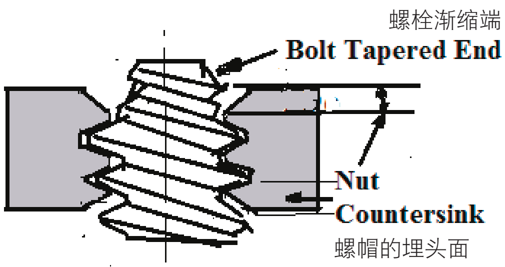
Nut Facts: True or False?

by Thomas Doppke

Over time several stories about fasteners and fastening have been circulated and, as with most things often told and re-told, they become believed. Are they really true or just tales? Let's look at several of these 'truths' about nuts.

Fact one is the often stated comment that **joints must be designed so that at least two threads protrude from the joint** to insure proper joint strength. Statements to this effect are found in many industrial and engineering documents. For example The Industrial Fasteners Institute Specification IFI 124 "… a minimum length of REPT screw equivalent to two thread pitches shall project through the top of the nut". Since most people tend to believe that anything published MUST be true, this design rule is sincerely followed. Let's see how much strength is lost if this is not adhered to.



It is known (based upon actual and numerous tests) that the threads of a bolt are loaded fairly equally along its entire length of engagement while the threads of an internal member are loaded progressively from the first engaged thread onwards. When loaded, the first engaged thread of a nut member supports about 38% of the entire load. As the load increases the threads begin to bend and the nut starts to dilate at the lower edge (first engaged thread side). To develop full joint strength as much of the load as possible must be within the nut geometry. A nut is manufactured with a countersunk area on either one or both sides to allow for easy access of the bolt member. Also the nut hole is usually made by "punching" it out which results in the hole on the side opposite the punch entry being slightly "blown out" (bell shaped) which causes that section, when threaded, to have a slightly larger minor diameter (decreased tensile stress area). This is in addition to the results of the countersinking. On the other side of the joint, the bolt is made with a chamfered end to also allow for easy entry. This chamfered area (called a header point) is allowed to be, by specification, from one half to 1 ½ thread pitches. A study made in 1977 showed that a nut can theoretically develop full strength in the joint when it is engaged 0.60 full diameters. However, to accommodate the "enlargements" mentioned above the usual rule is to make the nut height about 0.75 diameters thick.

Adding up all the information we conclude that this fact is **TRUE**. To summarize, the nuts have a countersink on both ends which allows this portion of the nut height to have a larger than specified minor diameter, which in turn, means less than full thread engagement with the mating bolt threads. Finally, nuts are designed to give the maximum amount of strength with a minimum amount of material. Too much steel and the part is costly, too heavy , takes more time to tap, less parts per box (shipping, handling) , etc. Flush assemblies may work as the designers seldom utilize the full strength of the parts anyway (other than construction work no one tightens fasteners to yield, 90% is the normal level). So while it may be possible to tighten to a flush a condition, two threads out is a safety measure more

「流言」真，抑或假？螺帽篇

长久以来，有些关于紧固件和紧固结合的故事流传着，很多因为一再被提起，所以就被信以为真了。至于是真实或只是传说而已？让我们来看看这些有关螺帽的「事实」吧。

流言一：常被提到的见解是，**在设计上至少要有两条螺纹由接头突出，才能保证适当的接合强度**。在许多工业的和工程的文件中，可以发现有关这种效果的陈述。譬如说，美国工业紧固件协会规格中的IFI 124明订「…REPT螺丝至少应有相当于两条螺纹间距的长度是突出于螺帽上部」。由于多数人倾向相信那些发表的事情「必须」是真实的，所以这个设计的规则，就被真心诚意地遵循着。我们来看看，若这个规则没有照着做的话，会损失多少强度。

