BASIC ELEMENTS OF SCREW THREAD DESIGN

IFI Note:

Unified screw threads are the standard for all inch series threaded mechanical fasteners manufactured and used throughout the world.

Presented in this Section are the nationally — and, internationally—recognized ANSI/ASME standards for Unified inch screw threads. Each has been suitably abstracted to apply to the fasteners covered in this book.

The contents of these engineering documents are technically precise with minimal explanation of principles or background rationale. For this reason, IFI considers it useful to introduce this Section with a "layman" discussion of screw thread basics. Its purpose is to explain in less formal language the more pertinent features of screw thread design and assist a fuller understanding of their proper application.

BASIC FEATURES OF SCREW THREADS

Screw threads give fasteners their ability to support and transfer loads.

There are over 125 separate geometrical features and dimensional characteristics in the design and construction of screw threads. With a familiarity of only about 30, the engineer can become comfortably conversant in the language of screw threads and gain an understanding of their performance capabilities. As these terms are being discussed, reference to the illustrations on pages A-27, 28, 29, and 34 will help.

A *screw thread* is a ridge of uniform section in the form of a helix on the external or internal surface of a cylinder. *External threads* are threads on bolts, screws and studs. *Internal threads* are those in nuts and tapped holes.

The configuration of the thread in an axial plane is *thread form* (profile) and the three parts making the form are the crest, root and flanks. The *crests* of threads are at the top, the *roots* at the bottom, and the *flanks* join them. The fundamental triangle is the triangle formed when the thread profile is extended to a sharp V at both crests and roots. The *height of fundamental triangle* (H) is the distance, measured radially, between the sharp crest and sharp root diameters. For Unified threads, H equals 0.866025 times thread pitch. The principal importance of H is in computation of thread construction values.

A thread having full form at both crests and roots is a *complete (or full form) thread.* When either the crest or root is not fully formed, it is an *incomplete thread.* Such threads occur at the ends of externally threaded fasteners which are pointed, at thread runouts where the threaded length blends into the unthreaded shank, and at the countersinks in the faces of nuts and tapped holes.

Thread pitch (p) is the distance, measured parallel to the thread axis, between corresponding points on adjacent threads. Unified screw threads are designated in *threads per inch,* which is the number of complete threads occurring in one inch of threaded length. Thread pitch is the reciprocal of threads per inch.

On an external thread, the diameter at the thread crests is the *major diameter,* and at its thread roots is the *minor diameter.* On an internal thread, it is the opposite, the diameter at the crests is the *minor diameter* and at the roots the *major diameter.*

The angle between a flank and a perpendicular to the thread axis is the *flank angle* and when both flanks have the same angle the thread is a *symmetrical thread* and the flank angle is termed *half-angle of thread.* Unified screw threads have a 30° flank angle and are symmetrical. For this reason they are frequently referred to as 60° threads.

Pitch diameter is the diameter of a theoretical cylinder that passes through the threads in such a position that the widths of the thread ridges and thread grooves are equal. On a perfect thread these widths would each equal one-half of the thread pitch. With anything less than a perfect thread, the actual pitch diameter, as measured at any point throughout the length or circumference of the thread, will vary dependent on variations in

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thread form as manufactured within the permitted limits of size. Consequently, among screw thread experts, the definition, measurement, and significance of pitch diameter is controversial. However, it is an important value for computation and thread design purposes, for production of manufacturing tools and dies, and for thread acceptance gages and gaging. The "perfect" pitch diameter is the generator of the *pitch cylinder,* the axis of which is the *thread axis.*

The *basic thread profile* establishes an absolute boundary between the product external and internal threads. If either trespasses beyond this boundary potential interference exists and the threads may not assemble. It is from this basic thread profile that the actual *limits of size* are derived by applying allowances and tolerances.

An *allowance* creates an intentional clearance between mating threads. This means that when both the external and internal threads are manufactured to their absolute *maximum material condition* there will be a positive space between them. For fasteners, the allowance is generally applied to the external thread, which means that its maximum major, pitch and minor diameters are less than basic by the amount of the allowance; the minimum diameters of the internal thread — its maximum material condition — are at basic. *Tolerances* are specified amounts by which dimensions are permitted to vary for manufacturing convenience. The tolerance is the difference between the maximum and minimum permitted limits. Thus, for external threads, its maximum material condition less its tolerances (moving toward the thread axis) defines its *minimum material condition.* And for internal threads, its maximum material condition plus its tolerances (moving away from the thread axis) defines its *minimum material condition.*

The combination of allowances and tolerances in mating threads is *fit* and is a measure of tightness or looseness between them. A *clearance fit* is one that always provides a free running assembly and an *interference fit* is one having specified limits of thread size that always result in a positive interference between the threads when assembled.

When assembling externally threaded fasteners into internally threaded nuts or tapped holes, the axial distance through which the fully formed threads of each are in contact is *length of thread engagement.* The distance these threads overlap in a radial direction is *depth of thread engagement.* Both length and depth of thread engagement are important values when computing thread strengths.

Thread series are groups of diameter-pitch combinations differing one from another by the number of threads per inch applied to a series of specific diameters. For fasteners, the popular thread series are Unified coarse, fine and 8-pitch.

Strengths of screw threads — i.e. their ability to support and transfer loads — are dependent on four stress areas. *Tensile stress area* is an assumed cross sectional area through the thread which is used when computing the load a fastener can support in tension. The tensile stress area is equivalent to the cross sectional area of a theoretical cylinder of the same material and mechanical properties which would support the same ultimate load when tested in tension to failure. *Thread root area* is the cross sectional area through an external thread at its minor diameter. Thread root area is used when computing a fastener's strength in transverse shear or in torsion. *Thread shear areas* — for external and internal threads — are the effective areas through the thread ridges, parallel to the thread axis and for the full length of thread engagement, which support the applied load in shear and resist the stripping out of either or both threads. The shear plane for the internal thread occurs at the major diameter of the external thread, and for external threads at the minor diameter of the internal thread.

The engineering standard giving terms, definitions, and symbols for screw threads is ANSI/ASME B1.7. Page A-18 presents an abstract of B1.7 listing the technical definitions of the terms discussed above and all others applicable to Unified inch screw threads for mechanical fasteners.

GUIDE TO THREAD SELECTION

Three considerations influence the choice of which thread is best suited for a given application — thread form, thread series and class of thread fit.

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THREAD FORM

There are literally dozens of different screw thread forms. However, for inch series mechanical fasteners, only three have significance — UN, UNR and UNJ. All are 60° symmetrical threads with essentially the same basic profile. The principal difference between them is the contour at the root of the external thread.

Prior to 1948, the American National thread was the standard in North America. In that year, USA, Canada and Great Britain agreed to adopt a single screw thread system to supplant the American National in USA and Canada and the Whitworth in Great Britain. They called this new system Unified and today it is the standard for all inch series fasteners throughout the world.

The Unified thread form is practically identical to the now obsolescent American National. In fact, threads manufactured to either are functionally interchangeable. (Refer to Figs. 1, 2A, 2B, and 3 of ANSI/ASME B1.1, page A-26.)

UN Threads

The UN thread form, as originally designed, provided for either a flat or rounded form in the root of the external thread. Each country made its choice in its own national standards. USA opted to permit flat roots, even though it was well recognized that stress concentrations in screw threads could be alleviated by rounding their roots. However, economics dictated. Thread roll dies and thread cutting tools were expensive and producing them with rounded crests to form radiused roots in the fastener threads added appreciably to their cost. Also, it was argued that new tools wear and within a few hundred pieces their crests round and product threads begin to take on the more desirable contoured form.

UNR Threads

During the 1950s, fastener performance demands escalated, particularly for safety-critical fasteners subjected to fatigue-inducing loads. It became imperative that fatigue-behavior improvements be explored. One obvious opportunity was to require controlled radiusing of the external thread root form. This led to the design and introduction of a modified thread form, designated UNR, with the single difference from UN being a mandatory root radius with limits of 0.108 to 0.144 times the thread pitch. The minimum radius of 0.108 p is the largest radius than can be fitted into the UN profile without violating the minimum material condition of the external thread. The maximum radius of 0.144 p is the largest radius that can be accommodated without causing theoretical interference with an internal thread at its maximum material condition.

When first introduced, it was necessary to specify UNR to assure delivery of fasteners with rounded roots. Today, however, virtually 100 percent of fasteners in nominal sizes 1 in. and smaller are manufactured with UNR threads, whether UNR is specified or not. This is because the threads on these sizes are normally produced by thread rolling, and thread roll dies with rounded crests are now the standard. For larger size fasteners, beyond thread rolling capability and whose threads are cut, when radiused root threads are needed, UNR should be specified, otherwise UN will probably be supplied.

UNK Threads

Shortly following the introduction of UNR threads came a further modification, designated UNK.

UNK threads are merely a more precisely engineered UNR thread, with exactly the same thread profile and the same root radii limits. The difference is that the minor diameter of the external thread is toleranced and inspection of the root to assure its radiusing is within specified limits is mandatory. UNK threads became the standard for socket head cap screws and socket set screws.

When thread acceptance gaging systems were developed in 1979 (ANSI/ASME B1.3, page A-53), UNK threads became obsolete. The reason was that UNR threads inspected using Gaging System 22 accomplished essentially the same objective.

UNJ Threads

The design of UNJ threads grew out of a search for an optimum thread form — one that

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would have superior fatigue resistant behavior without sacrificing static strength properties. In other words, just how generous could the root radius be.

The answer was a new thread form, designated UNJ, having root radius limits of 0.150 to 0.180 times thread pitch. With these enlarged radii, minor diameters of the external thread increase and intrude beyond the basic profile of the UN and UNR thread forms. Consequently, to offset any chance of interference between mating threads, the minor diameters of the UNJ internal threads had to be increased. This means a slight reduction in depth of thread engagement between mating UNJ external and internal threads. However, any strength loss is compensated for by requiring class 3A/3B thread tolerances (standard for UNJ threads) which maximizes the minimum material condition of both threads.

UNJ threads are now the standard for aerospace fasteners and have some usage in highly special industrial applications.

Thread Mating

UN internal threads assemble correctly with UN and UNR external threads. In fact, there is no UNR internal thread.

Theoretically, UN internal threads will not assemble with UNJ external threads. However, this combination has been used for years by some major fastener users and they claim trouble-free experience. Computer studies also substantiate that the risk of actual interference between manufactured parts is negligible. Even so, this practice is not recommended, especially if the fasteners are plated.

UNJ internal threads assemble correctly with UNJ external threads, and also fit both UN and UNR. However, this latter mating should be used with a degree of caution because the increased minor diameter of the UNJ internal thread reduces the stripping area (and stripping strength) of the external thread.

Some Additional Thoughts on Root Radiusing

For UN threads there is no mandatory root radius, the thread root can be flat. For UNR threads, the minimum root radius is 0.108 p, and for UNJ threads it is 0.150 p. It is difficult to believe that such modest differences could be important, but they are.

Radiusing the root of the external thread adds slightly to a fastener's static tensile strength. The reason is one of simple geometry. As root radius increases, minor diameter increases and the cross sectional area through the thread grows. However, the amount of area increase is so small that it is ignored and in stress computations the same tensile stress area is used for all thread forms.

The prime purpose of root radiusing is to enhance a fastener's fatigue-resistant behavior.

It is a rare assembly joined by mechanical fasteners that isn't exposed in some degree to dynamic loading during its service life. Extremely few remain static and totally insulated from some form of fluctuating stress, vibration, stress reversal, impact, or shock. Fortunately, in only a small percentage of joints are the fatigue properties of the fastener itself the primary design consideration. But, when they are, no opportunity for improvement can be ignored. It is here that root radiusing really counts. The larger the root radius, the better the fatigue properties of the fastener.

Fatigue failures of stressed parts generally occur at locations of high stress concentration — such as notches or abrupt changes in cross sectional configuration. Screw threads, with their variations in cross section and with thread roots acting as notches, are particularly susceptible. The highest stress concentrations in threads occur at their roots. The magnitude of the stress concentration factor relates directly to whether the root is radiused, and if so, to what degree.

Computing thread stress concentration factors is an extremely complex exercise. The answers are not always reliable. Consequently, physical research programs, including photoelastic studies, have been conducted to investigate the fatigue-behavior influence of thread root radiusing. A generalized conclusion is that when all other variables — such as, fastener size, thread pitch, material, manufacturing methods, etc. — are uniform, with the only variable being root radius, stress concentration factors can be reduced from about 6 for sharp or flat UN threads to less than 3 for UNJ threads. This means a possible doubling of fatigue life due to root radiusing. Research has further demonstrated this potential applies to all fastener strength levels.

Roots of internal threads are not normally rounded. Mandatory root radiusing would mean taps with rounded crests. The additional expense would not be cost effective. Fortunately, in a properly designed bolt/nut combination, the nut is the stronger member of the team with the intent that if failure should occur it will always be the externally threaded member. Refer to the bolt/nut compatability discussion, page B-26, for fuller information on this point.

THREAD SERIES

Thread series are groups of diameter/pitch combinations that differ by the number of threads per inch applied to a series of diameters. In the Unified screw thread system there are 11 standard thread series. Just 3 have any importance to mechanical fasteners — coarse (UNC), fine (UNF), and 8 thread (8-UN).

Today's Unified coarse thread series is patterned on the thread series introduced by Whitworth in the mid 19th century. The relatively coarse pitches he selected were probably chosen as much in recognition of the limited manufacturing skills at that time than for any other reason. Over the years, as production capabilities improved, it became economically feasible to produce threads to greater degrees of accuracy and with finer pitches. Many special purpose threads were developed and the one now known as the Unified fine thread series enjoyed considerable popularity. As screw thread technology advanced, it became evident that just two thread series — coarse and fine — were inadequate to efficiently satisfy all engineering applications. Consequently, a number of constant pitch series were added to the system — 8-UN is one of them. Such series have a single thread pitch which is common for all diameters in the series.

Fine vs. Coarse Threads

Few aspects of fastener engineering have been debated more vigorously than the merits of fine vs. coarse threads.

Proponents of fine threads point out —

-They are stronger in tension because of their larger tensile stress areas.

-Their larger minor diameters develop higher torsional and transverse shear strengths.

-Fine threads tap better into thin-walled members and more easily into hard materials.

-Because of their smaller helix angle, they permit closer adjustment accuracy, they require less torque to develop equivalent bolt preloads, and they offer more resistance to loosening when subjected to vibration.

In this latter argument, the first point (adjustment accuracy) is valid. The other two points (less torque and resistance to loosening) are not cogent because of the very small differentials between coarse thread performance coupled with other factors having much greater influence.

Advocates of coarse threads cite these advantages —

-Stripping strengths of coarse threads, particularly the internal thread, are generally greater over the same length of engagement.

-Because stress concentration factors at thread roots decrease as thread pitch increases, coarse thread products should exhibit a better fatigue resistance behavior.

-They are more tolerant to abuse during handling and shipping.

-They have less tendency to cross thread. They assemble and disassemble quicker and easier.

-They are more protective against deleterious loss of thread overlap due to nut dilation under load, and strength loss due to corrosion.

-Larger thread allowances in their class 2A/2B fit allows thicker coatings and platings before thread adjustments need be made.

-Coarse threads tap better into brittle materials that have a tendency to crumble or spall.

The debate has continued for years with no overwhelming support being generated for either series — a reasonably good indicator that merits and deficiencies are shared equally. The once noticeable trend during the past 20 years, however, has been a gradual shifting in popularity toward coarse threads. It is suspected the primary motivation has been the favorable economics of simplifica-

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tion rather than for any reason of technical superiority.

In North America, fine thread fasteners are virtually nonexistent in sizes smaller than No. 0 (0.190 in.) and larger than 1 in. Coarse thread fasteners are popular in the full range of sizes from the smallest to the largest. And, 8 thread series shares an equal popularity to coarse threads for fasteners larger than 1 in.

CLASSES OF THREAD FIT

Thread fit is a measure of looseness or tightness between mating threads. Classes of fit are specific combinations of allowances and tolerances applied to external and internal threads.

For Unified inch screw threads there are 3 thread classes for external threads — 1A, 2A and 3A, and 3 for internal threads — 1B, 2B and 3B. All are clearance fits, which means they assemble without interference. The higher the class number, the tighter the fit. The designator 'A' denotes an external thread, 'B' denotes an internal thread. The mating of class 1A and 1B threads provides the loosest fit, the mating of class 3A with 3B the tightest.

Additionally, there is a class 5 thread fit. Class 5 is an interference fit, which means that the external and internal threads are so toleranced that a positive interference occurs when they are mated. Class 5 interference fits are standard only for coarse thread series in sizes 1 in. and smaller. Refer to ANSI/ASME B1.12, page A-75.

Classes 1A/1B

Classes 1A and 1B are very loosely toleranced threads, with an allowance applied to the external thread. These classes are ideally suited when quick and easy assembly — and, disassembly — are a prime design consideration. They are standard only for coarse and fine thread series in sizes ¼ in. and larger. These classes are rarely specified for mechanical fasteners. In fact, it is doubtful if more than one-tenth of one percent of all fasteners produced in North America have this class of thread fit.

Classes 2A/2B

Classes 2A and 2B are by far the most popular thread classes specified for inch series mechanical fasteners. Close to 90 percent of all commercial and industrial fasteners produced in North America have this class of thread fit. Class 2A external threads have an allowance, class 2B internal threads do not. Classes 2A and 2B, for most engineering applications, offer the optimum thread fit that balances fastener performance, manufacturing convenience and economy.

Classes 3A/3B

Classes 3A and 3B threads are suited for closely toleranced fasteners such as socket cap and set screws, aerospace bolts and nuts, connecting rod bolts, and other high strength fasteners intended for service in applications where safety is a critical design consideration. Classes 3A and 3B have restrictive tolerances and no allowance.

Some Additional Thoughts on Thread Fits

One of the misconceptions about mating threads is the belief that the tighter their tolerances and the closer their fit, the higher the quality of the assembly and the better its service performance. However, like an optical illusion, what appears to be an obvious truth, is frequently false. Designers giving selection priority to closer fit threads may unsuspectingly create assembly problems and add unnecessarily to costs. And, for a number of reasons.

Fig. 1 illustrates the tolerance and allowance relationships for ½-13 UNC threads. For external threads, classes 1A and 2A have an allowance, class 3A does not. Class 1A tolerances are 50 percent larger than those of class 2A, class 3A just 75 percent. For internal threads, none of the 3 classes has an allowance. Class 2B tolerances are 30 percent greater than those of class 2A. Class 1B are 50 percent larger than those of class 2B, class 3B 75 percent.

Strengths

Strengths of mating threads are dependent on having adequate depths and lengths

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of thread engagement — depth being the distance of overlap in a transverse direction, length being the number of threads in longitudinal contact.

Viewing the different classes of thread fit, it would appear that the load supporting ability of the closer toleranced class 3A/3B fit would be stronger. But this is not necessarily true. Actually, no tensile strength difference can be assumed between the loosest and tightest fits. The reason is that a large portion of the permitted tolerance zone for each class is common. There is no assurance that the threads of one class, as actually manufactured, contain more material than those of another.

In the early 1940s, working as a consultant to IFI, Prof. E. A. Buckingham, Massachusetts Institute of Technology, conducted a series of tensile strength tests of fasteners of different sizes and materials. The isolated variable was thread fit. He concluded,

> "As far as these tensile tests show, class 1 is as good as class 3 fit. With bolts of ductile materials, nothing can be gained by using tolerances closer than those of class 1."

Of course, at that time, Prof. Buckingham was studying fasteners threaded with the now obsolete American National thread form. However, their classes of thread fit parallel those of the Unified system. He then added,

> "This witness knows of no definite test data of any kind which supports the notion that tight thread fit on diameter and small diameter tolerances give a stronger bolt and nut assembly."

And he continued,

> "Sometimes the looser fit shows a higher elastic limit load than does the tighter fit. This might be because the additional clearance of the looser fit permits the two members to position themselves in relation to each other so that the load is more uniformly distributed. It is surprising at times, when nature is given a chance to have its own way, how mechanical parts will position themselves to relieve excessive stresses."

That was over 40 years ago. Other studies have been completed since. All are supportive

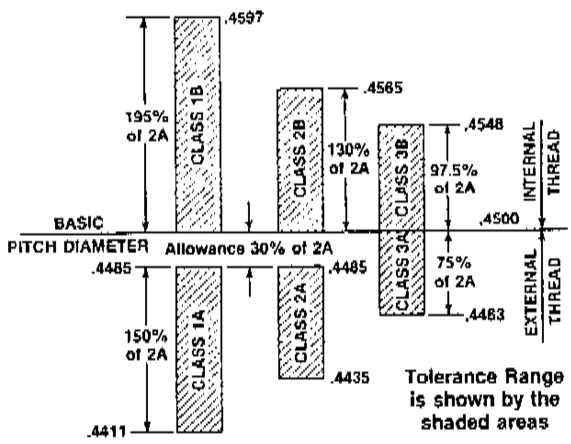

FIG. 1 RELATIONSHIP OF PITCH DIAMETER ALLOWANCES AND TOLERANCES FOR CLASSES OF FIT ON ½-13 UNC THREAD

of Buckingham's findings, none seriously challenges the validity of his observations.

While different thread fits may not exhibit any accountable differences in their tensile strengths, there is a difference in their ability to resist thread stripping.

When an external thread strips off, it generally strips at the cylindrical plane generated by the minor diameter of the internal thread. Similarly, the failure location when an internal thread strips out is at the major diameter of the external thread.

Thread shear areas are computed using minimum material conditions. Consequently, the no allowance and closer tolerances of class 3A/3B threads means more material to resist stripping. The difference between class 3A/3B and class 2A/2B threads can be significant. For example, ½-13 UNC 2A has a thread shear area of 0.779 sq. in. per inch of engaged length; ½-13 UNC 3A has 0.854 sq. in., an increase of nearly 10 percent over class 2A. For ½-13 UNC internal threads, class 2B has a shear area of 1.12 sq. in. per inch of engaged length; class 3B has 1.16 sq. in., an increase of 3.6 percent. This pattern is typical through the full range of fastener sizes, both coarse and fine threads. (See Table 1.)

While the possibility of thread stripping as a failure mode should be avoided, there may be isolated applications when the improvement in stripping strengths available from class 3A/3B threads could be important.

SCREW THREADS

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Table 1 Thread Stress Areas — Unified Inch Screw Threads

Nominal Size, Threads per Inch and Thread Series	A_s Tensile Stress Area sq. in.	A_r Thread Root Area sq. in.	AS_s Thread Stripping Areas sq. in. per in. of engagement, External Thread, Class 2A	AS_s External Thread, Class 3A	AS_n Internal Thread, Class 2B	AS_n Internal Thread, Class 3B
0 - 80 UNF	0.00180	0.00151	0.0673	0.0740	0.106	0.116
1 - 64 UNC	0.00263	0.00218	0.0835	0.0913	0.133	0.144
1 - 72 UNF	0.00278	0.00237	0.0831	0.0922	0.130	0.142
2 - 56 UNC	0.00370	0.00310	0.101	0.109	0.162	0.174
2 - 64 UNF	0.00394	0.00339	0.101	0.110	0.156	0.170
3 - 48 UNC	0.00487	0.00406	0.118	0.127	0.191	0.204
3 - 56 UNF	0.00523	0.00451	0.118	0.130	0.186	0.201
4 - 40 UNC	0.00604	0.00496	0.138	0.147	0.221	0.235
4 - 48 UNF	0.00661	0.00566	0.140	0.151	0.216	0.232
5 - 40 UNC	0.00796	0.00672	0.161	0.172	0.248	0.263
5 - 44 UNF	0.00830	0.00716	0.162	0.173	0.246	0.262
6 - 32 UNC	0.00909	0.00745	0.180	0.189	0.281	0.296
6 - 40 UNF	0.01015	0.00874	0.182	0.197	0.274	0.292
8 - 32 UNC	0.0140	0.0120	0.226	0.239	0.334	0.353
8 - 36 UNF	0.0147	0.0128	0.227	0.244	0.331	0.350
10 - 24 UNC	0.0175	0.0145	0.263	0.277	0.401	0.420
10 - 32 UNF	0.0200	0.0175	0.275	0.289	0.389	0.411
12 - 24 UNC	0.0242	0.0206	0.312	0.327	0.458	0.478
12 - 28 UNF	0.0258	0.0226	0.317	0.336	0.450	0.474
1/4 - 20 UNC	0.0318	0.0269	0.368	0.385	0.539	0.563
1/4 - 28 UNF	0.0364	0.0326	0.373	0.403	0.521	0.549
5/16 - 18 UNC	0.0524	0.0454	0.470	0.502	0.682	0.710
5/16 - 24 UNF	0.0580	0.0524	0.479	0.520	0.663	0.696
3/8 - 16 UNC	0.0775	0.0678	0.576	0.619	0.828	0.860
3/8 - 24 UNF	0.0878	0.0809	0.578	0.644	0.800	0.837
7/16 - 14 UNC	0.106	0.0933	0.677	0.734	0.981	1.01
7/16 - 20 UNF	0.119	0.109	0.685	0.761	0.908	0.991
1/2 - 13 UNC	0.142	0.126	0.779	0.854	1.12	1.16
1/2 - 20 UNF	0.160	0.149	0.799	0.887	1.08	1.13
9/16 - 12 UNC	0.182	0.162	0.893	0.974	1.27	1.32
9/16 - 18 UNF	0.203	0.189	0.901	1.02	1.23	1.29
5/8 - 11 UNC	0.226	0.202	0.998	1.09	1.42	1.47
5/8 - 18 UNF	0.256	0.240	0.998	1.13	1.37	1.43
3/4 - 10 UNC	0.334	0.302	1.21	1.34	1.72	1.78
3/4 - 16 UNF	0.373	0.351	1.23	1.38	1.66	1.73
7/8 - 9 UNC	0.462	0.419	1.43	1.58	2.03	2.09
7/8 - 14 UNF	0.509	0.480	1.44	1.63	1.96	2.03
1 - 8 UNC	0.606	0.551	1.66	1.82	2.33	2.40
1 - 12 UNF	0.663	0.625	1.66	1.87	2.27	2.35
1 - 14 UNS	0.680	0.646	1.67	1.89	2.23	2.33
1-1/8 - 7 UNC	0.763	0.693	1.88	2.04	2.65	2.72
1-1/8 - 8 UN	0.790	0.728	1.89	2.07	2.63	2.70
1-1/4 - 7 UNC	0.969	0.890	2.11	2.30	2.94	3.02
1-1/4 - 8 UN	1.000	0.929	2.12	2.33	2.92	3.00
1-3/8 - 6 UNC	1.16	1.05	2.34	2.52	3.27	3.35
1-3/8 - 8 UN	1.23	1.16	2.34	2.58	3.21	3.30
1-1/2 - 6 UNC	1.41	1.29	2.58	2.77	3.57	3.65
1-1/2 - 8 UN	1.49	1.41	2.57	2.84	3.50	3.61
1-5/8 - 8 UN	1.78	1.68	2.80	3.10	3.79	3.91
1-3/4 - 5 UNC	1.90	1.74	3.04	3.24	4.20	4.30
1-3/4 - 8 UN	2.08	1.98	3.03	3.35	4.08	4.21
1-7/8 - 8 UN	2.41	2.30	3.25	3.63	4.37	4.50
See Note — 1	2	3	4, 6		5, 6	

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Table 1 Thread Stress Areas — Unified Inch Screw Threads

Nominal Size, Threads per Inch and Thread Series	A_s Tensile Stress Area sq. in.	A_r Thread Root Area sq. in.	AS_s Thread Stripping Areas sq. in. per in. of engagement, External Thread Class 2A	AS_s External Thread Class 3A	AS_n Internal Thread Class 2B	AS_n Internal Thread Class 3B
2 - 4-1/2 UNC	2.50	2.30	3.53	3.72	4.83	4.93
2 - 8 UN	2.77	2.65	3.48	3.86	4.66	4.81
2-1/4 - 4-1/2 UNC	3.25	3.02	4.02	4.23	5.44	5.55
2-1/4 - 8 UN	3.56	3.42	3.93	4.37	5.24	5.40
2-1/2 - 4 UNC	4.00	3.72	4.50	4.70	6.07	6.20
2-1/2 - 8 UN	4.44	4.29	4.38	4.87	5.81	6.00
2-3/4 - 4 UNC	4.93	4.62	4.99	5.22	6.68	6.82
2-3/4 - 8 UN	5.43	5.26	4.83	5.38	6.39	6.60
3 - 4 UNC	5.97	5.62	5.48	5.74	7.29	7.44
3 - 8 UN	6.51	6.32	5.28	5.89	6.95	7.20
3-1/4 - 4 UNC	7.10	6.72	5.97	6.26	7.90	8.06
3-1/4 - 8 UN	7.69	7.49	5.73	6.40	7.53	7.79
3-1/2 - 4 UNC	8.33	7.92	6.47	6.77	8.51	8.68
3-1/2 - 8 UN	8.96	8.75	6.18	6.90	8.10	8.39
3-3/4 - 4 UNC	9.66	9.21	6.95	7.29	9.11	9.31
3-3/4 - 8 UN	10.34	10.11	6.61	7.41	8.67	8.98
4 - 4 UNC	11.08	10.61	7.44	7.81	9.71	9.92
4 - 8 UN	11.81	11.57	7.07	7.91	9.24	9.57
See Note — 1	2	3	4, 6		5, 6	

NOTES:

1. In the Unified screw thread system, 1-12 UNF is the standard for inch series fine threads. 1-14 UNS is considered a special diameter/pitch combination. However, the preponderance of all 1 in. fine thread products manufactured in North America are threaded 1-14 UNS.

2. $A_s = 0.7854\left(D - \frac{0.9743}{n}\right)^2$

   where A_s = tensile stress area, sq. in.
   D = nominal thread diameter (basic major dia.), in.
   n = threads per inch

3. $A_r = 0.7854\left(D - \frac{1.3}{n}\right)^2$

   where A_r = area at minor diameter of external thread, sq. in.
   D = nominal thread diameter (basic major dia.), in.
   n = threads per inch

4. $AS_s = 3.1416 \cdot Le \cdot Kn\ max \cdot n \left[\frac{1}{2n} + 0.57735\,(Es\ min - Kn\ max)\right]$

   where AS_s = thread stripping area of external thread, sq. in.
   Le = length of thread engagement, in.
   n = threads per inch
   Kn max = maximum minor diameter of internal thread, in.
   Es min = minimum pitch diameter of external thread, in.

5. $AS_n = 3.1416 \cdot Le \cdot Ds\ min \cdot n \left[\frac{1}{2n} + 0.57735\,(Ds\ min - En\ max)\right]$

   where AS_n = thread stripping area of internal thread, sq. in.
   Le = length of thread engagement, in.
   n = threads per inch
   Ds min = minimum major diameter of external thread, in.
   En max = maximum pitch diameter of internal thread, in.

6. For values of Kn, Es, Ds, and En refer to Table 4 of ANSI/ASME B1.1, page A-38.

BASIC ELEMENTS OF SCREW THREAD DESIGN

Platings and Coatings

A large percentage of all fasteners are plated or coated to give them corrosion protection or to enhance their aesthetic appearance. Platings and coatings have thickness and add to the size of the part.

The accommodation of platings and coatings on threaded fasteners is discussed in great detail in a later section. Suffice it to say here that the allowance on class 2A external threads — which normally is permitted to be consumed to accommodate a plating or coating — is usually adequate to accept commercial thicknesses of platings and coatings without special processing of the thread. Because class 3A threads have no allowance, there is a distinct possibility that threads after plating may not assemble. Consequently, preliminary attention must be given during thread manufacture to assure later assemblability of plated or coated class 3A/3B fasteners.

Elevated Temperatures

For fasteners exposed to elevated temperatures, generally above 500°F, it is desirable to provide a positive allowance between mating threads to furnish a space for lubrication to minimize thread galling and seizing. This is particularly important in any application requiring occasional fastener removal and reassembly.

High cycle wrenching, such as occurs on vehicle assembly lines, creates heat during fastener assembly because of friction between the mating threads. The closer the fit, the greater the friction.

Handling and Shipping

Most fasteners experience considerable abuse during handling prior to their actual use. External threads are easily nicked or otherwise surface damaged. While strength properties remain unaffected, their ability to freely assemble may be impaired. Threads with an allowance can absorb more punishment than those without.

Ductility

Fasteners of low and medium strength materials having good ductility are advantaged by having class 2A threads. The reason is that the allowance, coupled with their more liberal tolerances, provide breathing room between the mating threads to accommodate local yielding, thread bending and other elastic deformations. When stressed, the threads adjust to each other and the load is distributed more uniformly.

Conversely, fasteners of very high strengths and having low ductility should have close fitting threads. When the service load is applied, the close fit contains the thread and minimizes the degree to which it is able to deform. This is a principal reason why aerospace fasteners have class 3A/3B threads.

Vibration

It might appear that closer fitting threads could survive vibration without loosening better than those with an allowance and more generous tolerances. All else being equal, this is probably true. However, there are many other more reliable and less costly techniques for preventing loosening than dependency on thread fit.

Cost

Suffice it to say, the closer the tolerancing, the higher the cost.

PLATED AND COATED THREADS

Platings and coatings added to screw threads increase their size. When the plating or coating thickness becomes excessive, interference between mating threads occurs, and some adjustment in thread sizes prior to their plating or coating must be made to provide assemblability.

In North America, certain principles relating to plated and coated screw threads are generally recognized —

1. Except for heavy platings and coatings — for example, hot-dip galvanizing and mechanically deposited zinc coatings — external and internal threads, following plating or coating, must not transgress their basic profile.

2. Unless the fastener purchaser specifies in advance to the contrary, the allowance

BASIC ELEMENTS OF SCREW THREAD DESIGN

specified for class 2A external threads may be used to accommodate platings and coatings. This means that, following plating or coating, the maximum diameters of the thread are basic, i.e. the same as those specified for class 3A.

3. In certain applications it may be desirable to retain the class 2A allowance after plating or coating. In such instances, the thread class symbol is modified by adding the letter 'G'. Class 2AG threads are plated or coated threads with the same limits of size specified for non-plated or non-coated class 2A threads.

4. The allowance for class 1A external threads must be retained following plating or coating. Thus, the limits of size for class 1A threads are the same whether the fastener is plated, coated, or not.

5. Class 3A external threads and all classes of internal threads have no allowance.

This means that the maximum diameters of classes 1A, 2AG and 3A external threads, and the minimum diameters of classes 1B, 2B and 3B internal threads must be adjusted before plating or coating to provide the necessary space to accommodate the plating or coating thickness.

6. When the thickness of plating or coating is greater than can be conveniently accommodated within the allowance and/or "modestly" adjusted tolerances, the preferred practice is to overtap (increase) the internal thread rather than undercut (reduce) the external thread. This means the after plating or coating maximum material condition of the external thread is permitted to transgress beyond its basic profile.

On 60° threads, the effect of adding a uniform plating or coating on the thread flanks is to increase the pitch diameter of the external thread by four times the thickness of the plating or coating. On an internal thread, its pitch diameter is decreased by a like amount. (See Fig. 7, page A-49.)

Commercial plating and coating thicknesses are most commonly specified as a nominal or minimum value, rarely as a maximum. Also, most processes do not apply platings and coatings uniformly. Consequently, it is customarily assumed that the dimensional effect of plating or coating on a thread's pitch diameter is six times the specified nominal or minimum plating or coating thickness. With this margin, any plating or coating having a specified nominal or minimum thickness equal to or less than one-sixth the thread allowance can be accommodated on a class 2A thread without requiring it to be manufactured to adjusted size limits.

For example, commercial electrodeposited zinc plating has a nominal thickness of 0.00015 in. Therefore, any class 2A external thread with a specified allowance of 0.0009 in. or greater (8-32 UNC, 10-32 UNF and all larger sizes) can accept this plating thickness.

Conversely, dividing the specified thread allowance by six indicates the nominal thickness of plating or coating that can be accommodated without need for special thread manufacturing attention.

When plating or coating thicknesses on class 2A threads exceed those which can be accommodated within the available allowance, and for all other classes of thread, external and internal, the before plating or coating limits of size need adjustment. Detailed recommendations are given in Section 7 of ANSI/ASME B1.1, page A-48.

When inspecting for "after plating or coating" dimensional acceptability, all classes of threads, both external and internal, are gaged using the same gages and gaging requirements specified for the non-plated or non-coated threads of the same class — with two exceptions. The first is class 2A external threads. After plating or coating, class 2A threads are subject to acceptance using a basic class 3A GO gage and a class 2A NOT GO gage. The other exception is for threads with very heavy platings or coatings.

As mentioned earlier, when accommodating heavy thickness platings and coatings, North American practice is to overtap the internal thread rather than undercut the external. There are two good reasons.

Both overtapping and undercutting reduce the strength of mated threads. However, because the nut is usually designed to be the stronger member of a bolt/nut combination, it is better able to accept some strength loss without sacrificing the strength of the assembly. When the bolt thread is undercut, the mated strength of the combination is irreversibly compromised.

BASIC ELEMENTS OF SCREW THREAD DESIGN

The second reason is one of pure economics. Most hot-dip or mechanically galvanized fasteners are taken from a finished goods inventory of non-plated or non-coated products. Seldom are bolts or nuts manufactured with the intent a heavy plating or coating will be subsequently applied. Quantities rarely justify such special manufacture and customer specified thicknesses of platings and coatings vary. Stock bolts are generally threaded with class 2A tolerances, stock nuts are either left blank (to be tapped later) or are class 2B. It is considerably less expensive to plate or coat stock bolts and tap or retap stock nuts to fit the plated or coated bolts than to attempt to recut the bolt threads to reduce their size.

Guidance on the accommodation of heavy platings and coatings on fastener threads, and their inspection for dimensional acceptability is presented in ASTM A563, page B-108, for internal threads, and ASTM A307, page B-56, A449, page B-63, and A354, page B-68, for external threads.

THREAD ACCEPTABILITY

Screw threads have two functions. They must assemble and, once together, they must support a load. A thread's ability to assemble depends exclusively on its dimensional characteristics. Its strength capabilities are a combination of its dimensions and the mechanical properties of the fastener material. Obviously, an oversize thread cannot be assembled. Equally apparent is that an undersize thread may not support its intended load.

Screw thread acceptability is the determination that dimensional conformance is satisfied. It is done by inspecting the thread with gages and other measuring devices, of which there is a great number, ranging in sophistication from those which examine the entire thread as a single attribute to those that measure with exceptional accuracy each individual element.

Identifying the extent of needed inspection to establish a thread's acceptability is the responsibility of the purchaser. And for the simple reason that he (engineer, designer, etc.) is the one most familiar with the performance the fastener must deliver when installed and functioning in its service application. He knows how the fastener will be stressed, the environmental exposure, the magnitude of the applied loads, the safety criticality, and how the fastener interrelates with the other components in the total design. This is the information which dictates the level and intensity of inspection needed to assure that fastener threads are within their specified limits of size.

To assist in this decision, standard gaging systems have been structured and are presented in ANSI/ASME B1.3, page A-53. Each gaging system comprises a listing of those thread features which must be examined, and it details the gages and other measuring devices having the capability to establish whether a particular feature is within its permitted limits.

B1.3 carefully avoids naming which gage or device should be used, even for referee purposes. The choice is the option of each producer and purchaser. The standard merely states that if the characteristic being examined is found acceptable when any of the permitted gages or devices is used, then that characteristic is acceptable.

In B1.3 there are 3 gaging systems for external threads, and 3 for internal threads. The 3 systems are designated Systems 21, 22 and 23. The difference between them is the level of inspection considered necessary to satisfy that dimensional conformance has been achieved. The higher the system number, the more demanding its requirements.

All 3 systems, for both external and internal threads, require functional size examination of the maximum material condition to assure assemblability. The principal difference in their other requirements relates to inspection of the minimum material condition. System 21 examines using functional gaging, i.e. all elements are inspected as a single attribute with the gage or gaging device saying yes or no. Systems 22 and 23 require actual inspection of individual thread elements with System 23 being more inclusive and, consequently, more demanding.

System 21 is considered adequate for most low and medium strength externally threaded fasteners intended for use in general engineering applications. System 22 is frequently specified for high strength exter-

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nally threaded fasteners and those with class 3A threads, where the severity of the application indicates a need for closer inspection of the threads. System 23 is suited primarily for fasteners with UNJ threads.

System 21 is suitable for practically all internally threaded fasteners, except those with UNJ threads which should be inspected using System 23. While System 22 for internal threads is included in B1.3, rarely would there be a need for its use. When the application is so demanding that additional inspection of the internal thread is deemed necessary, the designer should give thought to specifying System 23.

Most fastener product standards now specify which of the gaging systems is considered appropriate for that particular product. In the absence of alternative instructions from the purchaser, that system becomes operational. However, the purchaser always retains the prerogative, and properly so, to specify a different system or to modify the one specified in any way he feels best to provide him the required level of confidence that dimensional conformance has been attained.

THREAD STRENGTH FEATURES

Assembled threads can fail in six ways —

-the externally threaded fastener fractures in tension,

-the external thread strips off,

-the internal thread strips out,

-the externally threaded fastener shears perpendicular to its axis,

-the externally threaded fastener is twisted off, and

-the. internally threaded fastener splits axially through its threaded section.

Tensile failures, thread stripping, and axial splitting of nuts can occur either during assembly or later in service. Torsional failures occur when the fastener is overtightened during its installation. Shear failures happen only after the service loads are applied.

COMPUTING FASTENER STRENGTHS

The four critical stress areas which give mated threads their load carrying capabilities are —

a) the tensile stress area which is the effective cross sectional area through the threaded section and which resists bolt fracture in tension.

b) the thread root area which is the cross sectional area through the threaded section at its minor diameter. It is this area which is used in computing the bolt's resistance to transverse shearing and also its resistance to being twisted off during tightening.

c) the shear area of the external thread which resists the stripping off of the bolt thread, and

d) the shear area of the internal thread which resists the stripping out of the nut thread.

Table 1 gives stress area values for Unified coarse, fine and 8 thread series. Also in the footnotes to Table 1 are the formulas for these areas.

