LOCTITE GUIDE TO DESIGNING ASSEMBLIES WITH THREADED FASTENERS.

GUIDE TO DESIGNING ASSEMBLIES WITH THREADED FASTENERS

By
Girard Haviland
Dale Barsness
Will Barbeau

Abstract: Although the "machine age" has evolved into the super-sophisticated "space age" the humble threaded fastener continues to confound product designers. Fasteners are taken for granted and the authors attribute this problem to a number of myths concerning fasteners, lack of awareness of the mathematics of proper fastener design, and failure to recognize the benefits of modern chemical threadlocking technologies.

INDEX

INTRODUCTION

Machine Design published a feature in 1975 titled: "Keeping Fasteners Tight." The editors were amazed to learn that it was *the best-read feature in the magazine!*

High interest in fastener technology is still true today. Consider: "50% of total production man hours are devoted to assembling parts. . . ."

Another quote from the same article: "The simplest way to cut assembly costs is to make sure that once installed, fasteners will stay tight."

What you should expect from fasteners can be summed up in three words: *predictable, uniform performance.* And yet, it's so hard to achieve. If you are assembly engineers in the auto industry, you are concerned about very precise torque tension on head bolts. You have machines to show you the torque values, and the only way you can get a desirable narrow envelope of 'scatter' in the torque readings is to use an anti-friction material under the bolt heads.

The point is, assembly techniques are getting tougher, not easier.

And if you are really a perfectionist, you are also looking for:

— easier assembly
— no accidental loosening
— no seizing
— easy disassembly
— re-useability
— reduced costs
— no corrosion

How much imperfection can you handle?

Perhaps you have heard talk about "statistical significance." It's an impressive technical idea. But the engineers who were getting 99.5% reliability from their axle ring gear bolts found that the "insignificant" half of a percentage of failure was a disaster. With 3 million axles a year they had to rework 15,000 of them at a cost of a million and a half dollars. Pure waste! Perfection is often the only acceptable standard. The point is, no one needs to accept *any* loosening of *any* fastener *any* longer!

Look back on your own career for a moment.

Did you ever get a chance to learn about fasteners in college? Or any school of hard knocks? Did you have to teach yourself most of what you know while on the job? It's a tough way to acquire a sophisticated technology. For years one of the authors has conducted surveys among highly qualified engineers who work in the fastener field. He found that 89% believe vibration alone causes bolt loosening and 29% believe that prevailing off-torque prevents loss of joint integrity. Both are false. They are part of a long-standing mythology about fasteners.

Some of the other well-accepted myths are:

— **A properly tightened bolt will never loosen.**
(Wrong: When friction in the threads and under the head of the bolt start to drop, it doesn't take long for a bolt to loosen.)

— **It takes thousands of hours of vibration to loosen a bolt.**
(Wrong: After side-sliding starts, as few as 100 cycles are needed to loosen a bolt.)

— **Lock wires prevent nut loosening.**
(Wrong: They simply prevent the bolt from falling out of the hole. They don't affect clamp load.)

— **Nylon rings can dampen machinery vibration.**
(Wrong: no effect.)

— **Fasteners take more torque to loosen than to tighten.**
(Wrong: It is easier to go downhill than uphill.)

— **"I know it's tightened properly because I 'torqued it' ".**
(Wrong: Friction is 80% of the effect of torque, not tightness.)

— **Split ring lock washers exert a powerful auxiliary pressure on the underside of the screw head to prevent assemblies from loosening.**
(Wrong: no effect)

— **High torque loads automatically mean high clamp loads.**
(Wrong: 85% of tightening torque is absorbed in the threads and under the head. Only 15% produces clamp load.)

PART I FASTENER INDUSTRY AND HISTORY

Never doubt that you are part of a considerable enterprise. Consider: last year, U.S. industry torqued, turned, crimped, drove, bucked, and clipped more than 400 billion fasteners in cars, appliances, buildings, aircraft and thousands of other products. That's nearly 2 billion per working day!

And fasteners are taken for granted!

A charming booklet you should read is called "The Heritage Of Mechanical Fasteners," published by the Industrial Fasteners Institute (IFI) in 1974. It equates man's progress from Stone Age to Space Age to his ability to fasten useful things together with the invention of nails, rivets, screwthreads and other basic fasteners. Standard threaded fasteners really became a part of our culture in 1799 when Eli Whitney piled muskets parts on the Secretary Of War and proceeded to select parts at random to assemble a complete working musket.

Interchangeable parts! Threaded fasteners of standard sizes!

From there on, the stars of fastener history have been the manufacturers who invented the machines for producing them. English inventor Henry Maudslay was called the 'father' of modern threaded fasteners for his screw-cutting lathe in 1800. American David Wilkinson was credited with fatherhood of the American tool industry for his screw-cutting slide-rest lathe.

The Industrial Revolution saw rapid development of the tool and fastener industries through the 19th century. With the arrival of the 20th century the automotive industry was born and with it came a need for fastener standards. The IFI itself began in 1931, and standardization progressed rapidly after that.

That's a very sketchy history of the fastener industry. The IFI booklet states it well, and knowledge of the industry's growth can put your own appreciation of fasteners in perspective.

PART II WHAT HELP FROM THE HANDBOOKS?

It's time, now, to face a major problem with threaded fasteners that is largely ignored by important segments of both the academic and industrial worlds: **threaded fasteners are reversible mechanisms, and can actually be self-reversing.** Put into English: "they can come loose."

If you dutifully pursue your fastener design responsibility by referring to either the basic Marks or Shigley handbooks you will find *no design data to help you deal with the self-reversing potential of the fasteners you select for your designs.* The design guidance fraternity will help you to select a fastener for your challenge, but they won't help you to hold it together. The assumption is: "if it's designed right, it won't come apart."

You live in a real world where you know that several companies have millions of dollars in sales each year selling threadlocking hardware and chemicals, yet you can find no technical documentation of these products. Magazine yearbooks published by *Machine Design, Assembly Engineering* and others now have "adhesive" and "Locking Devices" sections which describe products, not theory.

Why does this situation exist?

We don't know.

Part of it is inertia. The books are updated only every few years.

Part is a lack of applicable mathematics to deal with assembly dynamics.

Insofar as threadlocking adhesives are concerned, part is the gap between the mechanical and chemical worlds.

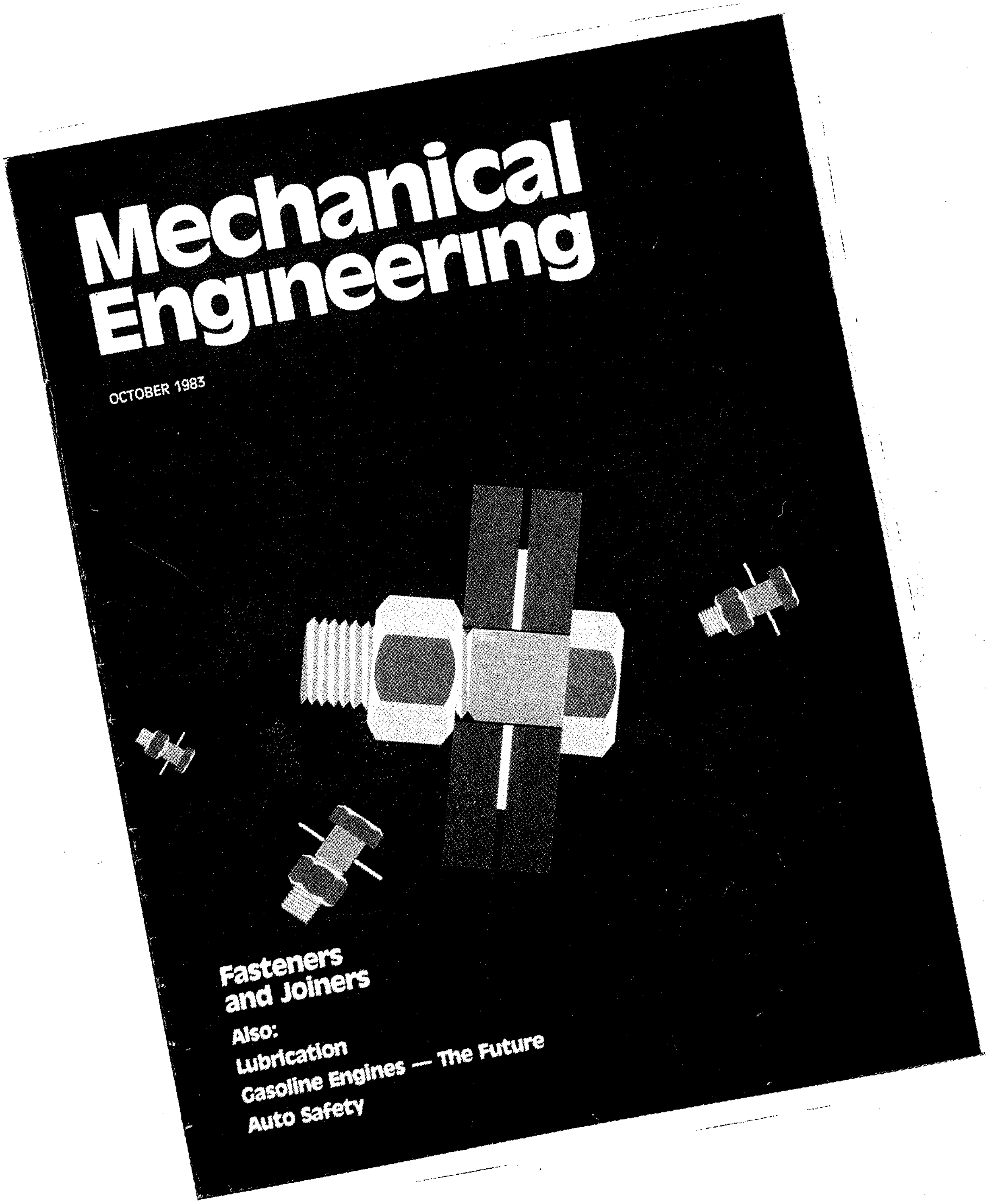
And, of course, some part must be attributed to the broad tendency to take fasteners for granted.

