METRIC SCREW THREADS

IFI NOTE:

At New Delhi, India in 1964, an historic meeting of ISO Technical Committee 1 (TC1) produced final agreements for all tolerances and allowances for a metric screw thread profile and a full range of pitch-diameter combinations. Subsequently, OMFS (Optimum Metric Fastener System) proposals were submitted by North America and a compromise approved by ISO/TC1 at Stockholm in 1976. Following this accord ASME Standards Committee B1 initiated development of a metric screw threads – M profile standard which was first published in 1979, and designated ANSI/ASME B1.13M. This standard was in basic agreement with the Stockholm accord and provided a system of metric screw threads and detailed information for diameter-pitch combinations. Since 1979, a new issue dated 1983 has been published and a second revision dated 1994 is included in this section.

SCREW THREAD CHARACTERISTICS

IFI has prepared an introduction to this section to explain in "layman" terms some concepts which will assist in a fuller understanding of screw threads.

While various forms of threads have existed for almost all of recorded history, it was not until Archimedes that the first known practical application of the screw thread was confirmed. His study developed the means of raising water from one level to another by using the basic screw thread form and helped foster irrigation. Since that time screw threads have evolved into extremely useful forms. To become reasonably thread friendly, a person must become familiar with about 30 basic terms. (Experts probably relate to 125 or more features or characteristics.)

A *screw thread* is a ridge of uniform section in the form of a helix on the external or internal surface of a cylinder. A thread formed on a cylinder is called a straight or parallel thread, while that formed on a cone or frustum of a cone is called a tapered thread. *External threads* are threads on bolts, screws and studs. *Internal threads* are those in nuts or tapped holes.

The thread form or profile in an axial plane is made up of three elements: the *root*, *crest* and *flank*. The *root* is at the bottom of the groove and is joined to the *crest*, which is at the top of the ridge, by the *flank*. The fundamental triangle is that formed when the profile is extended to a sharp V at both crest and root. The height of the *fundamental triangle* (H) is the distance measured radially between the sharp crest and sharp root diameters; see Fig. 1 of B1.13M on page A–21. For 60° metric threads – M profile H equals 0.866 0254 times thread pitch (compiled in Table 3, A–27.) The height of screw threads is controlled by the amount of truncation at the crest and root of the fundamental triangle with the basic M profile thread height at 0.625H or 0.541 266P.

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

METRIC SCREW THREADS

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.*

Almost all of the five major thread forms used today have a 60° included angle; thus the flank angle is 30° for the symmetrical metric thread — M profile.

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, with the major or minor diameter out of tolerance, 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. For metric threads the pitch is expressed in millimeters. This should be noted because for those most familiar with inch practice the pitch is customarily expressed in threads per inch (TPI). The metric coarse diameter/pitch series are such that they are in between the inch coarse and fine pitches which are found in the Unified thread series. While this may be viewed as an "optimum" or ideal choice, it presents another issue. The metric series also includes fine pitches which are considerably finer than the Unified fine series. Because they are "so fine" they are not recommended for use.

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. The pitch line in the thread profile provides a reference location for dimensioning the profile and defining the size of the thread. The thread standards use the pitch diameter of a perfect thread form at the maximum and minimum pitch diameter dimensions to define the thread envelope in space which theoretically defines the thread boundaries. The three dimensional envelope defined by standard maximum and minimum pitch diameter with a perfect thread profile is almost impossible to measure or gage. The nearest approach to evaluating the complex product thread contours and the envelope definition is laser surface measurements. The assurance of maximum material fit is commonly checked with GO Gages which attempt to make a functional diameter check. In theory, the GO Gage represents a perfect thread at the maximum external or minimum internal pitch diameter size. The imperfections in the gage thread contour all make the gage more critical so that product can always assemble if judged satisfactory by the GO Gage. The minimum external and maximum internal envelope are much more difficult to evaluate with a functional check because the profile errors on both product and NOT GO Gages make the product thread appear better. Currently no gaging system can evaluate the thread envelope over the entire thread. Consequently, among screw thread experts, the definition, measurement, and significance of pitch diameter is controversial. However, it is the important reference location and 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* is the cyclical outline, in an axial plane, of the permanently established boundary between the provinces of the external and internal threads. It is from this basic thread profile boundary that all allowances are applied. If the basic thread profile boundary is violated, the threads may not assemble. While ISO has used the term *fundamental deviation,* the ASME B-1 Standards Committee has maintained the common North American terminology of *allowance.* The absolute values are numerically equal.

