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SOME PROBLEMS OF FATIGUE OF BOLTS AND BOLTED JOINTS IN AIRCRAFT APPLICATIONS

LEONARD MORDFIN

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

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Leonard Mordfin

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by

Leonard Mordfin

SUMMARY

The profuse variety of aircraft bolts which is available has made the evaluation and specification of bolts for engine and structural use extremely complex, particularly insofar as fatigue and hot fatigue environments are concerned. The state of knowledge of fatigue of bolts and bolted joints is surveyed and critically appraised in terms of aeronautical practices. Using this material as a basis, recommendations are made regarding the evaluation and specification of aircraft bolts for fatigue situations and regarding the growing problem of errors in fastener replacement.

1. INTRODUCTION

The life expectancy of aircraft bolts is generally comparable to that of the fatigue-sensitive structures in which they are employed. Nevertheless, the Aerospace Industries Association estimates that the use of bolts in aircraft joints will decrease by about two-thirds over the next ten year period. There are many reasons for this anticipated trend; one of the most significant stems from the fact that the selection of the proper bolt for the severe environments associated with hypersonic flight is exceedingly complex.

To a large extent, the rapid development of aircraft bolts represents a tribute to the manufacturers, whose advances during the last decade have been considerable. Many of these advances, however, have been made at the expense of standardization since bolts for special purposes can be made far stronger than bolts for general use. This, together with the fact that ours is a competitive economy, has resulted in the manufacture of a staggering number of different bolt types, each costing considerably more than its general purpose antecedent. When the designer, specifier or purchaser is confronted with this situation, knowing that the bolts were developed primarily (although not exclusively) for greater static strength, his task of specifying bolts for particular fatigue applications becomes rather formidable.

A project was undertaken, therefore, to survey available knowledge on the fatigue properties of aircraft bolts, in the hopes that with this foundation, answers could be found for many of the perplexing questions which arise when attempting to prepare specifications for bolts to be used in fatigue applications.

While the above survey was being carried out, another investigation was in progress, the first phase of which was set forth as "... a survey on structural fatigue at elevated temperatures to establish current requirements and thinking in this area. Such survey shall be an extension of work previously performed*... The materials, fasteners, and configurations which are currently used, and are likely to be used, for skin-to-spar cap type connections in the heat-affected zones of supersonic aircraft shall be determined..." The purpose of this phase of work was to provide a basis for designing suitable specimens for a test program.

For obvious reasons, it was considered desirable to incorporate some of the results of the latter survey into the present report, particularly with regard to general design principles for increasing the fatigue strength of joints.

The purpose of this report, then, is to present a survey of available knowledge on the fatigue of bolts and bolted joints, and to use this information to discuss various aspects of the general problem of specifying bolts, materials and configurations for aircraft fatigue applications.

The survey itself turned out to be a more Herculean task than was originally anticipated. It is estimated that well over a thousand papers, reports and articles have been written on the subject. Fortunately, a number of excellent surveys have already been prepared (refs. 2 through 8) which cover most of the pertinent material written prior to 1955. The present effort, therefore, was largely devoted to reviewing the wealth of material written since that time. To this end, approximately 300 publications and reports were studied. As might be expected in so prolific a field, the duplication is sizeable. For this reason, it is felt that little would be accomplished by the listing of a complete bibliography. The following referencing system has, therefore, been adopted. Refs. 2 through 8 shall be considered basic and applicable to this entire report. In addition, several supplementary references are given for each section or subsection of the text. These are intended merely to introduce the reader to further information on the subject and actually represent only a fraction of the material which is reviewed in the text.

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*See reference 1

In presenting the results of the survey, it was considered desirable to divide the subject into numerous sections and subsections. Considerable overlap exists, but this is unavoidable. Two main sections are devoted, respectively, to fatigue of bolts as such, and to fatigue of joints. These are further divided into subsections dealing with various aspects of the problems. Although the author has taken the liberty of inserting his comments throughout the report, a third main section has been reserved exclusively for this purpose. Two more main sections are devoted to the effects of preloading of bolts and to temperature effects since these apply largely to both bolts and joints and could not be treated fully under either of the latter categories. The reader is referred to the table of contents for an outline of the report prior to further study.