Whatever the reasons, the situation underlies the need for this guide.

When it comes to the problem of figuring out how to hold your newly designed assembly together, you are on your own.

However, many have preceded you and their stories follow.

PART III DESIGNING WITH THREADED FASTENERS



Reprinted from
Mechanical Engineering

NOTES

Reprinted from MECHANICAL ENGINEERING, *Vol. 105, No. 10, October 1983*

Designing with Threaded Fasteners

Next to the common nail, screw products are the most useful fasteners ever developed, but they do have limitations that are not always understood or avoided

Girard S. Haviland
Product Engineering Manager
Loctite Corp.
Newington, Conn.

Archimedes developed and recorded the first spiral screw in 250 B.C. He used it for lifting irrigation water. Up until about A.D. 1450, the screw was used only as a power-transmitting device. Gutenberg used a screw-driven printing press in 1450 to make the first machine-printed Bible; and a few years later, clocks with the first slotted head screws appeared. In 1568, Benson designed a crude screw-cutting lathe; however, it was not until the Industrial Revolution starting in the late 1700s that William Wyatt and Jess Ramsden mass-produced screws in England. At the turn of the century, Henry Maudslay in England contributed to the first interchangeable screws and nuts by designing an accurate lathe. This, coincidentally, was at the same time that Eli Whitney in Connecticut was promoting and proving the principles of interchangeable production parts [1].

Today, according to government survey, assembling parts is half of all the manufacturing labor in the U.S. Five percent of the assembly cost is the fastener itself: nuts, bolts, rivets, pins, etc. Next to the common nail, screw products are the most useful fasteners ever developed. They do have limitations that are not always well understood or avoided. Therefore, it is well worthwhile to look at how a screw fastener works, and how it can be made most reliable and cost effective.

Designing the Bolted Joint

Getting the Right Clamping Force When someone buys a nut and bolt, he with but few exceptions buys just one thing, and that is clamping force. He wants to be able to predict what the force is going to be and how long it will stay at that value. In addition, at the end of some time, he may wish to remove the clamping force. Nuts and bolts fill this function well, but must be "engineered" properly to give satisfactory long-term results [2].

We tighten a screw or bolt by applying torque to the head. A clockwise torque makes the bolt-to-nut distance shorter. If a resistance is met (such as clamping a flange), the bolt will continue to rotate until a balance is obtained between the torque applied to the head and the sum of bolt tension and friction. The distribution of torque between these three factors is shown in Table I.

The equilibrium relationship is often expressed mathematically, $T = KDF$,

where T = Torque in.-lb (N.m)
D = Nominal diameter of bolt-in. (m)
F = Induced force or clamp load—lb (N)
K = An empirical constant, which takes into account friction and the variable diameter under the head and in the threads where friction is acting. (It is not the coefficient of friction although it is related to it.)

Values of K can be determined experimentally (see Table II). (For a mathematical analysis of force and friction see Appendix III.)

The variation in friction and, therefore, K is wide since it is the result of extremely high pressure between surfaces that may be rough, smooth, oxidized, chemically treated, and/or lubricated. Oily steel has a K that varies between 0.11 and 0.17 or ± 20 percent. Friction absorbs 80–90 percent of the tightening torque (Table I). Therefore it is prudent to test a particular combination in a torque testing device [3] to determine proper torque values for assuring good control of bolt tension. Technical data for lubricants and other thread treating materials will often have the K values plotted in torque tension

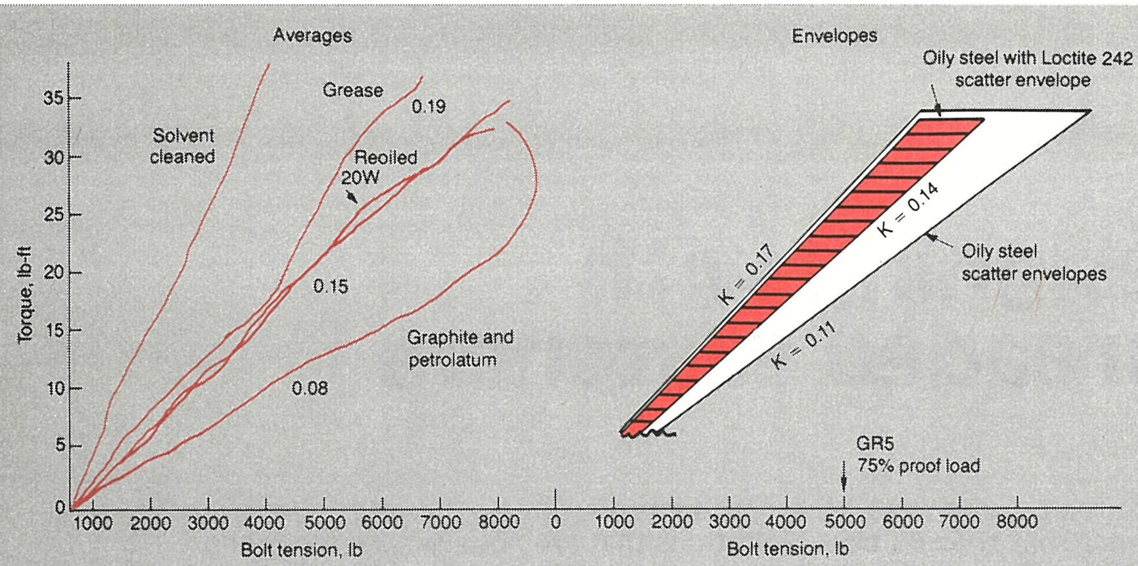


Fig. 1 Torque tension averages and envelopes grade 5 steel.

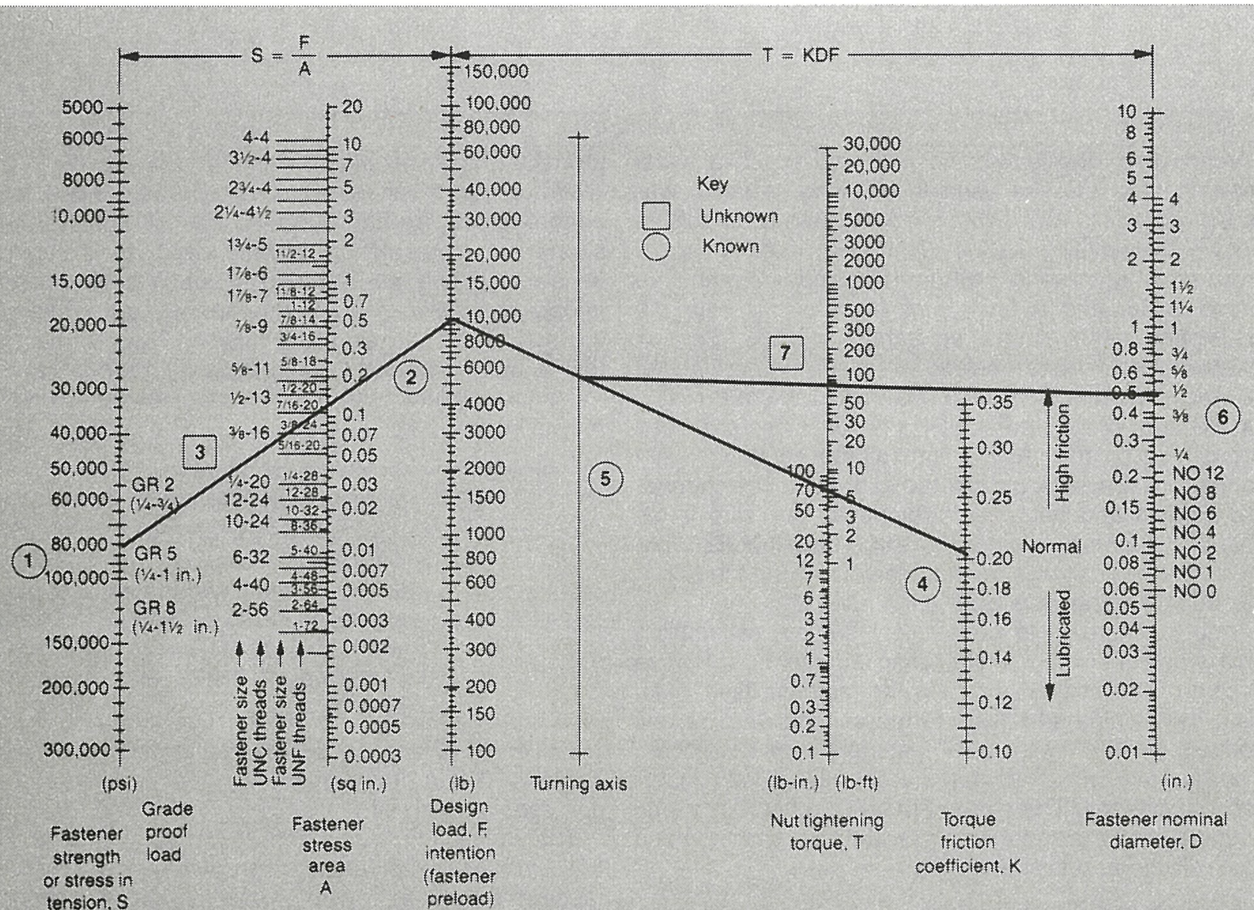


Fig. 2 Threaded fastener nomograph. Assume that fastener design load in a bolted joint is 8000 lb. Since friction conditions can vary as much as 20 percent, assume the 8000 lb represents 80 percent of clamping force. This means average clamping force is 10,000 lb. Assume the minimum yield strength of the fastener material being considered is 100,000 psi or, using 80 percent of this value, 80,000 psi. Draw line between 80,000 psi on *S* (step 1) and 10,000 lb on *F* (step 2). The answer, on scale *A* (step 3), is between 7⁄16 and ½ in. dia. The next larger size, ½ in., should be chosen. To determine the tightening torque required to develop a 10,000-lb clamping load, move to the right side of the nomograph. Assume a normal torque-friction coefficient value of 0.2 (step 4) and draw line to 10,000 lb on the *F* scale. Swing around the point of intersection (step 5) on the turning axis to point on the *D* scale for a ½-in. dia (step 6). Read required tightening torque (step 7), which is slightly over 80 lb-ft. (© *Machine Design.*)

Table I

Torque Absorption in a Tightened Bolt

	Percent of Tightening Torque	
	UNC*	UNF**
Bolt tension	15%	10%
Thread friction	39%	42%
Head friction	46%	48%
Total tightening torque	100%	100%
Loosening torque:	70%	80%

*Unified National Coarse
**Unified National Fine

Table II

Typical* K Values—Lubricating Threadlockers on Various Materials

Substrate	Oil	Lubricating Threadlocker
Steel	0.15	0.14
Phosphate	0.13	0.11
Cadmium	0.14	0.13
Stainless 404	0.22	0.17
Zinc	0.18	0.16
Brass	0.16	0.09
Silicon bronze	0.18	0.24
A1. 6262-Ta	0.17	0.29
Dry Degreased Fasteners		
Steel	0.20	0.20
Phosphate	0.24	0.14
Nylon	0.05	0.13
Zinc	0.17	0.15

All specimens were dipped in 5% oil solution and dried before the threadlocker was applied.
(Heat Bath Corp., Lab Oil 72D)

*Range of values for any lot of fasteners was ±15%; however, different fastener lots can increase the variation to ±20%.

curves as in Fig. 1. These values were obtained on 3/8 × 16 nuts and bolts where the nut was turned. Both the threads and the nut face were lubricated. An unlubricated thrust surface, either nut or bolt head, can almost double the K value.