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. Metric threads are designated by a tolerance grade and a tolerance position. The tolerance grade is a number which defines the dimensional tolerance. The tolerance position is a letter which defines the allowance or size of the maximum material limit. For external threads, a number and a lower case letter in combination are a tolerance class which may be applied to the pitch diameter and the major diameter. When both tolerance classes are the same for the pitch diameter and the major diameter, the tolerance class is only written once. The internal thread tolerance class indicates the tolerance grade and position by a number for grade and capital letter for tolerance position. If they are the same combination for the pitch diameter and minor diameter they are only written once. For external metric threads the tolerance classes 6g and 4g6g provide an allowance such that there is a specified minimum clearance between the maximum material condition of the thread and its basic size. For tolerance class fits 6H/6g and 6H/4g6g the basic size of the external thread coincides with the maximum material condition of the internal thread. Unless otherwise specified, the external thread allowance may be used to accommodate coating or plating thickness, provided that threads after coating or plating do not exceed basic size. *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.* See Fig. 5 of B1.13M, page A–25.

The combination of allowances (fundamental deviation) and tolerances in mating threads is called *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 results 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 factors when computing thread strengths.

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 as the bolt which would support the same ultimate load when the cylinder is 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, in

METRIC SCREW THREADS

torsion, or in fatigue. *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 axial 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.

GUIDE TO THREAD SELECTION

The two basic thread forms used in metric are designated as the M and the MJ. The external M form has a recommended minimum root radius of 0.125 × pitch which is mandatory for Property Class 8.8 and stronger. The external MJ form has a minimum root radius of 0.15011 × pitch. The MJ profile also includes a maximum root radius of 0.18042 × pitch. While this feature may not seem particularly significant for simple axial loading, the root radius has been found to contribute in a very significant way to fatigue performance or resistance. The radiused root should always be carefully considered by the designer when dealing with joints subject to fatigue considerations.

It is a rare assembly joined by mechanical fasteners that isn't exposed to some degree of 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.

THREAD SERIES

Thread series are groups of diameter/pitch combinations that differ by the diameters and/or pitches with which they have been combined. The present metric thread has 13 constant pitch series in addition to the coarse. All threaded mechanical fasteners included in this book have only one series of diameter-pitch combinations. These combinations are selected from the metric coarse thread series given in ASME B1.13M and in ISO 261.

(IFI Note: Standard practice in North America is to use only the metric coarse thread series. Because ISO continues to include fine pitch series products, the following discussion is included:)

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.

METRIC SCREW THREADS

- 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 are available for *e*, *f*, or *g*; which allows thicker coatings and platings before thread adjustments need be made. The tolerance position indicated by letters is the allowance or fundamental deviation. The lower case letters are used for external threads and capitals for internal threads.

  Tolerance position *g* offers a smaller allowance than *e* or *f* and tolerance position *h* offers no allowance at all.
- Coarse threads tap better into brittle materials that have a tendency to crumble.

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 one 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 simplification rather than for any reason of technical superiority. In North America, fine thread fasteners are rarely used and are almost nonexistent below M5 or above M25. In sizes above M25 a constant thread fine pitch series is often used.

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. Only two classes of thread fit, class 6H/6g and class 6H/4g6g, are recognized as standard for the threaded fasteners included in this book.

External threads of tolerance class 6g, combined with internal threads of tolerance class 6H, are suitable for metric applications where inch series Class 2A/2B threads were used.

External threads of tolerance class 4g6g are more closely toleranced, and are approximately equivalent to inch series Class 3A threads, with the exception that class 4g6g has an allowance.

Metric threads of tolerance class 4H5H/4h6h are the closest approximation to inch series 3A/3B Class of fit.

METRIC SCREW THREADS

Formulas for computing the limiting dimensions of all tolerance classes of metric threads are given in ASME B1.13M.

SELECTION OF THREAD FIT

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. Research indicates that sometimes a looser fit helps mating threads which are being loaded to adjust relative to each other such that the load is more uniformly distributed. Nuts, for example, are usually softer than the bolts or screws with which they are mated. This allows some deformation of the softer mating element to establish a resulting "best fit".