Much of the material presented deals with means of improving the fatigue strengths of bolts and joints. In some cases this material is based on a limited amount of experimental evidence. The designer is cautioned, therefore, to regard this information as introductory rather than as directly usable data unless suitable check testing is done.

2. EFFECTS OF THE PRELOADING OF BOLTS ON THE FATIGUE STRENGTH OF BOLTS AND BOLTED JOINTS

(Supplementary references: 9 through 16)

The most important factor among those which influence the fatigue strength of bolts and bolted joints is preloading of the bolts. The ability to preload bolts by the application of a high torque makes it possible to increase the fatigue strength of a bolted joint over that of an otherwise comparable hot or cold riveted joint.

The application of a preload to a bolt raises its mean stress but, for a given applied dynamic load, it reduces the effective cyclic stress amplitude felt by the bolt. In most bolting steels, fatigue life is more sensitive to stress amplitude than it is to mean stress, and hence fatigue life can be greatly increased by preloading. Increases of 50 to 250 percent are, in fact, common. Also, scatter is notably reduced. The benefits of preload increase with the amount of preload, and use of preloads up to 90 percent of the yield strength of the bolt are generally desirable. In no case, however, should the magnitude of the preload be sufficient to indent the joint material or, in combination with the external loads, to create a crack at the thread root.

Stress analyses* reveal that, for a given preload, the increase in fatigue life depends upon the relative stiffnesses of the bolt and the assembly, which includes the bolted materials, the nut, and all other parts which are elastically deformed. It is desirable that the bolt have

*See, for example, the derivations in references 8 or 12 as low a stiffness and the assembly as high a stiffness as possible. This fact should be borne in mind in the discussions of individual design features which follow.

Two practical factors limit the attainment of maximum benefits from preload. One is general ignorance of the exact magnitude of the preload being applied. The second is the gradual loss of preload which is experienced during use, resulting from loosening of the nut, wear of mating surfaces, stress relaxation, etc.

It has been estimated that only about a tenth of the torque applied to a bolt goes for preloading, the remainder being required to overcome friction. At least one manufacturer (ref. 16) states that when friction conditions are "normal" the applied torque should equal about twenty percent of the product of the bolt diameter and the desired preload. Average deviation has been quoted as seven percent, but deviations of the order of twenty-five percent are apparently not uncommon with aircraft bolts.

In special field applications gaged bolts, such as the proprietary Pic, Strainsert and Tru-Load bolts, can be used to give indications of preload. A more common but less versatile device for this purpose is the so-called preload indicating washer.

Periodic retightening of the bolt is required to restore preloads which have diminished during use. The rate of loss of preload varies inversely with the magnitude of the preload, and, for a given preload, the rate of loss varies inversely with the torque required. Sealing compounds, fabrics, rubber, or a heavy build-up of paint on the faying surfaces are undesirable since these extrude and leave a loose joint. Sometimes, however, they are necessary to minimize fretting, as will be shown later. The use of spring washers to maintain preload is a controversial practice since they also reduce the stiffness of the bolted assembly.

Preloaded bolts are preferable even when the bolt is subjected to working loads in shear. The clamping forces resulting from preloaded bolts reduce slip in the joint, and greater portions of the working loads are then carried by plate friction. This reduces bending of the bolts which is severely detrimental to fatigue life, particularly when the bending energy must be absorbed in the threaded portion of the bolt. With sufficient friction,the bolt will not feel small load fluctuations which would otherwise cause fatigue damage. Preload benefits of this type are greatest with thin sheet joints where the amount of load transferred by friction can be made large compared to that transferred by the bolts.

The eventual fatigue fracture of a joint which carried most of its load by friction is, perhaps, difficult to visualize. For this, the reader is referred to the excellent photograph in Fig. 2c on p. 1355 of ref. 9.

Special wrenching recesses, such as the patented Torq-Set and Hi-Torque recesses, have been designed to permit the application of high preloads to shear bolts which do not usually have very large heads. See, for example, bolts with flush countersunk heads, such as the NAS 1151 and 1220 series.

Several exceptions to these general rules should be pointed out. When the external fatigue loads are of an impact nature rather than smoothly cycled, high values of preload act to reduce the fatigue life of the bolt (ref. 14). Under certain conditions it is desirable to forego the benefits of preload in order to minimize the possibility of stress corrosion. In materials in which fatigue life is relatively sensitive to mean stress, preloading may be of questionable value.