Choosing the Right Bolt The slope of these straight line plots, that is Torque divided by Tension, is equal to factor K × D and is constant. Since D is also a constant (.375 in.) then each plot has a computable constant K. K = T/FD. K is the same for all diameters. Knowing the friction constant K, the designer can compute the torque tension relationships for other sizes of bolts. To simplify the computation, a nomograph has been drawn as in the right side of Fig. 2.

In addition to the graphing of the expression T = KDF, another critical calculation, S = F/A, has been interrelated on the same nomograph where S = bolt stress lb/in^2. With this nomograph, a complete fastener design can be approximated for rigid clamping by starting with an allowable stress for the steel grade or material selected. Typical manufacturer's catalogue material concerning allowable stress is shown in Fig. 3. Nonrigid joints require further analysis.

Gasketed Flange Bolting The use of a gasket in a joint changes the considerations for bolt selection [4]. The flexibility of the joint increases the possibility that the bolt will experience most of the applied load.

Most of the force for producing minimum sealing pressure in a gasket has to come from the bolts (a small amount may come from gasket adhesion and gasket swelling from chemical and pressure effects). It is therefore necessary that the bolts produce the designed stress in the gasket both on initial assembly and throughout its life. There are four reasons why bolts may fail to produce and hold the desired stress in the joint. They are:

1 Bolts that are too large or too small.
2 Improper tightening.
3 Extreme movement or vibration of the clamped surfaces.
4 Improper material selection with excessive gasket relaxation.

Bolt Sizing for Compressible Gaskets—The strength of the bolt must be high enough to support the preload, which in turn must be high enough to produce the minimum stress that will seal the internal pressure or applied loads. In general, the tension that the bolt supports is not increased by the applied load in a solid metal-to-metal connection. Certainly the preceding statement is not quite true, but it is true enough so that designers of machine joints can obtain adequate designs based on this assumption [5].

If the previous general statement were true, it would be possible to build an infinitely rigid structure. However, according to Hooke's Law, deflection is proportional to applied force. In most joints the ratio of rigidity to fastener rigidity is high enough to discount almost any addition to tension already in the bolt produced by an externally applied load (Fig. 4).

However, in a flexible joint with a soft gasket between bolted parts (Fig. 4), the rigidities of the joint and the bolt are quite different; here, a much greater proportion of the externally applied tension load is added to the bolt preload. The reason for this may become more obvious by studying the following equation:

$$P = P_i + CF_a \tag{1}$$

where P = final load on the bolt, lb; P_i = initial preload or clamping load developed through tightening, lb; F_a = externally applied load, lb; and the constant

$$C = \frac{\dfrac{E_b A_b}{L_b}}{\dfrac{E_b A_b}{L_b} + \dfrac{E_g A_g}{t_g}} \tag{2}$$

where E_b = modulus of elasticity of the bolt, psi; E_g = modulus of elasticity of the gasket, psi; A_b = effective cross-sectional area of bolt, sq. in.; A_g = loaded area of gasket, sq. in.; L_b = effective length of bolt, in.; and t_g = gasket thickness, in.

The value of the constant C falls between 0 and 1. The term E_gA_g/t_g in equation 2 will be large in comparison to E_bA_b/L_b if the gasket is hard, thin, and large in area. Then the constant C approaches zero. When no gasket is used between members in a rigid joint, C = 0. For very soft gaskets, C approaches 1. It is important to remember that equation 2 is only valid as long as the gasket remains in contact with joint members. If the bolt stretches to the point where the gasket is no longer in contact, equation 1 is simply $P = F_a$.

Fatigue Effects—The fatigue strength of a bolted joint must be evaluated two ways: fatigue of the bolt, and fatigue of the bolted material. The properly tightened bolt will not fail in fatigue in a rigid joint. Initial bolt tension will stay relatively constant until the external tension load on the joint exceeds the bolt load. Designers do not permit the calculated service load to be greater than the bolt preload. The bolt will experience no appreciable stress variation, and without stress variation, there can be no failure by fatigue, regardless of the number of load cycles on the joint.

This is not the case where considerable flexibility is present. Variable stress in screw or bolt fastenings increases with the flexibility of the connected parts. If flexibility is too great, the variable stress present may be high enough to cause eventual fatigue failure of the fastener regardless of the initial bolt preload.

The greatest single factor that can eliminate cyclic stress variation due to cyclic loading is proper pretensioning or preloading of the fastener. Test results indicate that rigid members bolted together by relatively elastic bolts offer the best method to prevent fatigue failure [5].

Selection of a Safe Design Load Level

Force Spectrum Analysis The designer is well versed in designing for fatigue resistance. A properly clamped joint is designed so that it will not see forces often enough to cause fatigue failures (see Gasketed Flange Bolting). The problem becomes apparent when a force spectrum of an application is studied. Figure 5 shows a small segment of an aircraft joint strain gage readout. (Falstaff, Fighter Aircraft Load Spectrum for Fatigue.)

The immediate question is, what should be the design

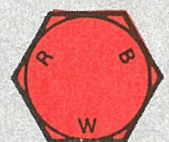
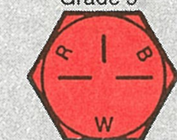
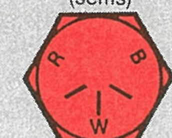
SAE grade and head marker	Nominal diameter, in.	Proof load, psi	Tensile strength, min. psi
 Grade 2	¼-¾ ⅞-1½	55,000 33,000	74,000 60,000
 Grade 5	¼-1 1⅛-1½	85,000 74,000	120,000 105,000
 Grade 5.1 (sems)	⅜ and smaller screw washer assemblies only	85,000	120,000
 Grade 5.2 Low carbon martensite	¼-1	85,000	120,000
 Grade 8	¼-1½	120,000	150,000
 Grade 8.2 Low carbon martensite	¼-1	120,000	150,000

Fig. 3 Allowable stress and head markings.

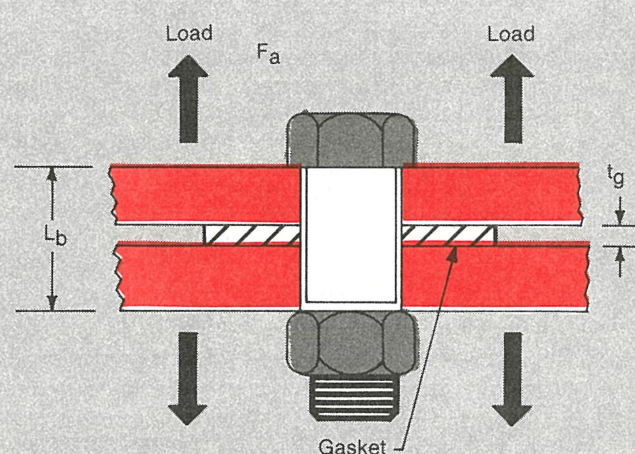
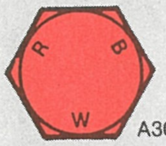
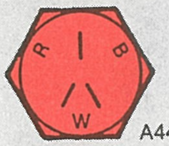
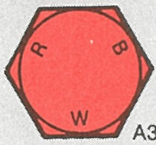
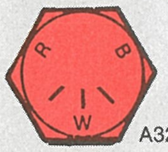
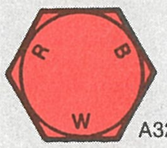
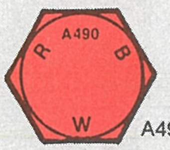
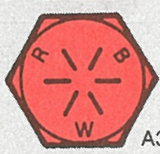


Fig. 4 Externally applied loading in a bolt.

ASTM grade and head marking	Nominal diameter, in.	Proof load, psi	Tensile strength, min. psi
 A307	¼-4 ¼-4	Grade A and B Grade B	60,000 min. 100,000 max.
 A449	¼-1 1⅛-1½ 1¾-3	85,000 74,000 55,000	120,000 105,000 90,000
 A325 Type 1 medium carbon steel	½-1 1⅛-1½	85,000 74,000	120,000 105,000
 A325 Type 2 low carbon martensite	½-1	85,000	120,000
 A325 Type 3 weathering steel	½-1 1⅛-1½	85,000 74,000	120,000 105,000
 A490	½-1½	120,000	150,000
 A354 Grade BD	¼-4	120,000	150,000

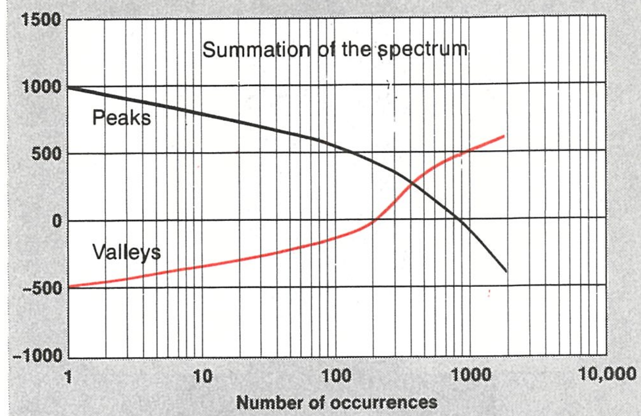


Fig. 6 Spectrum analysis of clamp loading.

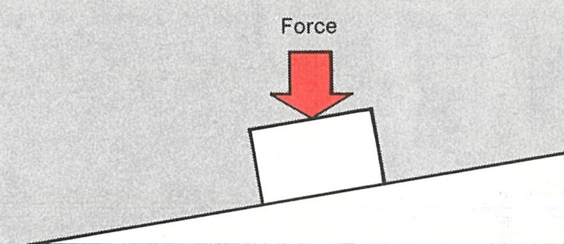


Fig. 7 Transverse sliding.

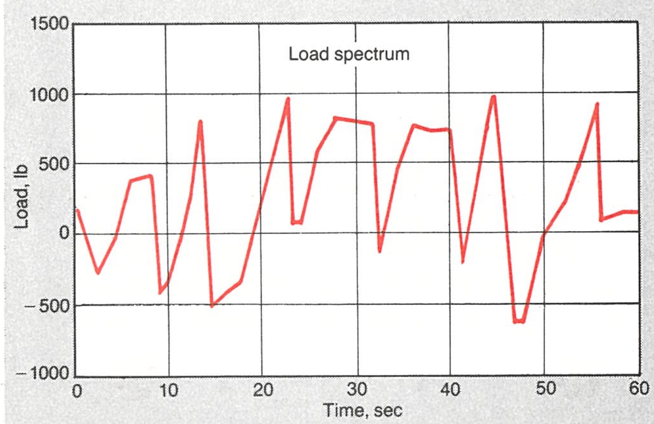


Fig. 5 Readout of an aircraft joint strain gage.

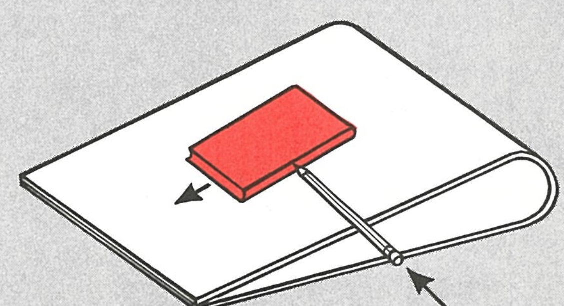


Fig. 8 Simulation of transverse sliding.

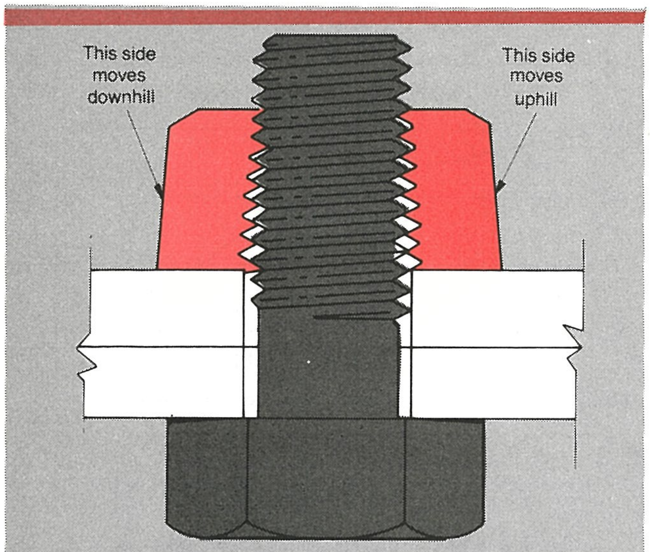


Fig. 9 Ratcheting action of a side-sliding thread.

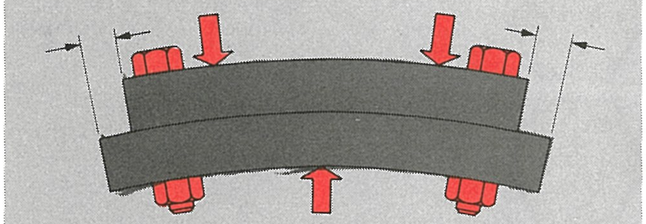


Fig. 11 Bending of a bolted assembly.

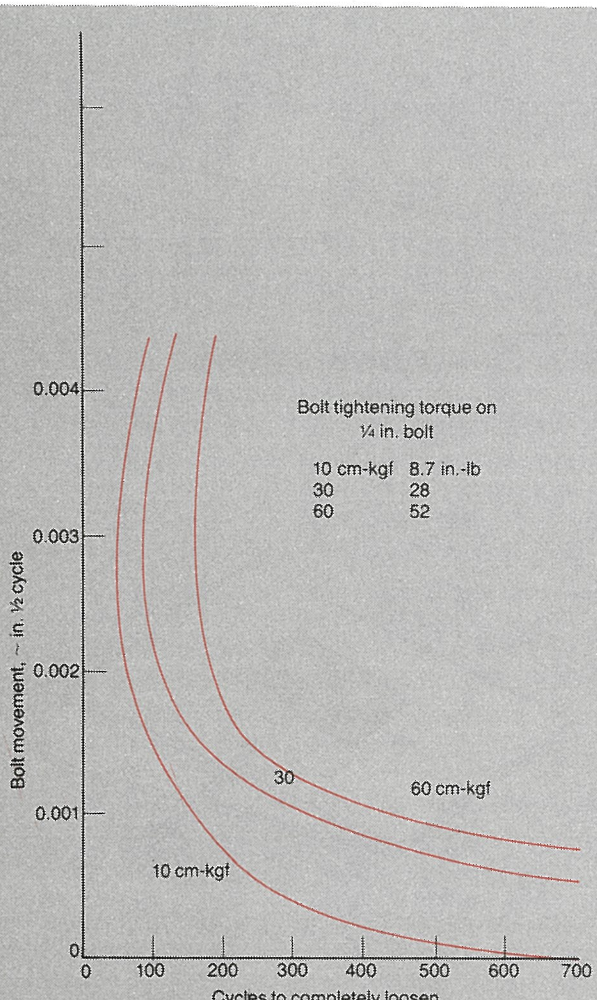


Fig. 10 Bolt movement versus cycles to loosen Haviland slide demonstrator.

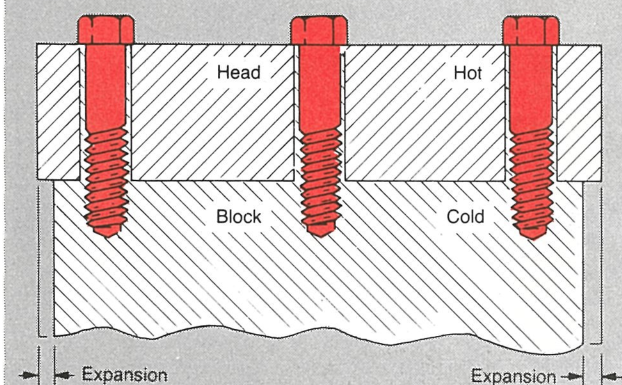


Fig. 12 Differential thermal expansion.

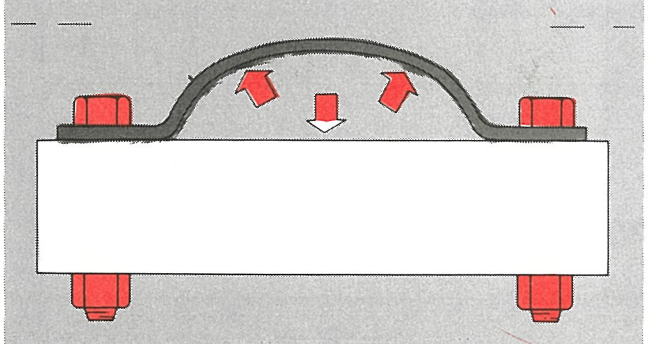


Fig. 13 Internal pressures or functional loads.

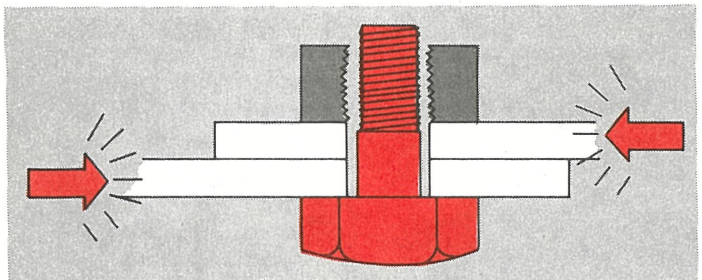


Fig. 14 Shock or impact.

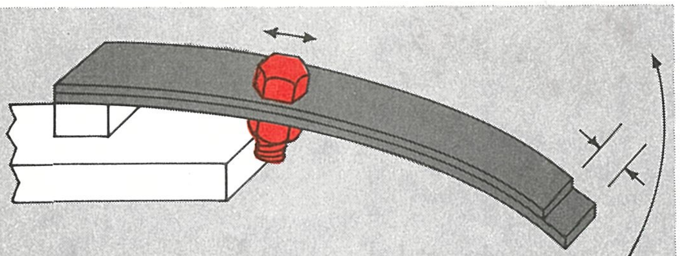
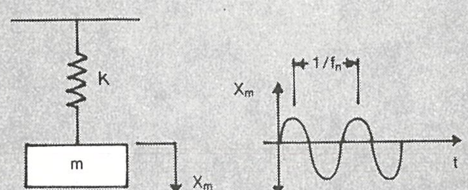
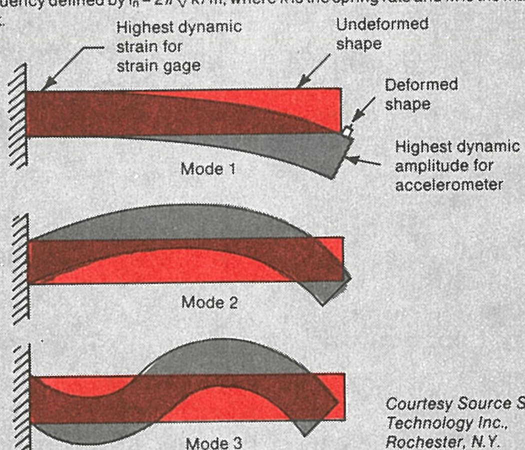


Fig. 15 A simple composite cantilever beam composed of two steel blades bolted together.



All mechanical components have natural frequencies of vibration. In the case of a single mass suspended from a spring, the spring-mass combination has only one natural frequency defined by $f_n = 2\pi\sqrt{k/m}$, where k is the spring rate and m is the mass of the block.



Courtesy Source Stress Technology Inc., Rochester, N.Y.

In the case of a continuous cantilever beam with distributed stiffness and mass, the beam has, theoretically, an infinite number of natural frequencies. Practically speaking, only the first four to six frequencies are of any importance. Associated with each natural frequency is a mode shape, that is, the shape that the beam deforms into as it vibrates at that particular natural frequency. Low natural frequencies have fairly simple mode shapes, while high natural frequencies have more complex mode shapes. When the component is excited, it vibrates at all its natural frequencies and in all its modes simultaneously. Generally, only the lowest few modes have any significant amplitude. The exception to this rule occurs when the component is excited by some periodic excitation with a frequency close to one of the higher natural frequencies.

Fig. 16 Natural frequencies and mode shapes.

load of the joint? We know that a joint clamped tighter than applied loads will not experience cyclic loads on the bolts. Should clamp load be so high that the bolt never sees the 1000-lb peak or can a calculated risk be made to allow a few hundred peaks, which surely could not cause fatigue if below the fatigue limit? The answer is twofold. First, the spectrum should be analyzed for number of occurences. Figure 6 is the result of such an analysis.

Second, another consideration as serious as fatigue must be dealt with. Overloading of a joint will cause self-loosening of the fasteners. Only 50 side slips of a joint will cause 20 percent clamp load loss [6]. As clamp load is lost, lower and more frequent loads cause slipping and failure becomes precipitous. Therefore, the designer should consider the bolt-securing means at the same time the bolt size is being selected. Many securing means affect the clamp load as well as the self-loosening tendencies. If the bolt can be secured against self-loosening, then a few hundred overload cycles will not affect performance. The design load can be a fraction of the peak load with obvious benefits of cost and weight.

The proper thread-securing means can be selected if the loosening process is well understood.

Loosening Tendencies of Bolted Joints

Clamping of Soft Materials The information obtained from Fig. 2 will be satisfactory only for fairly rigid joints. When clamping a "soft" joint such as a gasketed cover, it is often necessary to determine the bolt tightness by the capability of the gasket for supporting the load [4]. This means that if bolts are chosen that are too large, they will be stressed at a very low value. Since a steel bolt elongates 0.001 in. per inch of length for 30,000 psi stress, a lower stress means less elongation. If a gasket shrinks or creeps under load, the load loss is very rapid when stresses are initially low. For instance, a 2-in. bolt stressed at 60,000 psi will elongate 0.004 in. If the gasket shrinks .001 in., it will lose 25 percent of its load. If, however, a larger bolt had been chosen, which gave the same load at 30,000 psi, then the elongation would be 0.002 and if the gasket shrank .001, 50 percent of the load would be lost. The lesson here is to use the longest, smallest bolt possible that will supply the maximum permissible gasket load.

Bolt-locking materials, on first thought, do little good if the clamped parts themselves are shrinking away from the bolt. However, whenever there is shrinkage of gaskets, there will also be increased movement of parts in a sliding and longitudinal motion (relative to the bolt). Therefore a threadlocking material or device is beneficial, preventing catastrophic loosening (see Transverse Sliding below). It also provides thread sealing.

Brinneling of Bearing Surfaces Even "hard" flanges and gaskets can collapse under the clamping load if there are burrs under the head or poor finishes on the threads. A relaxed condition can be produced if hard washers are not used under the bearing face of the nut and/or bolt, whichever is to be turned. The primary function of the so-called lock washer is to provide a hard bearing surface at very small cost. It has virtually no securing function in spite of its name. (See Testing with Transverse Shock.)

Transverse Sliding All standard bolts and nuts are made with a clearance between them to assure easy assembly. Class 2A, the most common, will have from 0.0013 to 0.0114-in. lateral clearance. This means that the nut can be moved sideways by this amount. Now consider that the helical thread is nothing more than an inclined plane with the nut sitting on it, held against sliding by friction.

This condition can be simulated by placing a small memo pad or block onto the side of a slippery book. Tip the book until the pad almost slides. Now try to slide the pad sideways with your finger and watch what happens.

The pad slides downhill every time you push sideways. You don't have to push it downhill. Its weight moves it by itself. This is exactly what happens to a loaded thread made to slide sideways. Furthermore, a side-sliding thread has a ratcheting action. Consider a cross section through the centerline of a bolt and nut.

As the nut is moved into the page, the right side is moving uphill and the left downhill. Obviously, the uphill side will move with greater difficulty and acts as an anchor around which the nut rotates on the left side. If pulled from the page, the left side becomes the anchor and the right side rotates downhill. The next effect is small unwinding motions each time the nut is cycled sideways. Bolt movement versus cycles to totally loosen is plotted in Fig. 10.

From these experiments, and others, we can conclude that if sideways sliding is produced on screw threads, then the threads will unwind all by themselves. The higher the clamping force, the less likely there is to be side movement; but *if side movement occurs,* the force will unwind the threads regardless of its magnitude.

There are three common phenomenon which cause shear or side-sliding in bolted assemblies. It can be caused by bending the assembly (Fig. 11), it can be caused by differential thermal expansion such as exists in the head of an automobile engine (Fig. 12). The differential expansion of a six-cylinder in-line engine has been measured as long as .060 in. total under extreme temperature cycling. Sliding can be caused by internal pressures or functional loads. Shock or impact, which is often observed as "vibration," can cause sliding (Fig. 14).

What About Vibrational Loosening? Tests done at NASA/Goddard (also Ref. 7) on structures under high vibrational loads of varying frequency substantiated the following conclusions:

1 Vibrational energy had little or no effect on the loosening of a bolt unless side sliding of the threads was simultaneously occurring. Then, and only then, vibration helped "grease the skids" and loosening proceeded faster than without it.

2 Vibrational energy had a very great effect on the structures being bolted. If the response of the structure caused bending or side sliding, then bolts loosened the same as they did under slow sliding or single impacts.

One structure tested was a simple composite cantilever beam composed of two steel blades bolted together. Slow movement by hand loosened the ¼-in. bolt after about 100 cycles (see Fig. 15).

Imposition of a vibrational stress on the structure at 1000 Hz and 20 g rms in an electro-hydraulic test ma-

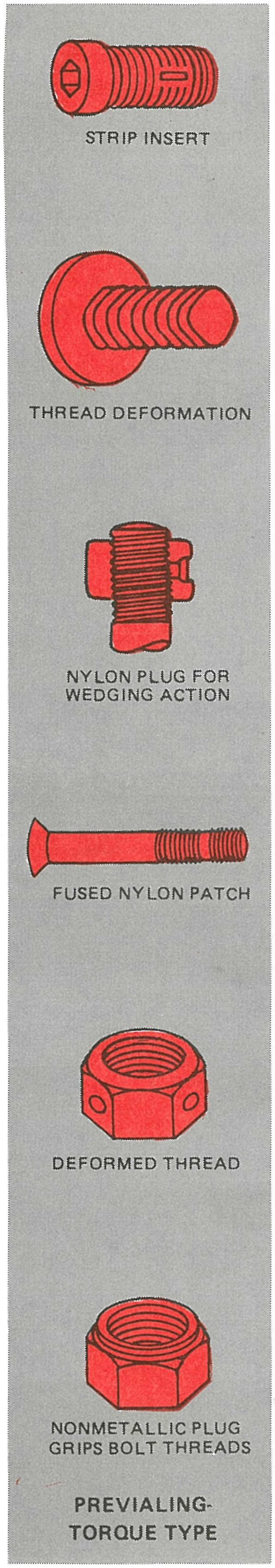


Figure 17

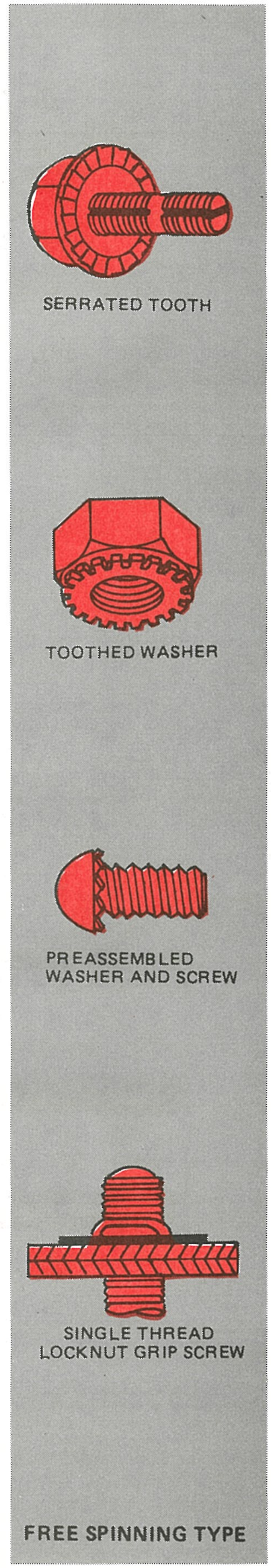


Figure 18

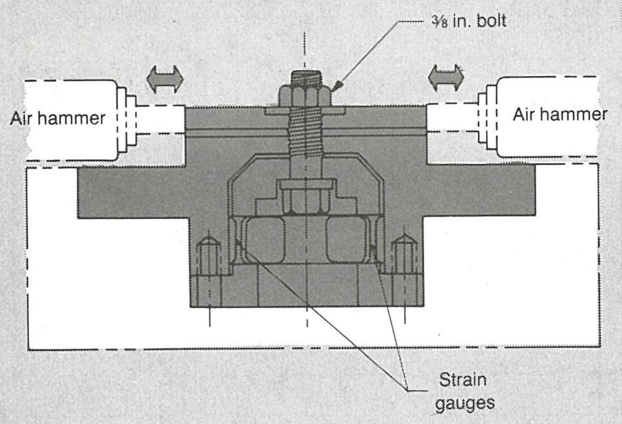


Fig. 19 Test machine.

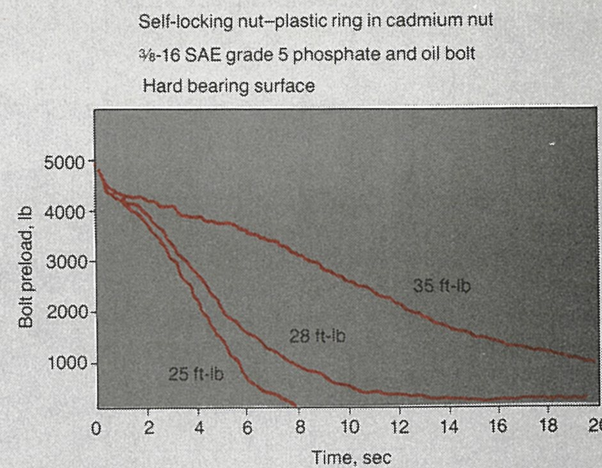
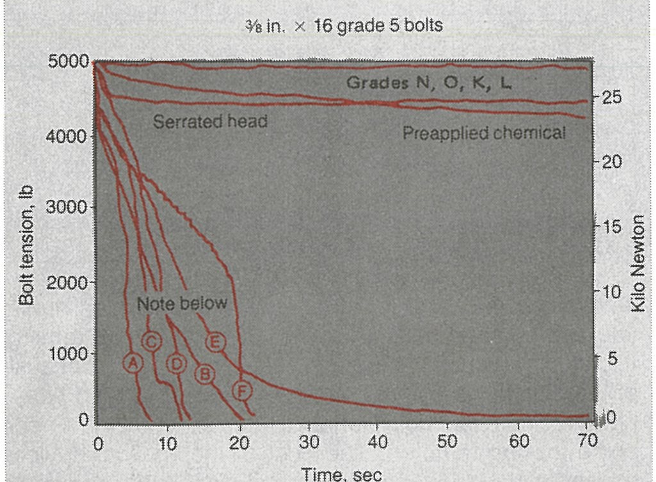


Fig. 21 Transverse shock test.



Note: Test time to reduce bolt tension by 50%

A	Lock washer	5-12 sec
B	Root interference lock nut	6-8 sec
C	Bolt patch	8-12 sec
D	Control—no locking	8-12 sec
E	Distorted lock nut	7-22 sec
F	Nylon ring	16-20 sec

Fig. 20 Transverse shock test.

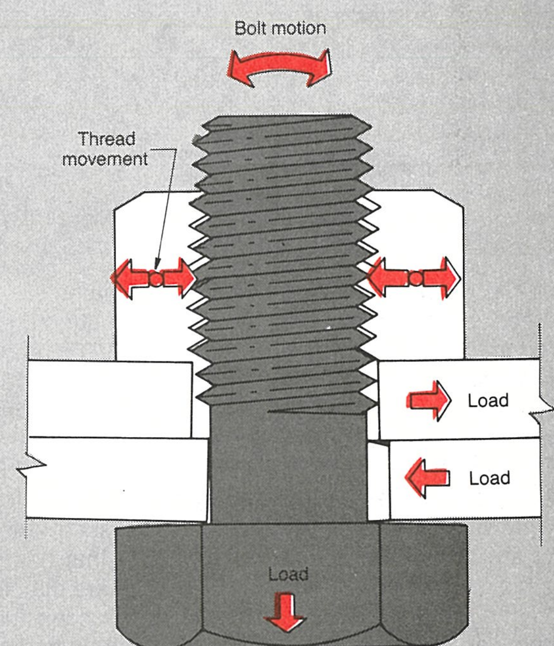


Fig. 22 Filled thread prevents bolt and thread movement.

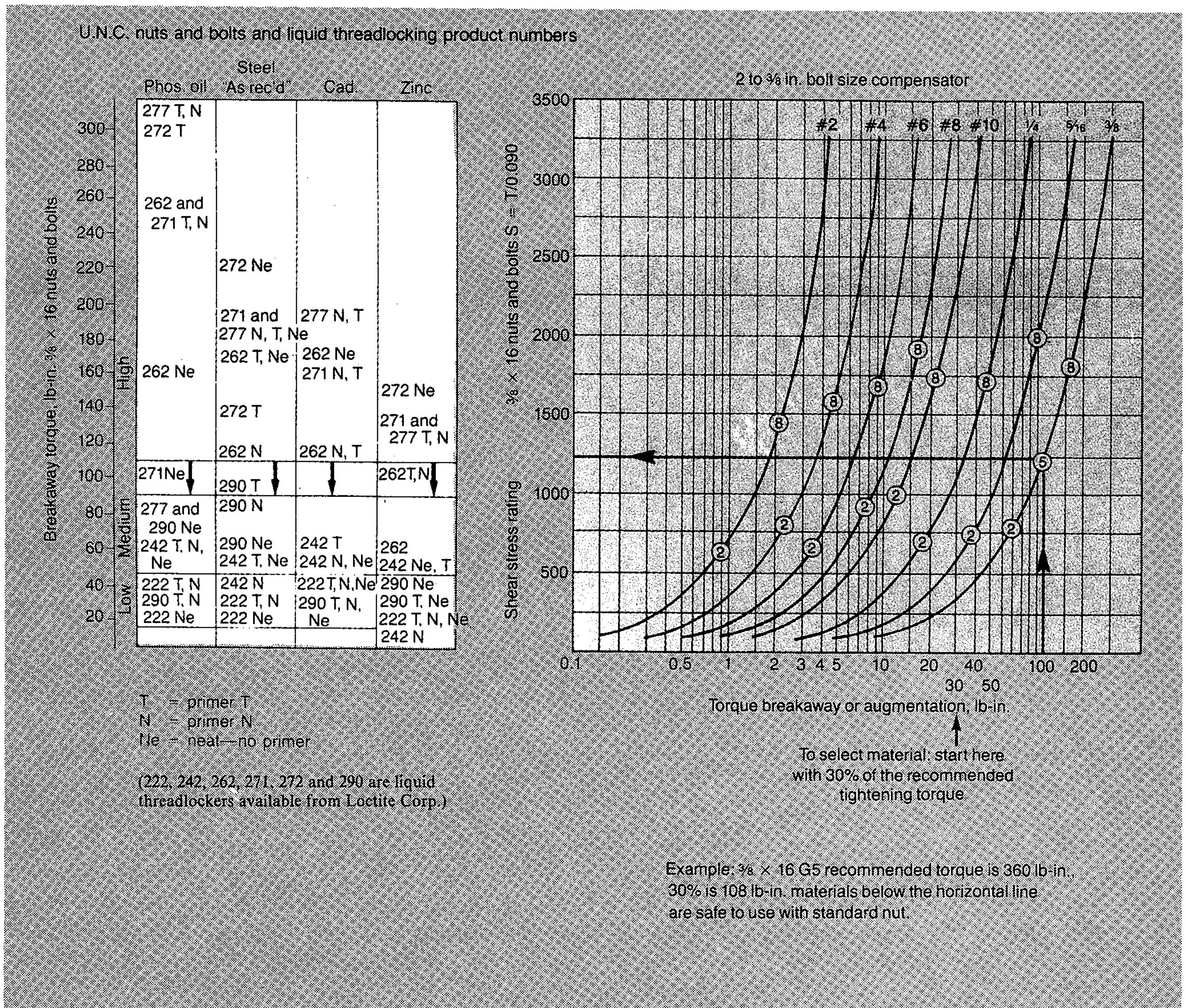


Fig. 23 Thread-locking performance chart.

chine did not produce loosening in 2.5 minutes.

Imposition of a vibrational stress on the structure with mixed frequencies from 200 to 2000 Hz at 10 g produced no loosening, but a 10-g stress at 20 to 400 Hz produced bending as in mode 1 (see Fig. 16) and loosening occurred in 5 to 10 seconds (100 to 200 mode cycles).

Keeping the Joint Tight *Prevention of Thread Rotation* Many mechanical devices have been used to prevent unwinding of nuts and bolts. A few such nuts and bolts are shown to indicate the ingenuity that has gone into this effort (Figs. 17, 18). Most devices are directed at preventing rotary motion and have been successful in a limited way. Worth noting are the nuts with compressible inserts and the nuts with teeth on the bearing surface. The insert is a prevailing torque type, which prevents rotation with a heavy friction drive. By its nature, it resists going on as well as coming off. It does not prevent side motion and, therefore, has a definite limit to its effectiveness. The serrated tooth nut is free-spinning and works by digging into the clamped material. It is very effective in preventing rotation. Unfortunately, the digging doesn't stop at the cessation of tightening, but continues during use and on short bolts causes loosening without turning. It can also cause failure of the clamped part because of the digging, which may initiate cracks. It is also possible for either the bolt or nut to turn so both may need some locking features. The cost of mechanical devices exceeds that of complete thread fillers, as shown in Table III.

Testing with Transverse Shock The most common and easiest way to test a bolted assembly is by artificially induced transverse motion. In the 1960s, G.H. Junkers designed such a machine, which included a sinusoidal sliding motion between two plates clamped with a bolt. He found that as few as 50 cycles would loosen (20 percent) a standard nut and bolt [6]. A similar but simpler machine was designed by a European automobile manufacturer and is the basic machine Loctite uses to evaluate threadlocking materials. A cross-sectional picture of this machine is shown in Fig. 19. It is called a Transverse Shock and Vibration Machine, or more accurately, transverse shock machine. It is fully as capable of rating

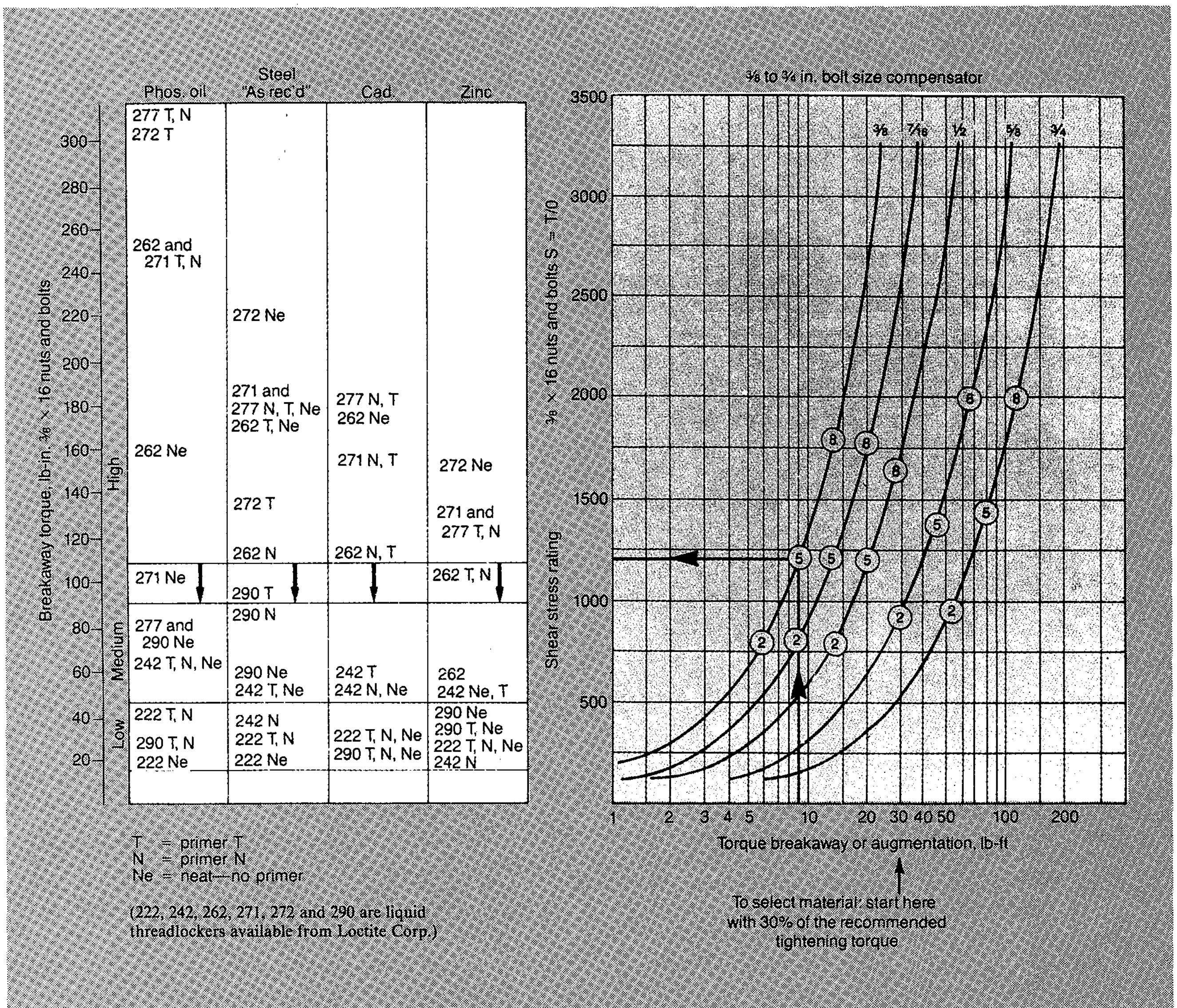


Fig. 24 Thread-locking performance chart.

various locking devices as the Junkers machine, even though it is not as versatile and measures only time to loosen and not stroke, energy, or cycles. It ends up with the same relative ratings and, more important, can be correlated with field results.

The test is severe enough (Fig. 20) to cause failure of most mechanical locking methods. Which doesn't mean that these methods are not useful to a certain degree; however, on a function and cost basis they will be hard to justify.

From a functional standpoint, the most common prevailing types interfere with the proper tightening. Figure 21 shows three individual runs on nuts with a ring-type insert, whereas Fig. 20 shows the averages of several specimens. Note, in Fig. 21, that a 40 percent variation in torque is needed to produce the same target clamp load. Also, the better the antiloosening performance, the less likely the clamp load will have been obtained. The chief features of mechanical devices are reusability, temperature resistance (all metal distorted threads), and inspectability. The relative cost of some of the common methods are shown in Table III. A comparison of features is shown in Table IV.

Prevention of Premature Loosening

The most effective way to prevent unwinding of threads is to prevent all thread movement in any direction. This is accomplished with liquid theadlockers, which fill and cure in all the open space between the threads, thereby preventing:

- sideways thread movement
- rotational thread movement
- tipping thread movement
- dilation thread movement.

As one might expect, this system is the most effective of all, as proven by laboratory tests and field experience.

The selection of a chemical threadlocking material will depend on the following criteria:

1 The ultimate shear strength of the fastener, which must not be exceeded. This is usually important only for screws under 5/16-in.-dia.

2 The severity of the loosening tendencies.

3 The size of the threads, and therefore the viscosity of the material to assure thread filling.

4 The method of application and requirements for testing or putting into service.

5 The environmental requirements of temperature and chemical resistance.

Ultimate Fastener Strength—The disassembly of bolts or screws without fastener damage is usually important. The ultimate strength of soft screws with slotted or Phillips heads should be determined by experiment. The driving system may fail before the shank. Most high-strength fasteners use an internal or external hexagonal drive capable of shearing the fastener without harming the drive system. The ultimate strength of an adhesive holding system can be found as follows:

- Using the recommended tightening torque nomograph (Fig. 2), follow the directions to pick the appropriate tightening torque for your bolt material to give a stress 75 percent of proof load. *Now calculate 30 percent of this tightening torque; this is the correct breakaway or augmentation torque for your bolt size* and for any selected locking material, since the fastener would otherwise break loose at 70 percent of tightening torque.

- Enter the chart on Figs. 23 or 24 at a torque augmentation equal to 30 percent of the tightening torque. This 30 percent torque value will add to the normal loosening torque so that the torque on break loose will be about equal to the original tightening torque. Go up to the curve for your bolt size and left to the columns showing material performance on various surfaces. Select a material that gives a stress within ± 25 percent of the line you have drawn. Use higher strengths for severe situations and lower for the more usual. Product numbers are for Loctite materials. Other materials may be evaluated by the shear stress given on the ordinate axis.

Selection of Material for Loosening Severity—For bolted elements that may see severe occasional overloads, which can cause very rapid loosening, use the strongest material commensurate with the strength of the bolt. Otherwise, pick a medium or low-strength material as these will prevent loosening in most situations. (Ref. TSV results Fig. 20.)

Selection of Viscosity to Assure Thread Filling—Select a viscosity that will apply easily, not run off, and will fill threads that have the maximum clearance (see Appendix I).

Preapplied dry materials are available for application situations where liquids are not acceptable. They are easy to inspect and provide some reuse. The application is usually carried out by a bolt supplier. These are supplied in at least four formulations: low-strength locking and sealing; medium-strength plated fasteners; medium strength; high strength.

Choice of Method of Application and Cure Speed—If quality control checks or functional stress are to be applied soon after assembly, then be sure that enough cure has taken place to avoid failure. Cure speed and breakaway torques can be selected from the manufacturer's product data sheets.

Environmental Requirements of Temperature and Chemical Resistance: Chemical Resistance—With time, temperature, and aggressive environments, the strength of the adhesive may decrease. Pick a stronger material or more resistant material if environments are severe (see Appendix I).

Torque Augmentation

Normal loosening torque of a UNC bolt will be about 70 percent of the torque to which it has been tightened (UNF = 80 percent).

The application of a threadlocking compound adds to or augments the normal loosening torque. The amount that it does this is called *torque augmentation*. This is shown in Fig. 25.

The value of torque augmentation is related to the breakaway torque* and may vary between 70 and 140 percent of the breakaway. For products 222, 242, and 262, augmentation is equal to breakaway.

Most structural fasteners are torqued to at least 75 percent of their minimum yield strength (proof load). To prevent shearing of a locked bolt while being loosened, a locking material should be used that has an augmentation or breakaway that would make the break loose torque roughly equal to the tightening torque.

Therefore, as a design rule to prevent shearing on loosening, select a material so that:

Breakaway = 30 percent of tightening torque, or
Breakaway = 40 percent of normal loosening torque

For a thread engagement of .8 dia. (std. nut) on ⅜ × 16 P & O bolts

Low strength threadlockers give 13 percent augmentation.

Medium strength threadlockers give 25 percent augmentation.

High strength threadlockers give 40 percent augmentation.

Augmentation values are plotted in Figs. 23 and 24 for many bolt sizes and surface finishes.

*Breakaway torque is the torsional strength of adhesive on an untorqued bolt (e.g., pretorque = zero).

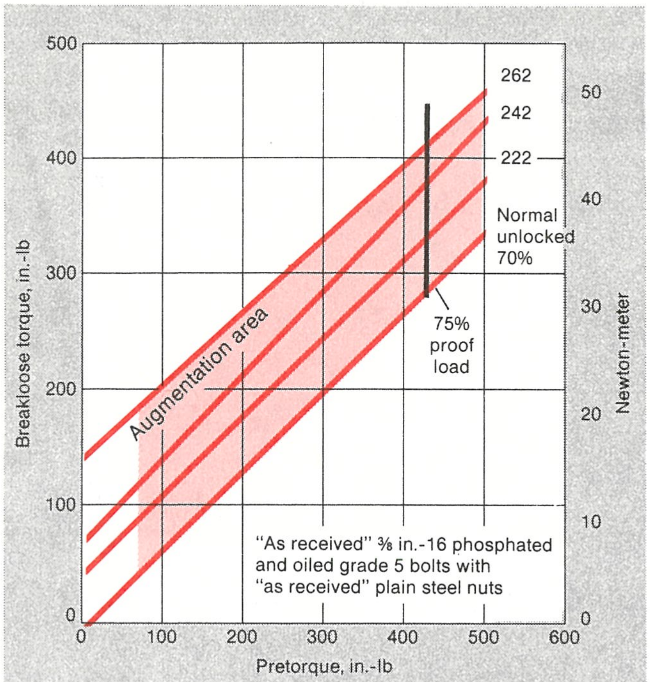


Fig. 25 Torque augmentation.

