Commemorating Thomas Dopppke, Former President of Technical Presentations Company

Thomas Dopppke, an esteemed fastener expert with over sixty years of in depth experience in the fastener industry and a long time contributing author to Fastener World Magazine, passed away at the end of October 2017. Jane Doppke, married to Thomas, said, "Tom was man who loved his family and friends. He was a wonderful husband, father and grandfather. There is a great hole in our hearts. We will work on filling it with wonderful memories." Fastener World Magazine would like to pay homage to him for his decades long contribution to the fastener industry.

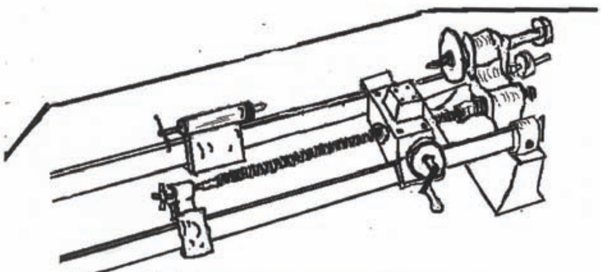
Thread Math

螺紋的規格原理

by Thomas Doppke

Threads are just spirals wound around a cylinder, aren't they? Then why do we have all those tables of numbers about diameters, allowances, fits and such? It sure is confusing and why bother with them? "If it fits together then use it" is a common philosophy.

When threaded fasteners were first used, even before that, when mating threads were first conceived, they were manufactured by hand fit. That is, each part was filed or formed to exactly mate with an opposing part. If one part was lost, the only solution was to use another set of parts or laboriously hand make a mating part. The invention of the thread cutting lathe allowed fairly close duplicates to be made and interchangeability, of a sort, was achieved.



H. Maudslay's Screw Cutting Lathe
The machine cuts the threads, not the operator

It wasn't until the rise of the "Industrial Revolution" that fastening became more than an occasional topic. The increased demand for goods of all shapes and types meant that manufacturers needed to increase the volume of production. Bolts and nuts, tapped holes to hold together machinery (from carriages to locomotives, pumps for the mines to forming presses and arbors, typewriters, and the great textile industry). The Civil War in the United States manifested a need for military hardware, guns to cannons, all made on machinery with fastened joints. But still, every fastener maker had his own thread; shape, diameter and fit were what he made on his equipment.

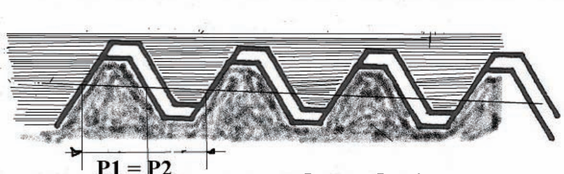
Earlier (late 18th century) the American Government had concerns about interchangeability and offered a government contract to the company able to demonstrate interchangeability of parts. Eli Whitney, of cotton gin fame, showed this by assembling almost 100 muskets from barrels of individual parts, winning the lucrative contract for military small arms (in about 1974 it was discovered that Eli had hand fitted the pieces together privately, so each would fit with each other first and then demonstrated the fact that each would fit together with another in front of the government inspectors (not the first fraud on a government contract!).

But there was no interchangeability between manufacturers. While the introduction of gages and better tools allowed the manufacturers to make interchangeable parts, each maker still made his own threads. The increase in global need for parts forced manufacturers to consider some sort of standardization, a way to ensure that parts made by different makers could be used interchangeably. Sir Joseph Whitworth, a British manufacturer studied the thread forms widely available at the time (circa 1840's) and came to the conclusion that a 55 degree thread angle was the most used at the time, in terms of ease of manufacture, stability, and set-up. Other thread forms in use varied from asymmetrical thread forms, various thread angles, and a wide variety of individualized ideas. His 55 degree angle thread form became the country's generally used standard. In 1864 the United States proposed a 60 degree thread angle and it was easier to manufacture and gage. The proposed thread had flat thread crests and a rounded root radius. The Whitworth thread had rounded crests and roots and was found to perform better in dynamic applications with the rounded roots also showing better fatigue performance. An international conference in 1898 formulated the S.I. Metric Thread Standard which became many member countries' own standard. Further work and the need for global cooperation refined the work to produce a series of 60 degree thread pitches.

Fine! We now have a kind of a standard. Let's go out and make threads! Wait! Some of our threads do not fit into some of our internally threaded parts (nuts, tapped holes, etc.). The same problem arose when the first "standard" threads were produced. Early fasteners "kind of" fit together. They were sort of loose but did tighten enough so that they held together adequately. Loads were not great and any attachment was good. But as joints come to near more and more loading, vibration, and demands for better security and strength, the need for tighter specifications became apparent. The loose fits needed to be tighter, meaning that the fits between mating parts had to be tighter. Also, many of the new applications required tighter fits than in the past. Devices requiring adjustability needed finer thread pitches than what was available.

As fastener manufacture evolved into more of an automated, machine controlled process with interchangeability of parts made by different makers used in original and replacement common attachments, the minor differences between threads and mating joints became an increasing problem. First, no part can be made exactly and repeatedly to the exact identical dimensions. Improved tooling and gaging practices could insure a very close dimensional part but some allowance had to be made for "machining variations". The bolt, obviously, must be smaller than the internal thread it is mated to. Secondly, as the need for additional classes of fits of fasteners arose (tighter fits for joints which require fine adjustments, i.e., optical instruments, weaving loom machinery); looser fits for parts used in high contamination areas of dirt and dust (guns in the field, locomotive machinery) and finally, a general class of fit for most general usage all pointed to the fact that there was an urgent need for the dimensioning of such fits to insure consistency in the industry.

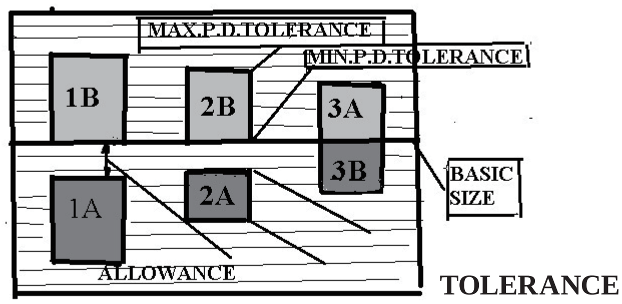
The proliferation of fastener dimensions grew from there. Sharp pointed threads were easily made by lathe cutting but offered a problem at first when mating to internal threads with the same contour. However, it was noticed that after the manufacture of the first few parts, the die tips wore down, producing a quasi-rounded fastener thread crest. While this improved the fit and assembly problems, the amount of rounding had to be controlled lest the fit becomes too loose. As with all dimensional data, the values for the corrected threads had to be measured from a known and measurable starting point (datum). For threads, this was the pitch diameter (the term "pitch line" is used here interchangeably). The origin of the idea of using the pitch diameter as a standard measuring point is unknown. Why it was picked has never been fully explained. It is an imaginary cylinder diameter where the distance between threads is equal to this through the threads (width). For a standard, it was certainly odd to use a datum that could not be physically touched. However, when fasteners are installed, their point of contact between the two mating surfaces (internal and external) is centered fairly well at the pitch line. The pitch line is important as it tells you if the minimum amount of material is present on the thread. It is used in calculating all the mechanical properties of the fastener. The root and crest are not, as illustrated, contacting surfaces and their dimensions are usually measured independently.



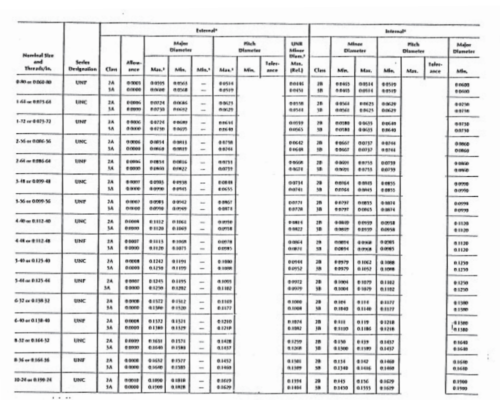
Thread Fix

Other columns on dimensional tables explain other thread factors and their effects on/to the functioning of the fastener and how they affect the pitch line fit. Two terms that confuse people are "tolerance" and "allowance". These terms have several columns in most dimension tables and tend to get ignored as users of the tables have little idea of what they mean. Tolerance is the total amount of variation that is permitted for the size of the thread. It is the difference between the maximum and minimum limit of the size of a given thread size. Rather than go into the various classes of fits and their allowances for Metric threads we will stay with the English (inch) system for now. The same thoughts occur with Metric parts but they have additional classes of fit and their classifications get a bit complex. Metrics will be discussed later.

Allowance is the difference between the basic thread size (1/2-13, ¼-20, etc.) and the maximum material size. For the first two classes of Inch threads, this is an increasing amount of "looseness". Inch external threads are labeled with an "A" (i.e., 1/2-13A) while internal threads receive the letter "B".