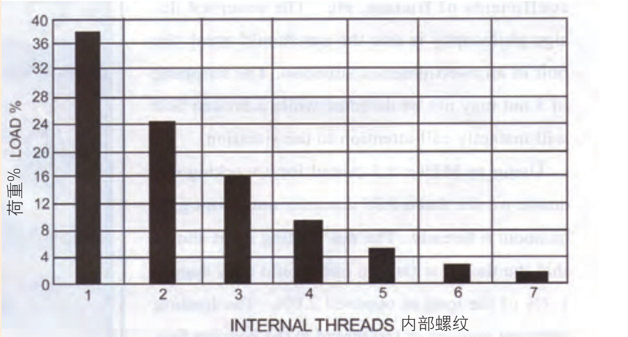
我们知道(基于实际和众多试验)螺栓螺纹在它整个结合长度上，是相当均等地承受荷重，而内部组元的螺纹结合，则是从第一条结合的螺纹向内逐步地承受荷重。当加上荷重时，在螺帽组元上第一条结合的螺纹，支撑了大约38%的全部荷重。当荷重增加时，螺纹开始弯曲，而在螺帽的较低端点开始扩张(第一条结合的螺纹端)。若想要尽可能发展出和荷重一样大的总体结合强度，必须是在螺帽的几何形状之中来形成。一般所制造出来的螺帽，在它的一端或是两端的埋头区域，必须容许很轻易的接合到螺栓组元。而螺帽的孔通常是由"冲孔"冲出来的，这会在相对于冲头进入的端点，产生有一点点"爆开来"(钟形的)的孔，而导致这一区段，在做螺牙时，有略为较大一点的直径(低张应力的区域)，这是除了埋头面之外产生的结果。在接头的另一端，做成螺栓需要有一个倒角端，以容许较容易地锁入。这个倒角的区域(称作打头点)，依照规范是要能够容许一半到1.5条螺纹间距。在1977年的一项研究显示，理论上来说，一个螺帽在接头面若有全部直径的0.60之接合时，就可以发展出全部强度的。然而，为了适应如上所述的"放大"，依一般的规则要求，则要做出大约高度为0.75直径之厚度的螺帽。

综合所有资讯，我们结论这一事实**为真**。总结来说，在螺帽的两端都有一埋头面，这可容许这一部份的螺帽高度，有较规范大些的小直径(minor diameter)。因而，这表示与配对的螺栓螺纹，会有较少于全部螺纹的结

than an absolute requirement. When considering that most designs will fluctuate and a condition of "at the edge" of tolerances of the various parts of the joint and fasteners could cause an minimum stack up of tolerances, the idea of the extra two threads sounds like good engineering practice. Fact two below will explain this in a bit more detail.

合。最后一点，螺帽是被设计用最少的材料量，来达到最大的强度。太多的钢铁会使零件有高成本、也会太重、花较多的时间去打螺纹、每箱装较少的零件(出货运送、处理)等。不过因设计者很少利用到零件的全部强度时，平头(Flush)的组合件或许是可行的(而不是像在营建工程，没有人会锁紧紧固件到降伏点，到90%是正常的水准)。所以若是可以锁紧到平头的状况时，露出两条螺纹是一种安全的措施，而不是绝对必要的要求。当考量到大多数的设计都会有波动性，而各式各样的接头零件和紧固件，都在公差"边缘上"的状况，这有可能产生最低的允差堆迭，所以多出两条螺纹的主意，听起来应该会是很好的工程实务。在下面的事实二会解释地更为详细一点。

Fact two. To increase the strength of the nut it should be made thicker. It has been calculated that the countersunk area of a nut contributes only about 40% of what a full thread in that area would. As we discussed above, an internal thread member carries about 38% of the load at the first engaged thread, the next threads are loaded with about 25%, then 16%, 9%, 5%, 3% , and 2% of the remainder of the nut/joint load. These figures are approximate but close, with the additional threads carrying less and less loading.



Approximate load carried by engaged internal threads(%)
由内部结合的螺纹所承受的大略荷重(%)

These additional threads would only carry less than one percentage of the load. As has been reported in numerous papers, the height necessary to carry a normal loading (90% yield) is approximately 0.75 diameters. Illustrating this, let's look at an M10 x 1.5 thread. At 0.75 diameters thick this is about 6 threads. The nut loading chart shows that the last thread, if fully engaged, should carry about 2% but with the reductions due to countersinks and bolt taper end it is really more like 1.2%.This loss of load carrying capability of the nut is well within the design parameters of the part. Therefore this fact is a **false**. Increasing the height of a nut above 0.75 diameters does not increase the strength of the joint any measureable amount.