Finally, the designer is cautioned against using bolt preload as a design parameter unless a foolproof technique for maintaining the desired amount of preload can be devised. Rather, bolt preload should be looked upon as a bonus value; a means for achieving reliability through fatigue lives which are greater than those actually specified.

3. FATIGUE OF BOLTS

Virtually all fatigue fractures of bolts occur in one of three places:

(1) At the first thread, that is, at the intersection of the thread with the shank

(2) At the first working thread, which is the first thread engaged by the nut

(3) At the shoulder, where the shank joins the head.

Increasing the fatigue strengths at these critical locations involves the use of proper thread shapes, materials, fabrication processes, nuts, fillets and other devices which minimize stress concentrations and notch sensitivities. These subjects are covered in the subsections which follow.

3.1 Shape of Bolt Threads

(Supplementary references: 17 through 24)

a. Root radius

This is the most important of the geometric considerations which affect the fatigue strength of threads. Fatigue strength increases with root radius. A V-bottomed thread gives notoriously poor fatigue strengths and the flat-bottomed American National thread is not much better. The Unified thread (MIL-S-7742A), which is the basis for most present-day aircraft bolt threads, permits the thread roots to be rounded, either intentionally by the use of a rounded tool or inadvertently as a result of tool wear. MIL-B-7838A improves the situation by actually specifying minimum acceptable root radii. Specification MIL-S-8879 goes a step further by requiring even greater root radii which reduce the thread depth to only 75 percent of the basic value instead of the 83 percent allowed by MIL-S-7742A. Presently in the developmental stage is a still newer modification which provides for still greater root radii that reduce the thread depth to only 55 percent of the basic value. This feature is particularly desirable for extremely notch sensitive materials such as beryllium.

b. Flank angle

The included flank angle for most standard thread systems is either 55 or 60°. Tests have shown that varying this angle from 45 to 65° does not affect fatigue strength significantly. However, increasing the angle to 90° has been shown to prolong fatigue life significantly. This feature is not now employed in aircraft bolts.

c. Depth of Engagement

In the Unified thread system, the depth of engagement of mating thread surfaces is nominally about 88 percent of the bolt thread depth. Within limits, truncation of the bolt thread crests does not affect its fatigue strength. However, a combined truncation of both the bolt and the nut threads, giving a depth of engagement of only 25 percent of the normal value, increases the fatigue strength by some 40 percent. The improvement is attributed to the better distribution of the load among the nut threads. This is most effective when the nut material is softer than the bolt material and when the bolt thread crests are rounded off following truncation.

d. Pitch

Cursory investigations have often led to the conclusion that decreasing the pitch of a thread raises its fatigue strength. However, it is now fairly well established that the beneficial effect resulted from the fact that smaller pitches are generally accompanied by larger minimum diameters. If the minimum diameter is held constant, fatigue strength is actually reduced slightly by decreasing the pitch.

e. Taper

Tapering bolt threads in a direction similar to pipe threads reduces fatigue strength. Tapering in the reverse direction, although generally impractical except in the case of specific studs, increases fatigue strength when used with untapered nuts.

f. Lok-threds

An entirely different, and less common, approach to thread design for fatigue strength is achieved by the use of threads which, in effect, create force fits. An example of this is the patented Lok-Thred. This thread is flat-bottomed, like the American National, but the minimum diameter is larger, the flat bottom is wider, and it is tapered at 6° to the axial direction. In cap screw or stud applications or with nuts softer than the bolt material, the Lok-Thred is reputed to reform the female thread, cold work it, and induce compressive stresses in it. This locking fit, which is contended to be replaceable, is said to eliminate cyclic wear on the mating surfaces and to counteract the tendency of the preload to lessen. The result, according to the manufacturer, is a high fatigue life for both the bolt and the female part. Available test data are inadequate to verify these claims.

3.2 Thread Forming

(Supplementary references: 19, 20, 25 through 32)

Two processing factors which are most important for high fatigue strength in threads are smooth surface finish and the development of surface compressive stresses. The best way to obtain both of these conditions is by roll-forming the threads after the bolt blanks have been heat treated. