearance holes for the 10-32 fasteners?

**Answer 2:** Since the motor mount will be mating with an existing bolt pattern (which will be precise), a close fit hole specification should be used. However, a free fit hole would be best for reducing assembly time. **Since the part requires high accuracy, regardless of the additional time it takes to create a close fit hole pattern, a close fit hole pattern is required.**

The clearance hole callout format is:

*Ø close fit drill diameter + depth;  
quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 0.196" THRU; 4 PLACES
----------------------------

**Example 3:** You need to specify the hole callout for three threaded (i.e. “tapped”) holes through an aluminum part. 6mm metric screws will mate with the part. How should you specify the hole callout?

**Answer 3:** Since the material is aluminum (which is relatively soft and weak compared to high strength steel fasteners), coarse threads should be specified.

The tapped hole callout format is:

*Ø tap drill diameter + depth;*  
*thread specification + depth;*  
*quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 5.00 THRU; M6x1.0 THRU; 3 PLACES
--

**Example 4:** You need to specify the hole callout for four tapped holes through a steel bracket. Five metric fasteners of size 4mm will mate with these holes. How should you specify this hole?

**Answer 4:** Knowing the workpiece is steel (which is relatively strong), fine threads should be specified for maximum joint strength.

The tapped hole callout format is:

*Ø tap drill diameter + depth;*  
*thread specification + depth;*  
*quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 3.50 THRU; M4x0.70 THRU; 4 PLACES
---

**Example 5:** You are asked to specify the hole callout for a single tapped hole through a steel wheel hub. A standard size 10 fastener will mate with this hole. How should you specify the hole callout?

**Answer 5:** Knowing the workpiece is steel (which is relatively strong), fine threads should be specified for maximum joint strength.

The tapped hole callout format is:

*Ø tap drill diameter + depth;  
thread specification + depth;  
quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 0.170 THRU; 10-32 UNF THRU
---------------------------------

**Example 6:** You are asked to specify the hole callout for six screw holes through an aluminum part. Six standard fasteners of size 3/8" will mate with these holes. How should you specify the hole callout?

**Answer 6:** Since the material is aluminum (which is relatively soft and weak compared to high strength steel fasteners), coarse threads should be specified.

The tapped hole callout format is:

*Ø tap drill diameter + depth;  
thread specification + depth;  
quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 0.313 THRU; 3/8-16 UNC THRU; 6 PLACES
---

**Example 7:** Specify seven holes tapped half way through a 1" part for use with 10-24 screws.

**Answer 7:** Although the material is not specified, you should be able to recognize the part is a weak material (such as aluminum or a casting) OR emphasis is placed on reducing assembly time at the expense of joint strength and / or weight.

The tapped hole callout format is:

*Ø tap drill diameter + depth;*  
*thread specification + depth;*  
*quantity of holes desired (not included if only one hole is specified)*

Notice that when working with blind threads (i.e. not thru) you must drill an additional screw diameter (Ø 0.190") beyond the desired thread depth. From the [drill and tap chart](#), the hole specification is:

Ø 0.157 0.7 DEEP; 10-24 UNC 0.50 DEEP; 7 PLACES
---

**Example 8:** Specify the hole callout for one 5/8" screw thread through an aluminum bracket and calculate the minimum part thickness to ensure proper thread strength.

**Answer 8:** Since the material is aluminum, coarse threads should be specified.

The tapped hole callout format is:

*Ø tap drill diameter + depth;*  
*thread specification + depth;*  
*quantity of holes desired (not included if only one hole is specified)*

From the [drill and tap chart](#), the hole specification is:

Ø 0.531 THRU; 5/8-11 UNC THRU
----------------------------------

( problem is continued on next page >> )

Since a MINIMUM of five threads must be engaged for a fastener joint to achieve full strength, the required workpiece thickness can be determined using the thread pitch (11 TPI in this case) and basic unit analysis:

$$5 \text{ threads} \times (11 \text{ threads/in})^{-1} = 5/11 \text{ in}$$

Therefore, the workpiece must be greater than 0.455" thick for proper strength.

**Example 9:** If you have two 16GA (gauge) steel sheetmetal parts that must be fastened together how would you design the fastener joint and what hole callouts should you specify for the mating pieces?

**Answer 9:** 16GA steel sheetmetal has a thickness of ~0.060". Five full threads of engagement are needed if the sheetmetal is going to be tapped. By unit analysis:

$$5 \text{ threads} / 0.060 \text{ in} = 83 \text{ threads/in}$$

Since no fastener on the [drill and tap chart](#) has 83 threads per inch or more, threading the sheetmetal is NOT a viable option. Therefore, thru holes should be specified in both parts for use in a bolted assembly (i.e. fasteners that mate with nuts). The size of the fastener is not specified and is a free design parameter. We choose a 10-32 UNF arbitrarily (no other design information is given). Note the fine thread option is selected to maximize the strength of the bolted assembly.

The clearance hole callout format is:

*Ø clearance drill diameter + depth;  
quantity of holes desired (not included if only one hole is specified)*

Depending on whether the design calls for close or free fit clearance holes (not enough information is given), find the required hole size using the [drill and tap chart](#):

Ø 0.196" THRU

OR

Ø 0.201" THRU

Alternatively, rivets could be used to semi-permanently join the two parts; however more work would be required to remove the rivets if the components required disassembly in the future.