流言二：**要增加螺帽的强度，它必须要做得厚一点**。有人曾计算过，若与用了全部螺纹接合比较的话，螺帽的埋头区域的贡献只有大约40%。如前所讨论，一个内部的螺纹组元，在第一个结合的螺纹上可承受约38%的荷重，接下来的螺纹为25%，其余的螺帽/接头负荷，依次为16%、9%、5%、3%和2%。这些是约略的数值但相当接近了，额外的螺纹会承受越来越少的荷重。

这些额外的螺纹应该只会承受到小于百分之一的荷重。一如在许多论文中所报告的，能承受正常荷重(90%降伏的)的高度，大约是直径的0.75倍。为了说明这个，我们来看看一个M10 x 1.5的螺纹，在0.75直径厚度时有大约6条螺纹。由螺帽荷重的图表显示，最后一条螺纹，若是完全结合的话，应该会承受到2%，但会因埋头和螺栓渐缩端的原因而减少，它实际上可能会只达到约1.2%吧。螺帽因此所损失之承重能力，是落在零件设计参数之内甚多的。所以这一事实**为伪**。增加螺帽的高度超过直径的0.75倍，不会增加任何可量到的接头强度。

Fact three. Nuts can be increased in strength by being made of harder material and/or wider. Making nuts wider and/or harder retards the dilation when they are loaded to their maximum load ability as discussed above. However, the additional strength gained is minimal. Since the threads primarily carry the load, width increases do little except increase cost and weight. Making the nut of harder material will increase the strength slightly but again, the threads carry the load. It is standard practice to match the nut strength to the bolt being used. Harder bolts are tightened to higher loads and will increase stress on the nut threads; however, it is usual for the nuts to be increased in strength as harder bolts are used. However, the overall strength of the nut will be of a slightly lower strength than bolt material. If higher loading values are required they can be best obtained by the use of larger size fasteners. So technically this is a **True** fact but barely.

流言三：用**较硬或是较宽的材料来做会增加螺帽的强度**。用较宽和(或)较硬的材料，如上所述，会在它们承受到最大荷重能力时阻碍扩张。然而，由于螺纹是主要承受荷重的，除了增加成本和重量外，宽度的增加是没贡献什么的，额外的强度增益是极微小的。用较硬的材料做螺帽将会增加一点强度，而同样地，是螺纹承受了荷重。标准的实务是螺帽要配合所用的螺栓，用较硬的螺栓会要较高的荷重来锁紧，这将会增加作用在螺帽螺纹上的应力，然而，当用了较硬的螺栓时，通常会需要较高强度的螺帽来配合。然而，螺帽的全体强度会较螺栓材料低一点。若要求较高的荷重值时，它们最好可由使用较大尺寸的紧固件来获得。因此，纯技术来看，这一事实**勉强为真**。

Fact four. **Locknuts increase the strength of the joint.** Locknuts are designed to and work by retarding the loosening action in joints. Nuts loosen when an external load is cyclically applied and released during service. After numerous applications of vibrational impacts the nut begins to turn off. If the nut is tightened to instill a preload greater than the actual service load that the joint will see the joint will not loosen. Proper torque will prevent this from happening most of the time but the real life conditions are such that severe events (rough roads, jolts and impacts) happen. These forces often exceed the maximum joint load capability momentarily. Enough impacts and the joint loses preload; enough loss and the parts loosen. Locknuts do not contribute anything to the strength of the joint. They only dampen the vibrational forces to slow down the loosening. Enough and harder impacts will eventually loosen any joint. Also, since the locking feature is usually located at the top end of the nut, they must be threaded onto a full dimension thread to fully engage. This extra height of the added locking feature should be added to the total nut height. This is a case where two threads protruding is a must condition. It is **false** that locknuts improve the strength of a nut.

流言四：防松螺帽(Locknuts)可增加接头的强度。防松螺帽被设计的作用是防止接头的松动。若外加荷重在服役中，是循环地施作和松开时，螺帽会因此而松动。在许多次振动性的冲击施作时，螺帽会开始松动脱落。若这个螺帽被慢慢地锁紧至预加荷重，若它是大于在实际服役所承受的荷重时，这螺帽就不会松开。适当的扭力在大多数的情况下，将会阻止这松动的现象发生，但是在真实生活中，会有严峻的状况发生(譬如：粗糙的路面、颠簸和冲击)，这些力常会在暂瞬间超过最大的接头荷重能力。足够的冲击会使接头失去所预加的荷重，而损失的荷重可能会大到足以让零件松动。防松螺帽对于接头的强度不会有任何贡献，它们只会使振荡的力量衰减而减缓松动，而足够且更大的冲击最终还是会松动任何接头。而且由于锁紧特征是常位于螺帽上部的端部，它们必须锁入且完全接合到一全尺寸的螺牙。这样的外加锁紧特征之额外高度，必须加到螺帽的全部高度。因而在必须要有两个螺纹突出的情况下，这种防松螺帽会改善螺帽的强度是**假的**。

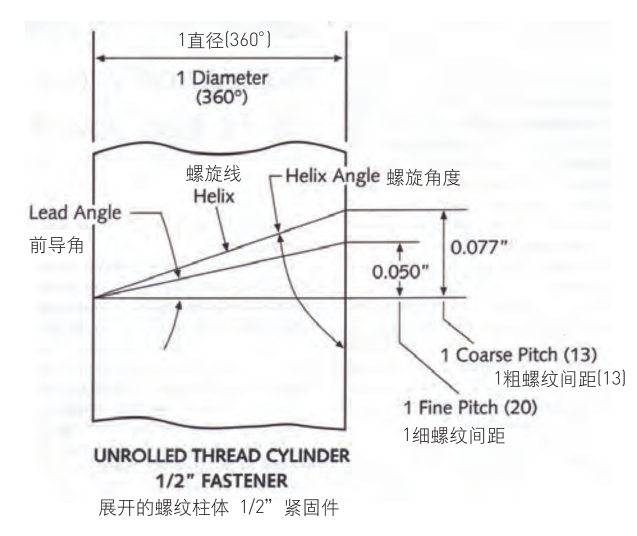
Fact five. **Thread adhesives help nut joint strength better than lock nuts.** Again adhesives retard vibrational loosening and nothing else. While they can keep a part from coming off, if the joint is moved the adhesive bond breaks and loosening may occur. The pros and cons of adhesives over locknuts are that the adhesive joint will require a much higher force to start loosening ("break-away") than a lock nut does. However, once started, adhesive joints loosen faster than locknutted ones. The metal feature locknut is reusable and is not affected by the things that deteriorate adhesive joints, such as time, humidity, and temperature. However, adhesives, when applied to internal threads, saves height that a locking feature added to the nut height would require. They are also cheaper to manufacture as a standard nut can be used with just the adhesive applied as needed. Fact five is **false**.

流言五：螺纹粘着剂(thread adhesive)对于螺帽接头的强度有优于防松螺帽的好处。再次地说明，粘着剂仅仅只会阻碍振荡性的松动。虽然它们可以防止零件脱落，而若接头被移动的话，粘着剂的结合会断开而开始发生松动。粘着剂与防松螺帽比较之好处与坏处，是粘着的接头会需要较防松螺帽更高些的力量才开始松动(断开)。然而，一旦开始松动，粘着的接头会较防松螺帽更快松开。金属特征的防松螺帽可以重复使用，且不会受到一些会劣化粘着接头之状况的影响，譬如时间、湿度和温度。然而，若在内部的螺纹施加粘着剂，是可以节省因锁紧特征的要求而增加的螺帽高度。同时，因为可以使用标准螺帽，它们制造起来较为便宜，因为只要施加所需的粘着剂即可。事实五**为伪**。

Fact six. **Fine thread nuts are stronger than coarse thread ones.** Fine threads came into being when the need for greater and finer adjustment of machinery was needed. Also, it was reasoned that fine threads are stronger since tests showed a higher failure value for finer pitches. This is, in fact, not true as the determining factor is not the tensile stress area but the amount of metal that is engaged to support the load. Fine threads have less thread height, which means that there is less thread flank to support the load. As the nut is loaded, the part begins to dilate which enlarges the nut inside diameter and lessens the amount of thread flank to support the load. As the illustration shows, the coarse thread (here a ½ inch diameter part) is formed with a more obtuse angle, meaning longer flanks, which equals more support for the loading. This makes them more resistant to stripping. The observed difference between coarse and fine thread tensile strengths is due to the number of threads engaged. Using this ½" example times the 0.75 diameter value; the coarse thread part (13 threads per inch) has 9.75 threads for its height. The 20 threads per inch part (standard fine thread) has 15 threads. As we mentioned before little is added for additional threads, but something is! These extra threads can take up a bit more loading (the amount varies slightly with diameter but is about

流言六：细牙螺帽较粗牙强度高。细螺纹的出现是因有较大和较细微的机器调整需求，同时因为试验结果显示有细螺纹间距的，会有较高的破断值，而推理这样有较高强度。此项事实**为伪**，因为决定因子不是拉伸应力的面积，而是多少金属的量被用来承受荷重。细螺纹有较少的螺纹高度，即表示有较少的螺纹侧面用来支撑荷重。当荷重加到螺帽上时，零件开始扩张而撑开螺帽内径，并减少了支撑荷重之螺纹侧面的量。如说明的图文显示，制成粗的螺纹(这里是½英吋的零件)会有一更钝的角度，即表示有较大的侧面，也等于有较多对荷重的支撑，这使得它们更能防止齿的折断(stripping)。所观察到粗和细纹之间，在

11-15% more). Since design parameters allow a margin for safety, this amount is negligible consideration for most designs. An individual thread to thread comparison would show that the one fine thread is weaker than one coarse one.



Is this "Fact" true or a false? There is some weight to both sides of the fact. They are slightly stronger but why they are is a question of the mechanics of the geometry. We leave this one in your hands- **True or false?**

拉伸强度上的差异，是来自结合之螺纹的数量。使用这个½"的例子乘上0.75的直径，(每一英吋有13条螺纹)在粗螺纹的零件的高度上有9.75条螺纹。每英吋20条螺纹的零件(标准的细螺纹)有15条螺纹。如我们前面曾说过的，额外的螺纹不会增加什么，但有些时候会喔！这些额外的螺纹可以承接更多一点的荷重(数量会因直径而略为变化，但大约会多了11-15%)。由于设计的参数容许一个安全系数的裕度，在大多数的设计上这数量是可被忽略的。以个别的螺纹对螺纹的比较，会显示细螺纹的是比一个粗螺纹的为弱。

这事实是真还是假呢？对事实的两方面均有一些权重，它们的强度会高一些，但是为什么这样，是和几何的机构有关的问题。我们将这交到你们手中来决定 – **是真或是假？**

Fact seven. Nuts should fail before the externally threaded member. This idea was based on the relative cost of the bolt vs the nut and upon the possible consequences of a component failure if the joint failure was unnoticed during assembly. The nut is usually a very inexpensive part while the bolt costs many times more. Good design practice dictates that the bolt should fail first. This is because a stripped bolt will be more readily observable on the production line and elsewhere than would a stripped nut. Nuts may strip at or near the set point torque of the joint and the operator will assume that the joint has been successfully attached when, in fact, the nut had stripped. While there may be reasons that an engineer would want the bolt to strip first, in most assemblies the nut is the member designed to fail. This fact is **False**.