BOLT FRACTURE IS PREFERRED FAILURE MODE

In selecting fasteners, designers should strive to assure that if a failure should occur — from overtightening during installation or overloading in service — that it be bolt fracture and not thread stripping. This is an extremely important point because recent trends have been to tighten bolts to high levels of preload — frequently and intentionally to or beyond their yield strengths. If, during tightening, the bolt breaks, it's sudden, it's visible, replacement is easy and the operator is alerted that some correction to the installation procedure is needed.

On the other hand, thread stripping is an insidious type of failure. It starts at the first stressed thread and gradually, the remaining threads peel off through the entire length of engagement. It's a progressive failure, frequently taking several hours before the nut completely disengages from the bolt. A thread stripping failure can be initiated unsuspectingly. Visually the assembly appears satisfactory, there is no warning that the tightening practice needs correction.

PREVENTING THREAD STRIPPING

The keys to preventing thread stripping are to provide sufficient length of thread en-

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gagement, to minimize nut dilation, and to select mating fasteners with appropriately related strength properties.

Length of Thread Engagement

A seemingly obvious way to increase the resistance of mated threads to stripping is to lengthen their engagement. It definitely helps. However, there is a limit beyond which lengthening the thread engagement adds nothing but cost.

When a bolt/nut assembly is axially loaded, the bolt is stressed in tension, the nut in compression. With rigid materials and threads of perfect form, the load would be distributed uniformly between the mated threads with each taking one full and equal share. However, fasteners are made of materials having elastic properties. And, it would be the remotest of coincidences if both the internal and external threads had perfect form.

An axially applied load stretches the bolt. This lengthens the effective lead of its thread. Simultaneously, the nut compresses and its thread lead is shortened. These deformations, while discreet, must balance each other both locally and through the entire length of thread contact. This results in a disproportionate distribution of the total load. The first engaged thread assumes a higher than average load, the remaining threads successively lower loads with the last or top thread the lowest. Analytical studies have shown that the load carried by the first thread can exceed twice the average for all threads with the load on the last thread less than one-half. This is the reason that bolt tensile failures usually occur at the first thread within the nut or tapped hole. These same studies further showed that, for the same length of thread engagement, the finer the thread pitch, the higher the average load on the first thread.

Increasing the length of thread engagement much beyond 1.0 times the nominal bolt diameter is self-defeating. The reason is that with this number of engaged threads the portion of the total load carried by the top threads is low and can only increase if the first threads so grossly deform that their excessive share of the load is relieved and transferred to successive threads. At this point, failure is quite probably imminent.

Nut Dilation

As an axially applied load increases, the bolt elongates, the nut compresses, and the nut walls begin to spread out, or dilate, because of the radial wedging action of the contacting threads. This radial force is resisted by hoop stresses in the nut, the lower the strength of the nut material and the thinner the nut wall section, the greater the dilation.

Controlling nut dilation is important because dilation occurs at the nut bearing face which is the location of the most highly stressed thread. If the nut moves radially, the depth of thread engagement is reduced. This in turn decreases the thread shear areas of both the bolt and nut threads and unit shear stresses increase. The finer the thread pitch, the more the situation is aggravated.

Hex nuts with widths across flats at least 1.5 times their nominal thread diameters will generally be found adequate. Those with smaller widths across flats should be viewed cautiously. Flanged nuts are superior in resisting dilation. For tapped holes, the surrounding material is comparatively so massive that dilation as a strength reducing factor may be ignored.

Fastener Material Strengths

When the relative strengths of the bolt and nut materials are approximately the same, as the axial load increases the threads of both bend, nut dilation intensifies, and the effective shear areas of both threads decrease. If a thread stripping failure happens, it is difficult to identify whether the bolt thread or that of the nut failed first.

If the strength of the bolt material is significantly greater than that of the nut material — the majority of all bolt/nut combinations — the bolt threads will not distort as readily and will constrain the nut threads from bending, even though the nut material may have a much lower yield strength. If, under these circumstances, a thread stripping failure occurs, the nut thread will strip out cleanly on the cylindrical plane generated by the major diameter of the bolt thread.

Similarly, when the strength of the nut material exceeds that of the bolt, thread bending and distortion of the bolt threads are constrained by those of the nut. If the assembly fails by thread stripping, the bolt threads will strip off at the cylindrical plane generated by the minor diameter of the nut thread.

Generally, the closer the nut and bolt material strengths are to each other, the lower the thread stripping failure load regardless of which one strips. If there is a strength disparity, the failure load will be slightly higher. This is a reason why nuts tested using hardened mandrels usually exhibit slightly higher failure loads than when tested using bolts of compatible strength properties.

Using the stress area values in Table 1 it is reasonably easy to compute the needed lengths of engagement and minimum strength properties (tensile/shear ratios) of either the externally or internally threaded fastener. The objective in selecting fasteners should be to make certain that the load the threads can support in shear (thread stripping area times length of thread engagement times material shear strength) safely exceeds the load the fastener can support in tension (tensile stress area times material tensile strength).

INFLUENCE OF EXPOSED THREADS IN THE GRIP

The total thickness of material to be joined by a fastener is known as the grip. Following fastener installation, the stressed length of the bolt essentially equals the grip, plus possibly one to two threads into the nut. The number of bolt threads included within the grip has a significant influence on the fastener's ultimate tensile strength — the load at which it fails.

When a bolt/nut combination is tensile tested to failure, if the nut is positioned anywhere along the length of the bolt thread so that at least four complete threads are exposed between the nut bearing face and the bolt thread runout, the tensile strength of the bolt remains unchanged. When the nut is brought closer to the thread runout, the tensile strength of the bolt increases, and could be as much as 20 percent greater when the nut is advanced against the thread runout.

While the bolt's apparent tensile strength increases with fewer threads exposed within the grip, the stripping strengths of the bolt and nut threads decrease. The reason is that bolt yielding — the prelude to fastener failure — is now occurring in the threaded length engaged within the nut. This reduces depth of thread engagement and increases the thread unit shear stresses in both threads. Conceivably, it is possible to change the failure mode from bolt fracture to thread stripping just by reducing the number of exposed bolt threads within the grip.

The standard method for tensile testing externally threaded fasteners (ASTM F606, para. 3.4, page B-147) specifies six complete threads be exposed within the grip. The one exception is high strength structural bolts which, because of their shorter thread lengths, require only four threads to be exposed.

In joint design, when the fastener length is properly compatible with the thickness of material being joined, thread lengths on standard fasteners assure a reasonable length of exposed threads within the grip. For other circumstances, prudent design suggests an absolute minimum of one complete thread — preferably, considerably more — be left within the grip.

DESIGNING SPECIAL THREADS

For maximum economy in fastener selection, first consideration should always be given standard size products with Unified screw threads of coarse, fine, or 8-thread series. This approach is basic and holds true for all engineering applications. The reason is quite simple. Standard diameter/pitch Unified screw threads are proven, they're completely standardized dimensionally, manufacturing tooling and thread gaging equipment are abundant commercially, and there is a broad choice of manufacturing facilities and suppliers.

Occasionally, however, for valid technical reasons, no standard diameter/pitch combination is suitable and the engineer must design a special thread. Consequently, it may be of some help to trace through an example.