Table III

Comparative Locking Costs

Type of Mechanical Locking Device	Added Cost per Thousand Over Plain Bolt and Nut
Plastic ring lock nut	125-163%
Deformed nut	187-300%
Plastic patch bolt	130-200%
Serrated head	250-412%
Deformed threads	350-425%
Liquid threadlocker	100%

TABLE IV

Comparative Performance of Locking Mechanisms

	Lubricity	Clamp Load Scatter	Locking Performance	On Torque	Reuseability	Simultaneous Thread Sealing
Liquid threadlockers	Exc.	Low	Exc.	Low	Poor	Yes
Preapplied threadlocker	Exc.	Low	Exc.	Low	Good	Yes
Plastic ring nut	Poor	High	Poor	High	Fair	No
Deformed nut	Poor	High	Poor	High	Fair	No
Plastic patch	Poor	High	Poor	High	Poor	No
Serrated head	Fair	Fair	Good	Low	Good	No
Deformed thread	Poor	High	Poor	High	Fair	No

Hot Strength—Threadlockers, like most organic materials, lose strength at elevated temperatures. Most show good strength up to 300°F (149°C). Hot-strength formulations can increase this temperature to 450°F (232°C) for those applications requiring it.

Application and Equipment—For medium or high production it makes economic sense to automate the application process. Many types of equipment are available that are beyond the scope of this article. Check with an equipment manufacturer who specializes in anaerobic materials for help in automating the process. The equipment pays for itself in both labor and material savings. It also provides a degree of dependability to the process that is difficult to obtain by hand application. ME

References

1 Industrial Fastener Institute, *The Heritage of Mechanical Fasteners,* 1974.

2 Haviland, G.S., "A Logical Approach to Secure Bolting," Soc. of Manufacturing Engineers #AD80-329.

3 e.g., Skidmore-Wilhelm Hydraulic Bolt Tension Tester.

4 *Sealing Technology,* Anderson, Haviland, Tokarski PP16-24, Loctite Corp.

5 "Fastening and Joining," *Machine Design,* ref. issue 1967, Vol. 14, Penton Pub.

6 Junker, G.H., "New Criteria for Self-Loosening of Fasteners Under Vibration," SAE Paper MO 690055, Jan. 1969.

7 Kerley, J.J., "The Use and Misuse of Six Billion Bolts Per Year," Environmental Test and Integration Branch, Engineering Services Div., NASA/Goddard Space Flight Center. Given at 35th meeting of Mechanical Failures Prevention Group, NBS.

8 Steele, J., and Rieger, N., "Coping with Vibratory Stress," *Machine Design,* Sept. 23, 1982, Penton Pub.

Appendix I

Selection of Viscosity to Assure Thread Filling

Material	Viscosity	Suggested Bolt Range	(Class 1) Maximum Diametral Clearance
Low strength thixotropic	1000 cp (1 Pa·s)	#2 to 1/2″	.016 in. (0.4 mm)
Medium strength thixotropic	1100 cp (1.1 Pa·s)	1/4 to 3/4″	.022 in. (0.6 mm)
High strength thixotropic	1500 cp (1.5 Pa·s)	3/8 to 1″	.025 in. (0.6 mm)
Very high strength Newtonian	750 cp (0.8 Pa·s)	3/8 to 1″	.025 in. (0.6 mm)
Very high strength Newtonian	7000 cp (7 Pa·s)	5/8 to 1″ +	.025 in. (0.6 mm)
High strength wicking Newtonian	12 cp (12 mPa·s)	#2 to 1/2″	.016 in. (0.4 mm)

Maximum clearance is computed from Tool Engineers Handbook tables of thread dimensions.
***Primer is recommended to assure cure in gaps.**

Appendix II

Solvent resistance tested per MIL-S-22473-D
Percent retained strength after 30 days at 188°F

Thixotropic Materials

Solvent	Low Strength	Medium Strength	High Strength
Air reference @ 188°F	100%	100%	100%
Motor oil (% of reference)	67	100	100
Water (% of reference)	35	27	100
Glycol/water (% of reference)	27	30	98
Transmission fluid (% of reference)	88	100	100
Gasoline (% of reference)	67	95	86
Skydrol (% of reference)	82	95	78

Newtonian Materials

Solvent	High Strength Thin	High Strength Thick	Ultra Thin
Air reference @ 188°F	100%	100%	100%
Motor oil (% of reference)	70	83	86
Water (% of reference)	110	64	74
Glycol/water (% of reference)	65	59	74
Transmission fluid (% of reference)	95	90	90
Gasoline (% of reference)	50	90	90
Skydrol (% of reference)	70	90	90

Appendix II

I. The Mathematics of Bolt Tightening

A simplified mathematical solution to the torque tension relationship gives results very close to test results. To start the analysis, one 360° thread segment of a bolt is figuratively unwrapped. The fact that the face is tipped 30° from the plane normal to the bolt axis is ignored. A diagram can then be drawn as shown. The block represents an element of the nut bearing against the ramp formed by the unwrapped and flattened thread. Since the system is in equilibrium, all forces, with due regard to direction and sign, will balance another. In other words, all the forces acting parallel to the ramp will sum up to zero and the sum of all forces acting normal to the ramp will equal zero.

① $\Sigma F_{\|} = 0$

$$T/r \operatorname{Cos} \alpha - \mu N - F \operatorname{Sin} \alpha = 0$$

$\Sigma FN = 0$

$$N - F \operatorname{Cos} \alpha - T/r \operatorname{Sin} \alpha = 0$$

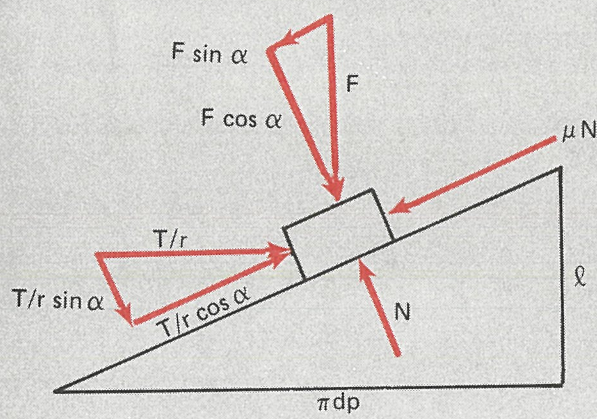
② or $N = F \operatorname{Cos} \alpha + T/r \operatorname{Sin} \alpha$ small value drop

② into ① $T/r \operatorname{Cos} \alpha - \mu F \operatorname{Cos} \alpha - F \operatorname{Sin} \alpha = 0$

$$\text{or } T = r \left(\mu F \frac{\operatorname{Cos} \alpha}{\operatorname{Cos} \alpha} + F \frac{\operatorname{Sin} \alpha}{\operatorname{Cos} \alpha} \right)$$

$$T = rF (\mu + \operatorname{Tan} \alpha)$$

③ in lb-ft $T = \dfrac{dp}{24} F (\mu + \operatorname{Tan} \alpha)$



F = Force applied by the bolt
T = Torque applied to the bolt
N = Normal force on friction surface
μ = Coefficient of friction = 0.15
dp = Diametral pitch
α = Helix angle whose tangent = $\dfrac{\ell}{\pi \, dp}$
ℓ = Lead of thread

Using the Formula ③ $T = \dfrac{dp}{24} F (\mu + \operatorname{Tan} \alpha)$

For 3/8 x 16 UNC*	3/8 x 24 UNF**
F = 5000 lb given	F = 5000 lb given
$\alpha = 3.5°$ $\mu = 0.15$	$\alpha = 2.2°$ $\mu = 0.15$
dp = 0.330 in.	dp = 0.344 in.
F Cos α = N = 4990 lb	F Cos α = N = 4996 lb
μN = 750 lb	μN = 750 lb
T = 14.5 lb-ft	T = 13.5 lb-ft

Unified National Coarse
**Unified National Fine

Conclusion: Fine thread required less torque for same force.

Friction Force Under the Head

Again, with a 5000 lb preload and assuming effective bearing diameter of the nut of 0.400 in., the torque required to overcome the bearing friction is

T = moment arm x force
de effective diameter of bearing surface (3/8 = 0.40)
T/r = μF

$$T = r\mu F = \frac{de\mu F}{24} = \frac{0.4}{24} \times .15 \times 5000 \text{ lb} = 12.5 \text{ lb-ft}$$

Total Torque

UNC		UNF	
14.5	54%	13.5	52%
12.5	46%	12.5	48%
27 lb-ft		26 lb-ft	

Loosening Torque

In a similar manner, loosening torque can be computed.

Again for F = 5000 lb and μ = 0.15

Thread loosening T	UNC	UNF
	3/8 x 16 = 6.0 lb-ft	3/8 x 24 = 8.0 lb-ft

Conclusion: 1) Fine thread higher
2) Less than 60% of on-torque

Total loosening (add 12.5 for head)

	UNC	UNF
Total loosening	18.5 lb-ft	20.5 lb-ft
Now tightening torque was	27 lb-ft	26 lb-ft

Conclusion: UNC loosening torque is 70% of tightening torque
UNF loosening torque is 80% of tightening torque

If one assumed that the screw thread was 100% efficient and there were no friction, then the torque to produce a 5000-lb load would be:

	UNC		UNF	
	4.19 lb-ft	15%	2.7 lb-ft	10%
or to induce preload		15%		10%
to overcome thread friction		39%		42%
bearing surface friction		46%		48%

Conclusion: Friction is the key factor using up 85 to 90% of the total input.

NOTES