In the example shown in Fig. 1 for M12 × 1.75 Class 6e overlaps 6g by 75% and 6h by 52%. Class 6g overlaps 6h by 77%. Thus, within the system of allowances and tolerances a manufactured thread of one tolerance position may actually be the same as a manufactured thread having a different tolerance position.

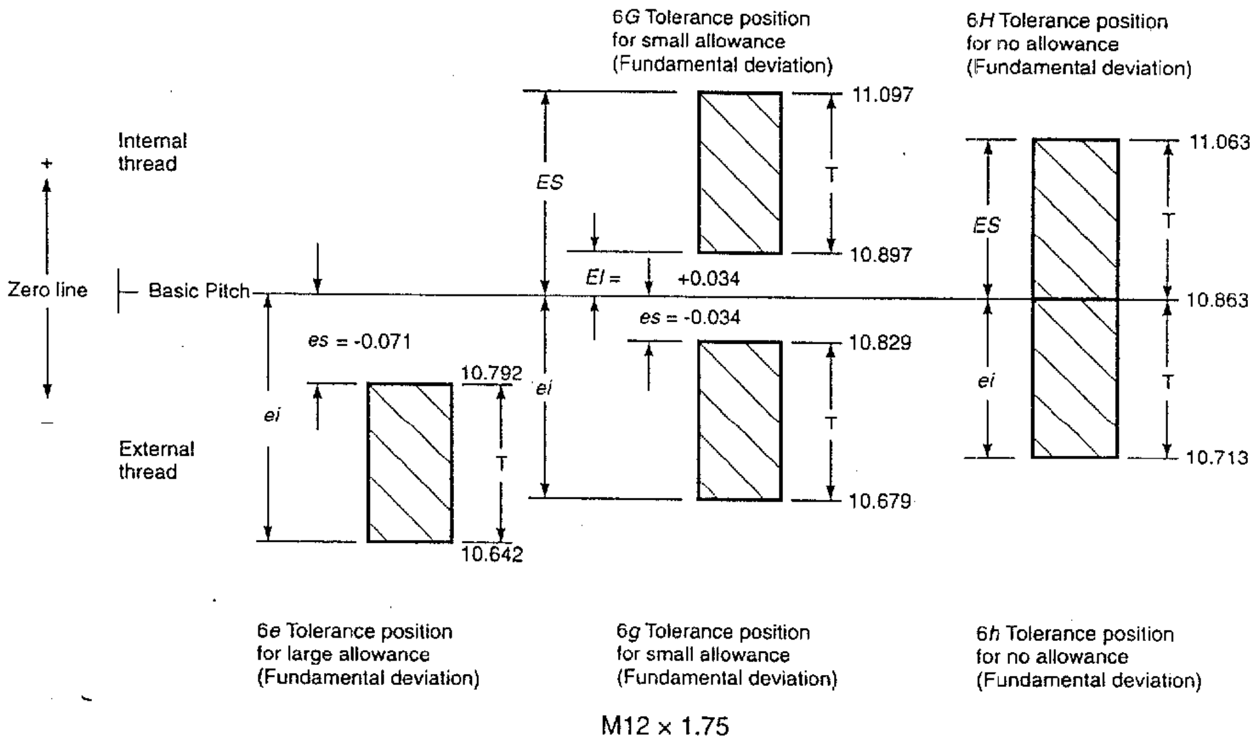

Fig. 1 Metric Tolerance System for Screw Threads

METRIC SCREW THREADS

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 over-tightened during its installation. Shear failures happen only after the service loads are applied.

MATING STRENGTHS

Mating threads have strengths which are a function of depths of engagement (overlap of mating threads) and lengths of engagement (the number of mating threads in contact in the axial direction). Research on the subject of class of fit and tensile strength, however, indicates that a tighter thread fit for ductile materials does not demonstrate a superior tensile strength of the mated bolt and nut assemblies. (See studies in the 40's conducted by Prof. E. A. Buckingham at MIT). Thus, it is not expected that any appreciable gain in tensile strength would be achieved by selection of a 6H/6h combination.

While different thread fits may not exhibit any accountable differences in tensile strengths, there is a difference in their ability to resist thread stripping. Thread shear areas are computed using minimum material conditions. Consequently, a no allowance fit means more material which will resist 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 of an internal thread in the stripping mode will be at the major diameter of the external thread.

DESIGN DATA

There are three stressed areas of mating threads which determine the load carrying capability of the bolt/nut assembly.