流言七：螺帽应该要在外部结合组元之前先行失效。这是基于螺栓和螺帽的相对成本的概念，以及若在组装时，接合失效未被注意到时，组元失效所产生的可能后果。螺帽通常是非常便宜的零件，而螺栓则贵很多倍。好的设计实务明确要求螺栓要先失效，这是因为磨损的螺栓在生产线上或其它地方，会比磨损的螺帽更易观察到。螺帽可能会在预设的接合扭力，或接近预设值时磨损，操作员会假设接头已经成功地接上了，而事实上螺帽可能已磨损。虽然工程师希望螺栓会先磨损，在大多数的组合件中，螺帽是被设计为最先磨损的组元。这一事实**为伪**。

Fact eight. Flanged nuts are a better than nut/washer assemblies. While there are arguments for both sides of this question, the answer lies with the joint mechanics. One piece flange nuts are stronger, being of one piece construction, and will resist failure when used in joints subjected to bending modes. High loads and bending fatigue are the prime problems in critical attachments. Nut and washer assemblies are seen most commonly on sheet metal applications. The washer can be made much wider than the flange of a flange nut can be formed. The purpose of nut and washer assemblies is to hold sheet metal together without the washer embedding; the spread of the washer being the object that does this. These applications generally have wide torque ranges and are quickly assembled with inexpensive tooling.

Flange nuts are the members that hold together solid, robust attachments, those will not move around. They are inexpensive as opposed to the two piece nut/washer parts but require a special set up for each differing configuration. The nut/washer assemblies can be made with a standard nut base and differing washer diameters and shapes added with little engineering.

流言八：有突缘的螺帽比螺帽/垫片的组装件还要好。虽然这一问题的两面都有论点，答案是与接合机制有关。以一个突缘的结构件来说，一个突缘螺帽会有较高的强度，在使用于承受弯折模式的接头时，它将能够耐受破断。高的荷重和弯折疲劳是关键附接装置之主要问题。螺帽和垫片的组装件，是在金属板片的应用上很常见的。垫片可做成较突缘螺帽可用到的突缘宽许多。螺帽和垫片组装件的目的，是在不需要用到崁入垫片之下，来将板片金属夹持在一起；垫片的延展就是为了达到这个目的。这些应用一般会有较广的扭力范围，而用便宜的工具即可很快地组合起来。

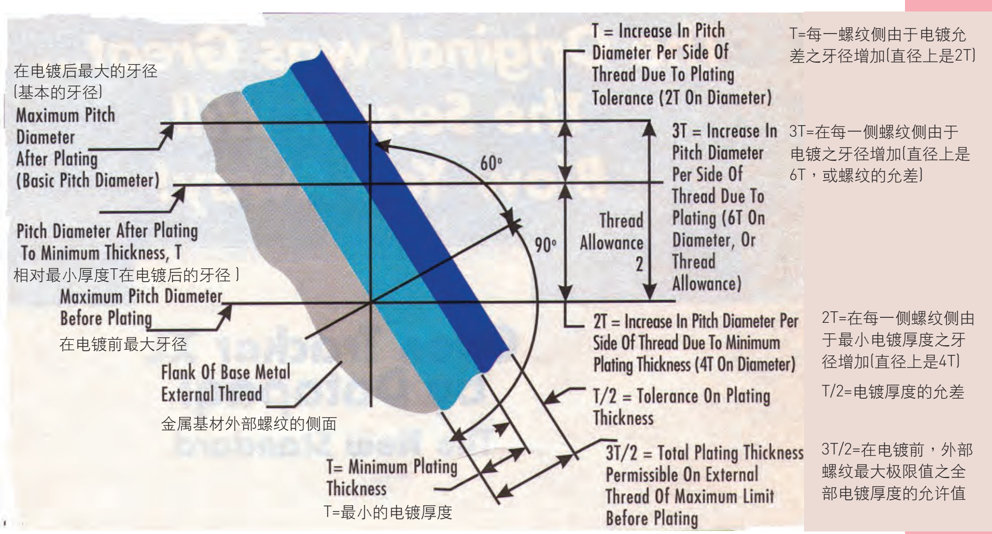
突缘螺帽是将固体、稳固的附件，即那些不会随处移动的东西夹持在一起的组元。它们相对于两件式的螺帽/垫片零件是便宜的，但是需要针对不同的组态有特别的准备工作。螺帽/垫片的组装件可以用标准螺帽为基础，配用不同直径和形状的垫片，再加上很少的工程操作。

Flanged parts are used to span slots and oversized holes and the flanged nut variety is usually insufficient in diameter for most conditions. The question of the verity of this fact again is left to you, the reader.