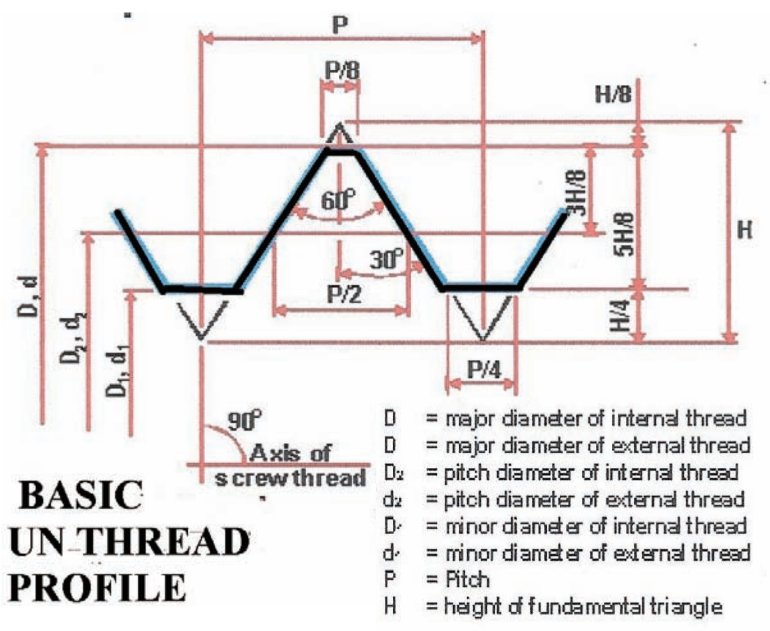


So to give us dimensions of our thread we have a table that lists the values for the maximum major and minimum major diameter of our external thread, its minor diameter (root), its pitch diameter, the amount of truncation of the thread crest and root, the allowance and tolerance values for external threads. There is no allowance for internal threads. This factor (or lack of) was developed when the problems of variation to the dimensions on both sides of the mated joint were considered. To be truly interchangeable the parts had to have dimensions within some accepted limits of tolerance. If dimensions on both sides of the joint were floating within their own tolerance ranges, the problem would be that a part from nut company A with a plus allowance would fit very loosely into a bolt from company B with a plus allowance also. Conversely, the opposite condition would bring about parts that may not assemble, causing high torques, possible jamming and interference conditions. The answer was to allow the addition of the allowance to one side only (see graph above). Restricting the allowance to the external threads only made the parts truly interchangeable. The table shown below is a typical example of one of the many pages found in many specification publications which shows some of the scary listings for numbers, strange formulas and calculations and arcane data that formerly were unintelligible. This illustration is unclear as the data shown may not be current and it is from an old, outdated booklet. Fastener technical data is still in a state of development. Many countries have active fastener societies who are involved in writing and correcting technical data to conform to current practices and problems. Check with your local group to see the availability of the latest specification tables and any re-issued and amended ones.

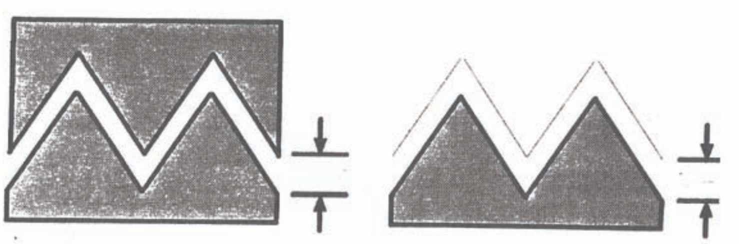


Unfortunately, this causes another problem. With the requirement today for protective finishes (platings, paints, and so on), the amount of allowance available, which can be used for coating thickness addition, is often too little to allow the finished part to meet its class fit limits. In other words, the parts have become too large and exceed their tolerance limits, causing tight fits, if fitting at all. Undersizing the threads is a solution that is often tried, but the fact that there is no allowance for internal threads (which are usually also plated) just exacerbates the situation.

To insure consistency, the amount of truncation of the thread crests is controlled by the tables as is the rounding of the roots. As these additional figures are added to the growing stack of numbers in the specification, our simple thread is taking on more factors for consideration. Here is a basic profile of what we have so far.



While the need for dimensional control has probably been proven by now, the differences between allowance and tolerance are still confusing to some (my proof reader for one). A simple explanation is shown below.



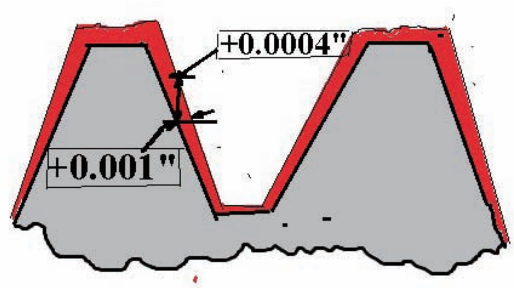
ALLOWANCE
The space between mating parts to allow for ease of assembly

TOLERANCE
The amount of plus and minus in the thread height. (Maximum and minimum material condition)

Now we come to Metric threads. The fastener community decided that certain problems that occurred occasionally with inch series threads could be eliminated if there were even more specifications on dimensions. One condition, as mentioned above, is the use today of thick and heavy coatings on fasteners for protection against corrosion. In times past, the entire structure would rust, but the increased use of better materials and finishes improved the cosmetic appearance to the point where a seldom noticed corroded fastener stood out measurably.

Whereas the Inch system was limited in any other than standard joints, the Metric system had classes of fits and tolerances to fit almost any application. As with the Inch system, the allowance is applied to only the external threads.

External threads are denoted with a lower case letter, internal threads with a capital letter: Bolt- M10 x 1.5 5g6g. M10 x 1.5 being the basic nominal size and pitch in mm per thread (this is a change from the inch system which denoted threads per inch). The last numbers are the tolerance class designation symbols. The numbers 5 and 6 are the tolerance grade symbols and the two g's are the tolerance position symbols. The first grouping (5g) refers to the flank diametral displacement (another term for pitch diameter). Flank diametral displacement is the amount that the pitch line is moved from pitch line basic. Any additions to a flank at the itch line result in an overall increase in the pitch line by a factor of at least four. The addition of 0.001" of plating will increase the pitch line (and subsequent fit) by an amount of 0.004".



Flank diametrical displacement

The meaning of the 6g grouping is the crest diameter (major diameter) tolerance symbols which is fairly self-explanatory. The tolerance symbols indicate several things at a glance. First is the amount of allowance for that fastener (large, small or no). Second, what is the workable length of engagement of the threads (one inch system problem was that long lengths of engagement caused jamming as small differences between individual threads would add up to jamming conditions, even though each individual thread was within specification). Finally, the thread grouping as to pitch (size). The specification allows for general pitches denoted as coarse, medium, and fine thread. Additional pitches are also permitted as specials.

As if the situation could not get any more complex, the fastener societies decided to add a few more symbols to the equation. To denote how the joint fits together (nut to bolt), the thread designation may also look like this: M12 x 1.5 6H/5g6g. The capital H is the symbol for an internal thread (nut) with a tolerance class of 6H and is to fit to a bolt with a 5g pitch diameter and a 6g major diameter.

Preferred Tolerance Classes

Quality	External threads (bolts)									Internal threads (nuts)					
	Tolerance position e (large allowance)			Tolerance position g (small allowance)			Tolerance position h (no allowance)			Tolerance position G (small allowance)			Tolerance position H (no allowance)		
	Length of engagement			Length of engagement			Length of engagement			Length of engagement			Length of engagement		
	Group S	Group N	Group L	Group S	Group N	Group L	Group S	Group N	Group L	Group S	Group N	Group L	Group S	Group N	Group L
Fine							3h4h	4h	5h4h				4H	5H	6H
Medium		6e	7e6e	5g6g	$\boxed{6g}$	7g6g	5h6h	6h	7h6h	5G	6G	7G	5H	$\boxed{6H}$	7H
Coarse					8g	9g8g					7G	8G		7H	8H

Preferred tolerance classes. In selecting tolerance class, select first from the large bold print, second from the medium-size print, and third from the small-size print. Classes shown in boxes are for commercial threads.

The chart shows the preferred class combinations as boxed items. Almost all fasteners in use today are, thankfully, Grade 6, medium tolerance quality and normal length of engagement applications. Nuts are almost always 6H. What started as a simple idea to make things the same several dozen of years ago (hundreds?) has expanded into a voluminous system that attempts to control every aspect of the thread's geometry. While trying to include every possible condition of fit, common sense dedicates that it cannot possibly be done. New situations and conditions, materials arise, necessitating changes and additions to the specifications. The Metric system has, for example:

5 tolerance grades (4-8) available for minor diameter of internal threads

3 tolerance grades (4,6, and 8) for major diameter of external threads

5 tolerance grades (4-8) for pitch diameter tolerance of internal threads

7 tolerance grades (3-9) for pitch diameter tolerance of external threads

Although you may think that this certainly enough there is always the guy who says, "how about ...?" The only saving grace to this confusion is that most fasteners fall within the general use classification and the problem of figuring out the various dimensions are mostly mute. If the need arises a slow, studious look at tables will afford the correct numbers if the points mentioned here are heeded.