A. The tensile stress area is the effective cross sectional area through the thread. The tensile stress area of the external thread is used in all computations of ultimate, yield, and proof load strengths of externally and internally threaded fasteners subjected to direct tensile or compression loads applied through their threads. Tensile stress areas of metric threads are computed using the formula:

$$A_s = 0.7854\,(D - 0.9382P)^2$$

where:

A_s = tensile stress area, mm^2

D = nominal thread diameter (basic major diameter), mm

P = thread pitch, mm

B. The shear area of the external thread, and

C. The shear area of the internal thread.

The thread shear areas resist bolt and nut thread stripping. The shear area of the external thread depends principally on the length of engaged thread and the minor diameter of the internal thread; the shear

METRIC SCREW THREADS

area of the internal thread depends on the length of engaged thread and the major diameter of the external thread. Thread shear areas are computed using the FED-STD-H28/2, "Screw Thread Standards for Federal Services, Section 2," formulas:

$$AS_n = \left(\frac{3.1416}{P}\right)(LE)(d_{min})[0.5P + 0.57735(d_{min} - D_{2\,max})]$$

and

$$AS_s = \left(\frac{3.1416}{P}\right)(LE)(D_{1\,max})[0.5P + 0.57735(d_{2\,min} - D_{1\,max})]$$

where:

AS_n = minimum thread shear area for internal threads, mm^2

AS_s = minimum thread shear area for external threads, mm^2

P = thread pitch, mm

LE = length of thread engagement, mm

d_{min} = minimum major diameter of external thread, mm

$D_{2\,max}$ = maximum pitch diameter of internal thread, mm

$D_{1\,max}$ = maximum minor diameter of internal thread, mm

$d_{2\,min}$ = minimum pitch diameter of external thread, mm

Table 1 lists the tensile stress and shear areas for general purpose screw threads of tolerance class of fit 6H/6g. For tolerance class 6H/4g6g threads, the tensile stress areas and shear stress areas of internal threads are the same, however, the shear stress areas for external threads are slightly greater. For design purposes, the values given in Table 1 are recommended for all tolerance classes of fit.

The shear stress areas in Table 1 are for 1 mm of engaged thread length, i.e., the length of actual mating of complete (full thread) internal and external threads. When computing the length of thread engagement in a bolt/nut combination, the nut thickness must be reduced because of incomplete thread due to its countersink(s). The following empirical formulas may be used:

For M8 nuts and smaller:

LE = nut thickness – C (D + 0.75 – D_1 min)

For M10 nuts and larger:

LE = nut thickness – C (1.08D – D_1 min)

where:

LE = length of thread engagement, mm

D = nominal thread diameter, mm

D_1 = minor diameter of nut thread, mm

nut thickness = actual thickness (if unknown, use minimum thickness as specified in product standard)

C = a constant equal to 0.42 if the nut is countersunk on one face only, and 0.84 if both faces are countersunk

TAP DRILL DIAMETER

According to FED-STD-H28/2, for lengths of thread engagements in the range of 0.67D to 1.5D (the overwhelming majority of all fastener applications), the optimum strength of the bolt/nut or tapped hole combination, coupled with favorable tapping conditions, is achieved when the hole size before tapping has a minimum diameter equal to the minimum minor thread diameter plus one-half

METRIC SCREW THREADS

Table 1 Thread Stress Areas

Basic Major Dia and Thread Pitch	Tensile Stress Area mm²	Thread Shear Area, mm² per mm of Engaged Threads	
	A_s	AS_s	AS_n
M1.6 x 0.35 M2 x 0.4 M2.5 x 0.45	1.27 2.07 3.39	1.87 2.48 3.18	2.65 3.44 4.54
M3 x 0.5 M3.5 x 0.6 M4 x 0.7	5.03 6.78 8.78	3.91 4.67 5.47	5.54 6.60 7.77
M5 x 0.8 M6 x 1 M8 x 1.25	14.2 20.1 36.6	7.08 8.65 12.2	9.99 12.2 16.8
M10 x 1.5 M12 x 1.75 M14 x 2	58.0 84.3 115	15.6 19.0 22.4	21.5 26.1 31.0
M16 x 2 M20 x 2.5 M22 x 2.5	157 245 303	26.1 33.3 37.0	35.6 45.4 50.0
M24 x 3 M27 x 3 M30 x 3.5	353 459 561	40.5 46.2 51.6	55.0 62.0 69.6
M36 x 4 M42 x 4.5 M48 x 5	817 1120 1470	63.1 74.3 85.8	84.1 99.2 114
M56 x 5.5 M64 x 6 M72 x 6	2030 2680 3460	101 117 133	134 154 173
M80 x 6 M90 x 6 M100 x 6	4340 5590 6990	149 169 189	193 217 241

the minor diameter tolerance and a maximum hole size equal to the maximum minor thread diameter.

Table 2 recommends tap drill sizes, as selected from ISO 2306, "Drills for Use Prior to Tapping Screw Threads," which most closely satisfy the H28 design recommendation.

PLATING AND COATING OF THREADS

A large percentage of all fasteners are plated or coated to provide corrosion resistance and/or to enhance their appearance. These platings or coatings all have a thickness which increases the fastener size.

METRIC SCREW THREADS

Table 2 Recommended Tap Drill Diameters for Metric Threads

Basic Major Dia and Thread Pitch	Internal Thread Minor Dia		Tap Drill Dia
	Max	Min	
M1.6 × 0.35	1.321	1.221	1.25
M2 × 0.4	1.679	1.567	1.6
M2.5 × 0.45	2.138	2.013	2.05
M3 × 0.5	2.599	2.459	2.5
M3.5 × 0.6	3.010	2.850	2.9
M4 × 0.7	3.422	3.242	3.3
M5 × 0.8	4.334	4.134	4.2
M6 × 1	5.153	4.917	5.0
M8 × 1.25	6.912	6.647	6.8
M10 × 1.5	8.676	8.376	8.5
M12 × 1.75	10.441	10.106	10.2
M14 × 2	12.210	11.835	12.0
M16 × 2	14.210	13.835	14.0
M20 × 2.5	17.744	17.294	17.5
M22 × 2.5	19.744	19.294	19.5
M24 × 3	21.252	20.752	21.0
M27 × 3	24.252	23.752	24.0
M30 × 3.5	26.771	26.211	26.5
M36 × 4	32.270	31.670	32.0
M42 × 4.5	37.799	37.129	37.5
M48 × 5	43.297	42.587	43.0
M56 × 5.5	50.796	50.046	50.5
M64 × 6	58.305	57.505	58.0
M72 × 6	66.305	65.505	66.0
M80 × 6	74.305	73.505	74.0
M90 × 6	84.305	83.505	84.0
M100 × 6	94.305	93.505	94.0

The accommodation of these platings or coatings is discussed in greater detail in another section of this book. External threads of tolerance classes 6g and 4g6g provide an allowance, i.e., there is a specified minimum clearance between the maximum material condition of the thread and its basic size. For tolerance class fits 6H/6g and 6H/4g6g, the basic size of the external thread coincides with the maximum material condition of the internal thread.

Unless otherwise specified by the purchaser, the external thread allowance may be used to accommodate coating or plating thickness, providing that threads after coating or plating do not exceed their specified basic size. Dimensional acceptance of coated or plated external threads is based on using a basic (tolerance position h) size GO thread gage.

As a guide, the thickness of plating or coating that can be accommodated on an external thread of tolerance class 6g without exceeding its basic size is one-fourth of the specified allowance. For coatings or platings of greater thickness, it is usually necessary to produce the external threads to undersized limits and/or tap the internal threads to oversize limits. Refer to Section 8 of ASME B1.13M beginning on page A–39.

For heavy platings, such as hot-dip or mechanical galvanizing, it is customary practice to galvanize external threads of tolerance class 6g (or 4g6g) and provide assemblability by overtapping the internal thread. Suitable limits for oversize tapping of internal threads to accommodate galvanized external threads are specified in ASTM A563M, "Carbon and Alloy Steel Metric Nuts", page B–68.

When the allowance must be retained on coated or plated external threads, the thread class designation shall be followed by the words, "after plating" or "after coating" to alert the producer that special thread manufacture may be necessary.

ELEVATED TEMPERATURE

For fasteners exposed to elevated temperatures, generally above 260°C, it is useful to provide an allowance between mating threads to allow space for a lubricant to minimize thread galling and seizing. This is particularly important where occasional removal and reassembly is anticipated. Clearly, the closer the fit, the more friction present.