突缘的零件是被用于张开条状开口和超大尺寸的孔，而各式的突缘螺帽直径，通常是不足以应付大多数状况的直径数值。验证这一事实的问题就再次留给您－读者罗。

Fact nine. Today' s heavy coatings for corrosion resistance require undersizing the nuts. Industrial standards allow for a certain amount of tolerance in the fit of bolts to nuts to compensate for manufacturing variations, dimensional plus and minus, and other factors. If these allowances are exceeded than the parts will not fit together well, if at all. To allow deviations from the set dimension for both bolts and nuts would create an unmanageable situation. Therefore, the industry has decreed that the allowance be applied to the external member only to accommodate the various processes including plating. Since the coatings are applied to both joint components, nuts are often overcoated and cause problems in assembly. Since few fastener makers make both bolts and nuts they are left with a problem. If the nut is to meet quality control standards (thread fit and function) he must overtap the part. But by overtapping he also decreases the strength of the nut. The overtap amount is not strongly controlled by the industry (ASTM A563 only specifies a minimum amount but no maximum amount). Many European metric nuts of class 8 are made to a 0.8 diameter thickness (See above comments on thickness) and have been found to be inadequate in thickness to prevent thread stripping when overtapped. Users have been cautioned not to use any class 8 nuts because of this.

The plating, measured on flat surfaces, is not especially thick, but when it is added to a thread the totals can be alarming. This is because the thickness is added to a rounded surface and at an angle. The illustration shows that the total for a plated thread is approximately 6 times the flat surface dimension.



The following chart is a short list of how thick plating is allowed to be for various thread pitches on bolts. Remember, nuts have no allowance.

Threads per Inch	Maximum Thickness
32 or less	0.00015"
30 to 13	0.00020
10 to 5	0.00030
Greater than 5	0.00050

Since most new, heavy coatings average at least 0.00100" (flat surface measurement) a thread could be as thick as 0.00400 to 0.0060" on a bolt. Again, there is not allowance on nuts!!

Overtapping? **No answer.** Too thin and the part will be rejected for corrosion insufficiency; too thick and it will not pass gaging.

流言九：现今的厚防蚀涂层需要较一般为小的螺帽。工业标准容许在螺帽和螺栓配装时，有某种数值的允差来补偿制造上的变动－尺寸的加减和其它因素。若这些容许度超过了，即使不是根本不行，那么这些零件也不会配合得很好。容许螺栓和螺帽两者都偏离所设定的尺寸，将可能会产生无法管理的情况。所以工业界已强制规定，允许度(allowance)仅可用在外部的组元，来对不同制程(包括电镀)有所调适。由于涂层是被用在接头的两种组元上，而螺帽常被过度包复(overcoated)的，故会造成组装的问题。由于很少的紧固件制造者是螺栓和螺帽两者都做，因此会遗留下一些问题。若螺帽是要符合品管规范(螺纹的适配和功能)，必须对零件过度攻牙(overtap)(ASTM A563仅只规定最小的量，但没有最大的量)。很多欧洲公制的8级螺帽，是以0.8直径的厚度做出来的(请参看对于厚度的建议)，而这曾被发现，当在过度攻牙时，对于防止螺纹的磨损是不恰当的。用户们已对此有所警惕，因而不用任何8级的螺帽。

电镀(在平面上量测)是不会特别厚，但是当它加到全部数值时，可能就需要被警示了。这是因为厚度是有角度地加在圆的表面上，如说明图表所示，一个有涂层的螺纹，全部的尺寸大约是施作在平面上的六倍。

下述的图表是在螺栓上，针对各种螺纹间距可允许多厚电镀层的简短清单，请记住螺帽是没有允差的。

每英吋螺纹	最大厚度
32 (含)以下	0.00015"
30 to 13	0.00020
10 to 5	0.00030
大于 5	0.00050

由于大多数新的、厚实的涂层平均至少有0.00100"(平坦表面的量度)，螺栓上的单一螺纹很可能会厚达0.00400到0.0060"。再次说明，螺帽是没有允差度的。

如果过度攻牙？**无解**。太薄的话，零件将因耐蚀性不足而会被剔退，太厚的话，则无法通过量规检验。