It's easier to tap brittle material if coarse-pitch threads are used. Such threads are also easier to use in most cases: easier to start, faster rundown, etc.

B. Fine-Pitch Threads

Fine-pitch threads must be close fitting—made to Class 3 tolerances—to have acceptable stripping strength, but if this is done the bolts these threads are used on can have higher tensile strengths because the thread root and pitch diameters—and therefore the tensile stress area, A_S—are greater than they would be for a coarse-pitch thread on the same nominal diameter. This advantage can be obtained, however, only with a suitably long length of engagement between male and female threads. We'll study this subject later on in this chapter.

Fine-pitch threads are stronger in torsion, which means that they can be loaded to higher preloads before yielding. They also resist self-loosening under vibration or shock, and resist stress corrosion cracking, better than do coarse pitch threads.

C. Constant-Pitch Threads

Constant-pitch threads are designed for applications where there will be repeated assembly and disassembly and/or where it may be necessary to rethread the part in service. They're used for adjusting collars, for thin nuts or threaded sleeves on shafts. They're also used in the design of compact parts [2, 4].

The 8-thread series is used on large-diameter fasteners and was originally intended for bolts used in gasketed joints containing high pressure. It's also widely used as a substitute for coarse series fasteners when the basic fastener diameter exceeds 1 in.

The 12-thread series is used as a continuation of the fine thread series when bolt diameters exceed $1\frac{1}{2}$ inch. It was also originally intended for pressure vessels but has now found wider use.

The 16-thread series is also used on large-diameter fasteners, again for those requiring fine-pitch threads. It's used as well for adjusting collars and as a continuation of the extra fine pitch series for bolt diameters over $1\frac{1}{16}$ inch.

D. Miscellaneous Factors Affecting Choice

We'll see other thread characteristics which may affect our choice of thread as we proceed through the book, but a few miscellaneous comments may be in order here.

A tighter fit, i.e., 3A vs. 2A, gives a 10% increase in thread stripping strength, because there's more root cross section to be sheared. The rounded roots of the J profile will increase the strength still further.

The UNJ or MJ threads also have more resistance to fatigue than do the UN/UNR or M threads [21].

Threads tend to strip before the bolt breaks if the male-female fit is loose [10].

The number of threads in the grip (between the face of the nut and the head of the bolt) affects the ductility and stiffness of the fastener. Since we (usually) want ductility and low stiffness (a more resilient spring for better energy storage) it would seem that we'd usually want fully threaded fasteners. We'll be especially interested in ductility if using yield control to tighten the fasteners. (See Chap. 8.)

Factors like the shear strength of the fastener and its fit with its hole, however, often argue instead for partial threads and an unthreaded body of nominal or reduced diameter.

VIII. THE STRENGTH OF THREADS

There's a surprising amount of disagreement on what parameters determine the strength of a thread and on how best to evaluate the quality—including the strength—of a threaded fastener before use. Let's take a look at some conventional wisdom concerning thread strength and then look at some recent thoughts and concerns about thread strength and quality.

A. Basic Considerations

As we saw in Chap. 1, one of our principal design goals is "a fastener strong enough to support the maximum preload it might receive during assembly, plus the maximum additional loads it might see in service, as a result of forces applied to the joint, differential thermal expansion, etc." The larger the nominal diameter of a fastener, of course, the stronger it will be. As far as static loads are concerned, therefore, we'd like the shank or body of the bolt to be the full, basic, or nominal diameter of the thread, or at least to be greater than the root diameter of the threads [2].

We must then specify a length of thread engagement capable of developing the full strength of that body. This is just another way of saying that we want the body to break before the threads strip, because a broken bolt is easier to detect than a stripped thread.

When the threads strip they do so by shearing in one of three ways. If the nut material is stronger than the material from which the bolt is made, the threads will strip at the roots of the bolt teeth. If the bolt material is stronger, stripping will occur at the roots of the nut threads. If the materials have equal strengths, both nut and bolt threads will strip simultaneously, at their pitch diameters.

Studies made at the National Bureau of Standards many years ago showed that the tensile/shear strength ratio for common fastener materials varied from 1.7 to 2.0. As a result, the stripping areas (A_{TS}) defined in the formulas below (on which the recommended lengths of thread engagement are based) are set at twice the tensile stress area (A_S) of the same thread [5].

If the fastener is to be subjected to fatigue or impact loads, we'd like it to be more resilient than a fastener subjected to static loads. Some recommend a shank (body) diameter about 60% of that used for static loads if the fastener will see impact loads, or a shank diameter of 90% of the static diameter if it will experience fatigue loads (repeated load cycles) [2].

B. The Static Shear Strength of a Thread

We use a very simple equation to estimate the force required to strip (shear) the threads of a bolt or nut:

$$F = S_U A_{TS} \tag{3.1}$$

where S_U = ultimate shear strength of the nut or bolt materials (psi, MPa)

A_{TS} = cross-sectional area through which the shear occurs (in.^{2}, mm^2) (This is not the cross-sectional area of the body, as we'll see in a minute.)

As mentioned earlier, thread failure will occur in either the nut or bolt threads—or in both simultaneously—depending on the relative strengths of the nut and bolt materials. A different expression must be used to compute the shear stress area for each type of failure [11].

The following equations were taken for the first edition of this book from ANSI B1.1-1974 [11]. Modified versions of the equations for shear areas, A_{TS}, can be found in the current edition of B1.1 [2] but the equations for length of engagement, L_e, have been removed. All of the equations, however, can be found in the current FED-STD-H28/2B [3] but with modified nomenclature.

Simplified formulas for shear areas are also given in H28 and have been included below.

Those who wish to convert these equations to the present nomenclature can use Table 3.3.

Table 3.3 Old vs. New notation Used in Screw Thread Formulas

Old notation (Used in this book)	New notation
D_S	d
E_S	d
K_n	d_2
E_n	D_1
L_e	D_2
A_{TS}	LE
	AS_S for bolts
	AS_n for nuts

B. Nut Material Stronger than Bolt Material

Failure occurs at the root of bolt threads. The equations for shear area (A_{TS}) and the length of thread engagement (L_e in inches or millimeters) required to develop full strength of the threads are as follows:

$$A_{TS} = \pi n L_e K_{n\max}\left[\frac{1}{2n} + 0.57735(E_{S\min} - K_{n\max})\right] \tag{3.2}$$

$$L_e = \frac{2A_S}{\pi n K_{n\max}[(1/2n) + 0.57735(E_{S\min} - K_{n\max})]} \tag{3.3}$$

where A_{TS} = shear area at root of bolt threads (in.^{2}, mm^2)

n = number of threads per inch

A_S = tensile stress area of bolt (in.^{2}, mm^2)

$K_{n\max}$ = maximum inner diameter (ID) of nut (in., mm)

$E_{S\min}$ = minimum pitch diameter (PD) of bolt (in., mm)

As mentioned above, FED-STD-H28 gives simplified expressions for shear areas. Here's the expression used when the nut material is stronger than the bolt material:

$$A_{TS} = \pi E_S \frac{5L_e}{8} \tag{3.4}$$

where E_S = the basic (or nominal) pitch diameter of the external thread (in., mm) (see comments below).

Let's do an example to see how much difference we get when using the simple vs. the more complex expression and to see where to find the data required to use these equations. Incidentally, most of these equations—and many of the others found in this text—have been repeated in calculator/computer format in Appendix H for your convenience in this electronic age.

We'll take, as our example, a $\frac{3}{4}$–12 UN Class 2A thread and compute A_{TS} for a length of engagement equal to one diameter (the thickness of a heavy hex nut). We get the necessary data from a recent edition of either ASME B1.1 or *Machinery's Handbook* [24]. These tell us that:

$$K_{nmax} = 0.678 \text{ in.}$$

$$E_{Smin} = 0.6887 \text{ in.}$$

$$E_S = 0.6959 \text{ in. (see note below)}$$

Note that the basic or nominal pitch diameter of the bolt is equal to the minimum pitch diameter of the internal (nut) thread, because a zero allowance is assigned to the internal thread which, therefore, conforms to the basic profile at the pitch line and elsewhere.