HANDLING AND SHIPPING

Most fasteners are exposed to abuse during handling prior to actual installation. 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 are more forgiving in these situations.

VIBRATION

It may appear that closer fitting threads could survive vibration without loosening better than those with an allowance. 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.

DEFINITIONS

Terms commonly associated with screw threads are defined in ANSI/ASME B1.7M, "Nomenclature, Definitions, and Letter Symbols for Screw Threads and Related Features." The screw thread terms used in this book, as abstracted from ANSI/ASME B1.7M, are as follows:

Allowance. Is the prescribed difference between the design size and the basic size of a thread.

Basic Profile of Thread. The cyclical outline, in an axial plane, of the permanently established boundary between the provinces of the external and internal threads. All deviations are with respect to this boundary.

Basic Size. Is that size from which the limits of size are derived by the application of allowances and tolerances.

Characteristics. Qualities, peculiarities, or features which are conspicuous or prominent details of the thread.

Class of Thread. An alpha numerical designation to indicate the standard grade of tolerance and allowance specified for a thread.

Clearance Fit. A fit between mating assembled parts that provides a clearance at their maximum material condition.

Complete Thread. The thread whose profile lies within the size limits. A complete (full form) thread is that cross section of a threaded length having full form at crest and root.

Crest. Is that surface of the thread which joins the flanks of the thread and is farthest from the cylinder or cone from which the thread projects.

Depth of Thread Engagement. The radial distance, crest-to-crest, by which thread forms overlap between two co-axially assembled mating threads.

Design Size. Is the basic size with allowance applied, from which the limits of size are derived by the application of tolerance. If there is no allowance, the design size is the same as the basic size.

Design Thread Form. The design thread form is the maximum material profile permitted for the external or internal thread for a specified thread class or tolerance class. In practice, unless otherwise specified, the form of root is an indeterminate contour not encroaching on the maximum material form of the mating thread when assembled.

Effective Thread. The effective (or useful) thread includes the complete thread, and those

METRIC SCREW THREADS

portions of the incomplete thread which are fully formed at the root but not at the crest; thus excluding the vanish thread.

Element. Elements of a thread are flank angle, root contour, crest contour, pitch, lead angle, surface finish, major, minor, and pitch diameters.

External Thread. Is one on a cylindrical external surface.

Fit. The relationship resulting from the designed difference, before assembly, between the sizes of two mating parts which are to be assembled. A general term used to signify range of tightness or looseness which results from application of a specific combination of allowances and tolerances in mating parts.

Flank. The flank (or side) of a thread is the part of a helical thread surface connecting the crest with the root. Theoretically, a straight line in an axial plane section.

Flank Angle. The flank angles are the angles between the individual flanks and the perpendicular to the axis of the thread, measured in an axial plane. Flank angle of a symmetrical thread is commonly termed the half-angle of thread.

Form of Thread. The form of a thread is its profile in an axial plane for a length of one pitch of the complete thread.

Full Form Thread. See Complete Thread.

Functional Diameter. See Pitch Diameter.

Height of Fundamental Triangle. The height of the fundamental triangle of a thread, that is, the height of a sharp-V thread, is the distance, measured radially, between the sharp crest and sharp root.

Height of Thread. The height (or depth) of thread is the distance, measured radially, between the major and minor cylinders.

Helix. A helix is the curve formed on any cylinder by a straight line angular to the axis, and in a plane that is wrapped around the cylinder.

Helix Angle. The helix angle is the angle made by the helix of the thread and its relation to the thread axis. The helix angle is the complement of the lead angle.

Incomplete Thread. Is the threaded profile having either crests or roots, or both crests and roots not fully formed, resulting from their intersection with the cylindrical or end surface of the work or the vanish cone. It may occur at either end of the thread.

Interference Fit. A fit between mating assembled parts that always provides an interference.

Internal Thread. A thread formed on the inside of a cylindrical or conical surface.

Lead. The axial distance between two consecutive points of intersection of a helix by a line parallel to the axis of the cylinder on which it lies, i.e., the axial movement of a threaded part rotated one turn in its mating thread.

Lead Angle. On a straight thread, the lead angle is the angle made by the helix of the thread at the pitch line with a plane perpendicular to the axis.

Length of Complete Thread. Is the axial length of a thread section having full form at both crest and root; (that is, the vanish threads are not included) but, also including a maximum of two pitches at the start of the thread which may have a chamfer or incomplete crests.

(IFI Note: When designing threaded products, it is necessary to take cognizance of: (1) such permissible length of chamfer and (2) the first threads which by virtue of gaging practice may exceed the product limits.

METRIC SCREW THREADS

However, when the application is such as to require a minimum or maximum number, or length of complete threads, the specification shall so state. Similar specification is required for a definite length of engagement.)

Length of Thread Engagement. The axial distance over which two mating threads, each having full form at both crest and root, are designed to engage.

Limits of Size. Are the applicable maximum and minimum sizes.

Major Diameter. On a straight thread the major diameter is that of the major cylinder.

Maximum Material Condition (MMC). The condition where a feature of size contains the maximum amount of material within the stated limits of size. For example, minimum internal thread size, maximum external thread size.

Minimum Material Condition (Least Material Condition (LMC)). The condition where a feature of size contains the least amount of material within the stated limits of size. For example, maximum internal thread size, minimum external thread size.

Minor Diameter. On a straight thread the minor diameter is that of the minor cylinder.

Nominal Size. Is the designation which is used for the purpose of general identification. For threads the nominal major diameter is used.

Pitch. The pitch of a thread having uniform spacing is the distance, measured parallel to its axis, between corresponding points on adjacent thread forms in the same axial plane and on the same side of the axis. Pitch is equal to the lead divided by the number of thread starts.

Pitch Cylinder. An imaginary cylinder of such diameter and location of its axis that its surface would pass through a straight thread in such a manner as to make the widths of the thread ridge and the thread groove equal and, therefore, is located equidistantly between the sharp major and minor cylinders of a given thread form. On a theoretically perfect thread these widths are equal to one-half the basic pitch.

Pitch Diameter. On a straight thread the pitch diameter is the diameter of the pitch cylinder.

Profile of Thread. See Form of Thread.

Reference Dimension. A dimension, usually without tolerance, used for information purposes only. It does not govern production or inspection operations. A reference dimension is derived from other values shown on the drawing or on related drawings.

Root. Is that surface of the thread which joins the flanks of adjacent thread forms and is immediately adjacent to the cylinder or cone from which the thread projects.

Runout. Is a total tolerance used to control the functional relationship of one or more features of a part to a datum axis. There are two types of runout control –

a) Circular Runout provides composite control of circular elements of a surface.

b) Total Runout provides composite control of all surface elements.

Size. Size is a designation of magnitude. When a value is assigned to a dimension it is referred to as the size of that dimension.

Thread Runout. See Vanish Thread.

Thread Series. Are groups of diameter/pitch combinations distinguished from each other by the pitch of the thread applied to specific diameters.

METRIC SCREW THREADS

Tolerance. The total amount by which a specific dimension is permitted to vary. The tolerance is the difference between the maximum and minimum limits.

Total Thread. Includes the complete and all of the incomplete thread; thus including the vanish thread, and the lead thread.

Vanish Thread (Partial Thread, Wash-Out Thread or Thread Runout). Is that portion of the incomplete thread which is not fully formed at the root or at crest and root. It is produced by the chamfer at the starting end of the thread forming tool.

Variables. Are quantities or measurements, that may assume a succession of observed values, when measured under different (or similar) conditions.

Virtual Diameter. See Functional Diameter.

THREAD ACCEPTABILITY

The purchaser (designer, engineer, etc.) is the only one completely familiar with the service application in which the threaded fastener he orders will be a component part. He alone knows the design parameters, the severity of the intended service, and the performance expected. Consequently, he must accept the responsibility for identifying the level of assurance needed that thread dimensional conformance be satisfied.

In ASME B1.3M, gaging systems have been established to give the user an opportunity to conveniently select the system which offers the level of assurance he decides is warranted. This should be a design, not a procurement decision. While each system is comprehensive, there are provisions for modifying any of them to give more, or less, inspection to any single thread characteristic depending on its importance to the needed performance when in service. ASME B1.3M includes a listing of gages and gaging equipment which have the capability to establish for each thread characteristic whether or not the dimensional requirements of that thread characteristic have been satisfied. This standard carefully avoids indicating any preference as to which gage or gaging equipment should be used when inspecting a thread characteristic. Instead, the standard allows the producer and purchaser to independently choose which of the gages they each prefer to use for inspection of the thread. The standard then states that within the specified gaging system for thread acceptability by any of the listed gages is the criterion for acceptance of the thread characteristic being examined. Satisfying the designated gaging system is compliance of the thread to the standard dimensional requirement.