Let's assume that in the design of a machine 13mm - 11 steel bolts must be used to connect one of the component parts to an aluminum base. The tapped holes in the base can be up to 1 inch deep. Because of the severity of the service exposure the bolts must be furnished with an especially heavy electrodeposited zinc coating having a mini-

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mum thickness of 0.0003 in. The threads will be of Unified form and have class 2A/2B tolerances and the external thread allowance must be maintained after plating. The service load fluctuates, with the computed tensile load on the bolt reaching a maximum of 10,000 lbs. Initial bolt tightening will preload the bolt beyond this service load.

Determine the limits of size of the external and internal screw threads and check the strength adequacy of the fastener for the application.

For 13mm - 11 screw threads,

D = 13mm = 0.5118 in.
n = threads per inch = 11
p = thread pitch = 1/n = 0.0910 in.

For the external thread, referring to 8.3, page A-51,

Basic major dia = D = 0.5118 in.
Basic pitch dia = E = D − h_b = 0.5118 − 0.64952 x 0.0910 = 0.4527 in.

And, referring to 5.8.1, page A-37,

Pitch dia tolerance, class 2A = $0.0015\sqrt[3]{D} + 0.0015\sqrt{L_e} + 0.015\sqrt[3]{p^2}$ = 0.0055 in.

(Note: L_e = length of thread engagement, assumed to be 1.5 D = 0.768 in., to be checked later)

Major dia tolerance, class 2A = $0.060\sqrt[3]{p^2}$ = 0.0121 in.
Allowance, class 2A = 0.300 class 2A p.d. tol = 0.0016 in.

Consequently,

Major dia, max = 0.5118 − 0.0016 = 0.5102 in.
Major dia, min = 0.5102 − 0.0121 = 0.4981 in.
Pitch dia, max = 0.4527 − 0.0016 = 0.4511 in.
Pitch dia, min = 0.4511 − 0.0055 = 0.4456 in.

But, these limits must now be adjusted so that the class 2A allowance is retained after plating the external thread with a minimum coating thickness of 0.0003 in.

Referring to 7.5.2, page A-50, these are the "before plating" limits,

Major dia, max = 0.5102 − 3 x 0.0003 = 0.5093 in.
Major dia, min = 0.4981 − 2 x 0.0003 = 0.4975 in.
Pitch dia, max = 0.4511 − 6 x 0.0003 = 0.4493 in.
Pitch dia, min = 0.4456 − 4 x 0.0003 = 0.4444 in.

For the internal thread, referring to 8.3, page A-51,

Basic minor dia = K = D − $2h_n$ = 0.5118 − 2 x 0.54127 x 0.0910 = 0.4133 in.
Basic pitch dia = E = 0.4527 in.

And, referring to 5.8.2, page A-42,

Pitch dia tolerance, class 2B = 1.300 class 2A p.d. tol = 0.0072 in.
Minor dia tolerance, class 2B = 0.25 p − 0.4 p^2 = 0.0194 in.

Consequently,

Minor dia, min = 0.4133 in.
Minor dia, max = 0.4133 + 0.0194 = 0.4327 in.
Pitch dia, min = 0.4527 in.
Pitch dia, max = 0.4527 + 0.0072 = 0.4599 in.

Now that the limits of size of the screw threads are known, fastener strength capabilities can be examined.

Referring to the stress area formulas given in the footnotes to Table 1, page A-9,

Tensile stress area =

$$A_s = 0.7854\left(D - \frac{0.9743}{n}\right)^2$$
$$= 0.141 \text{ sq. in.}$$

Stripping area, ext thd = As_s

$$= 3.1416 \times L_e \times K_n \max \times n\left[\frac{1}{2n} + 0.57735\,(E_s \min - K_n \max)\right]$$

$$= 3.1416 \times L_e \times 0.4327 \times$$

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$$11\left[\frac{1}{2 \times 11} + 0.57735\,(0.4444 - 0.4327)\right]$$

$$= 0.781\ L_e$$

Stripping area, int thd $-$ AS_n

$$= 3.1416 \times L_e \times D_s \text{ min} \times$$

$$n\left[\frac{1}{2n} + 0.57735\,(D_s \text{ min} - E_n \text{ max})\right]$$

$$= 3.1416 \times L_e \times 0.4975 \times$$

$$11\left[\frac{1}{2 \times 11} + 0.57735\,(0.4975 - 0.4599)\right]$$

$$= 1.15\ L_e$$

The bolt, when carrying 10,000 lbs, is stressed 10,000 ÷ 0.141 = 70,000 psi. Referring to Table 2, page B-12, the most suitable steel would be SAE Grade 5 with its specified proof load stress of 85,000 psi and minimum tensile strength of 120,000 psi.

As emphasized earlier in this article, it is important that if a failure should occur through overtightening during installation or overstressing in service, that the failure mode be bolt fracture and not thread stripping. Consequently, the length of thread engagement needed to provide sufficient strength against thread stripping must be calculated. Both external and internal threads must be studied. And, to be completely safe, the load which must be resisted in thread shear is the maximum load the bolt could support in tension not its specified minimum.

The specified max hardness of Grade 5 steel bolts is Rockwell C34. Referring to Table 1, page B-7, a hardness of C34 equates approximately to a tensile strength of 153,000 psi, which means a possible bolt breaking strength of about 21,500 lbs.

The minimum shearing strength of Grade 5 steel may be assumed as about 60 percent of its minimum specified tensile strength, say 70,000 psi. The thread stripping area is 0.781 L_e sq. in. and computing for L_e –

$$L_e = \frac{21,500}{70,000 \times 0.781} = 0.393 \text{ in.}$$

A reasonable minimum shear strength for aluminum is 25,000 psi. The stripping area of the tapped hole is 1.15 L_e, and solving for L_e,

$$L_e = \frac{21,500}{25,000 \times 1.15} = 0.748 \text{ in.}$$

As a length of thread engagement equal to 1.5 D (0.768 in.) was assumed earlier, this length is adequate and fits within the depth of tapped hole which the aluminum base could accommodate.

ENGINEERING STANDARDS

All necessary information, data and requirements for Unified inch screw threads are covered in a package of 5 separate standards.

ANSI/ASME B1.1 "Unified Inch Screw Threads (UN and UNR Thread Form)," page A-26, specifies tables of dimensions, thread form, thread series, tolerances, and designations.

ANSI/ASME B1.15, now under preparation, will provide similar information for UNJ threads.

ANSI/ASME B1.2 "Gages and Gaging for Unified Inch Screw Threads," page A-62, covers gaging devices used for the inspection of Unified threads and includes descriptions, design tolerances, calibration of gages, and correct use.

ANSI/ASME B1.3 "Screw Thread Gaging Systems for Dimensional Acceptability," page A-53, presents screw thread gaging systems suitable for determining the acceptability of Unified inch screw threads on externally and internally threaded fasteners.

ANSI/ASME B1.7 "Nomenclature, Definitions and Letter Symbols for Screw Threads," page A-18, establishes uniform practices particular to screw thread nomenclature and terms relating to types of screw threads, size and fit, geometrical elements, and dimensions.

Copies of these documents are available from The American Society of Mechanical Engineers, United Engineering Center, 345 E. 47th Street, New York, N.Y. 10017.