From Eq. (3.2) in calculator format:

$$A_{TS} = 3.14159 \times 12 \times 0.75 \times 0.678 \times [1/(2 \times 12) + 0.57735(0.6887 - 0.678)] = 0.917 \text{ in.}^2$$

From Eq. (3.4)

$$A_{TS} = 3.14159 \times 0.6959 \times 5 \times 0.75/8 = 1.022 \text{ in.}^2$$

The difference is 12% in this case, with the simplified expression giving a slightly less conservative result (it would take more force to shear 1.022 in.^{2} than 0.917 in.^{2}).

C. Nut Material Weaker than Bolt Material

These are the equations you would use to compute the strength of a tapped hole in a joint material such as aluminum or cast iron. The use of aluminum is increasing in automotive and military applications, for example, as designers struggle to reduce weight. The strengths of various bolt and joint materials will be found in Chap. 4 in Tables 4.1, 4.2, and 4.14.

Failure occurs at the root of nut threads. The equations are

$$A_{TS} = \pi n L_e D_{min} \left[\frac{1}{2n} + 0.57735(D_{smin} - E_{nmax}) \right] \quad (3.5)$$

$$L_e = \frac{S_{st}(2A_S)}{S_{nt}\pi n D_{smin}[(1/2n) + 0.57735(D_{smin} - E_{nmax})]} \quad (3.6)$$

where D_{smin} = minimum OD of bolt threads (in., mm)
E_{nmax} = maximum PD of nut (in., mm)
S_{st} = tensile strength of the bolt material (psi, MPa)
n = threads per inch
A_S = tensile stress area of bolt (in.^{2}, mm^2)
S_{nt} = ultimate tensile strength of the nut material
A_{TS} = shear area of root of nut threads (in.^{2}, mm^2)
L_e = length of thread engagement required to develop full strength (in., mm)

The simple expression for shear area when the bolt material is stronger than the nut material, again from FED-STD-H28, is

$$A_{TS} = \pi E_n \frac{3L_e}{4} \quad (3.7)$$

where E_n = the basic (minimum) pitch diameter of the nut (in, mm).

D. Nut and Bolt of Equal-Strength Materials

Failure occurs simultaneously in both parts, at the pitch line. The equations are

$$A_{TS} = \pi E_S \frac{L_e}{2} \quad (3.8)$$

$$L_e = \frac{4A_S}{\pi E_S} \quad (3.9)$$

where E_S = nominal pitch diameter of the bolt (in., mm)
A_S = tensile stress area of bolt (in.^{2}, mm^2)
A_{TS} = shear area at pitch line of both threads (in.^{2}, mm^2)
L_e = length of thread engagement required to develop full strength (in., mm)

E. Things Which Modify the Static Strength of Threads

We saw earlier that there are a number of factors which can modify the anticipated tensile strength of a bolt—such things as high temperature, corrosion, torsion, or cyclic loading. These things can also modify the strength of threads. So can some other factors which aren't quite as obvious. For example:

Nut dilation [12, 13]. If the walls of the nut are not thick enough, the wedging action of the threads will dilate the nut, partially extracting the nut threads from the bolt threads. This reduces thread engagement

TORQUE-PRELOAD RELATIONSHIP

$$T_{in} = F \cdot \left(\frac{P}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right)$$

T_{in} = torque applied to the fastener (in.-lb, mm-N)
F = preload created in the fastener (lb, N)
P = the pitch of the threads (in., mm)
μ_t = the coefficient of friction between nut and bolt threads
r_t = the effective contact radius of the threads (in., mm)
β = the half-angle of the threads (30° for UN or ISO threads)
μ_n = the coefficient of friction between the face of the nut and the upper surface of the joint
r_n = the effective radius of contact between the nut and joint surface (in., mm)

For most cases the input torque is absorbed:

- **Bolt stretch component** **10%**
- **Thread friction** **40%**
- **Nut-clamping surface friction** **50%**