GAGES AND GAGING PRACTICES

The gages and gaging equipment most commonly used in North America when inspecting the dimensional conformance of metric screw threads are described in ANSI/ASME B1.16M, "Gages and Gaging for Metric M Screw Threads". ISO 1502, "Gaging of ISO General Purpose Metric Screw Threads", is the international standard describing the gages and gaging practice used elsewhere throughout the world. There are fundamental differences between B1.16M and ISO 1502. The most notable difference is that North American gage design practice is to place gage tolerances entirely within product tolerances, while ISO gage design practice is to place the gage tolerances partially within and partially outside of the specified product limit of size. The effect of this difference is most evident when inspecting the min material condition of a product thread. The smallest size of the LO thread ring gage as specified in B1.16M is coincident with the min material limit specified for an externally threaded product; and the largest size of the HI thread plug gage is coincident with the min material limit specified for an internally threaded product. Theoretically, when the gage wears it may reject conforming product. To compensate for this possibility, when HI and LO thread gages are used, the gage is permitted to enter the product providing that within two full turns of gage entry a definite drag (interference between gage and product threads) is experienced. ISO 1502 gages are more favorably toleranced in respect to the product threads they examine. However, when used, they are applied using the NOT GO concept which means that the gage is not permitted to enter beyond two full threads. If it engages the product thread beyond two full turns, even with definite drag, the product is subject to rejection.

In the actual inspection of threaded fasteners the occurrence of product acceptability using one practice and rejectability using the other is highly remote. However, it is confusing and a disservice to manufacturers and users of threaded products that two sets of gages and two gaging practices exist internationally. In 1980, IFI recommended to ASME Standards Committee B1, which has responsibility for screw thread standardization in the USA, that a possible single practice might be the use of B1.16M HI and LO gages in combination with the ISO NOT GO gaging practice. This would obligate improved manufacturing control by all threaded parts producers regardless of which of the two gaging practices they now use. However, the benefits and ultimate economies of a single international practice would justify such a decision. Committee B1 was receptive. In 1983, the USA proposed to ISO/TC1 a revision to ISO 1502 which was subsequently defeated. However, the USA was authorized to lead a working group to try and formulate a single world practice. This work is still going forward.

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

All three 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 externally threaded fasteners and those with class 4h6h threads, where the severity of the application indicates a need for closer inspection of the minimum material size.

System 21 is suitable for practically all internally threaded fasteners, except those with UNJ threads which should be inspected using System 22. While System 23 for internal threads is included in B1.3M, 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. Detailed internal thread inspection is handicapped by the physical constraints and access problems on the gaging equipment.

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 the referee technique. 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. Once the acceptability system is established, judgement of product thread is based on gage acceptance, not dimensional conformance.

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 on page A–9 gives thread stress area values for the metric thread series. Also, in the design data section are the formulas for these areas (page A–7).

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 engagement, 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.

METRIC SCREW THREADS

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 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 60° 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 nut 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 screw holes with large surrounding material, dilation as a strength reducing factor may not be as important.

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

METRIC SCREW THREADS

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, page A–9, 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 F606M, Para. 3.4, page B–133) 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 more – be left within the grip.

METRIC SCREW THREADS

ENGINEERING STANDARDS

All necessary information, data and requirements for metric screw threads; M Profile and MJ Profile are covered in a package of seven separate standards.

ASME B1.3M, *"Screw Thread Gaging Systems for Dimensional Acceptability – Inch and Metric Screw Threads (UN, UNR, UNJ, M and MJ)"*.

ANSI/ASME B1.7M, *"Nomenclature, Definitions and Letter Symbols for Screw Threads"*.

ASME B1.13M, *"Metric Screw Threads – M Profile"*.

ANSI/ASME B1.16M, *"Gages and Gaging for Metric M Screw Threads"*.

ANSI/ASME B1.21M, *"Metric Screw Threads – MJ Profile"*.

ANSI/ASME B1.22M, *"Gages and Gaging for MJ Series Metric Screw Threads"*.

ASME B1.30M, *"Screw Threads – Standard Practice for Calculating and Rounding Dimensions"*.

Copies of these documents are available from The American Society of Mechanical Engineers, Three Park Avenue, New York, NY 10016-5990.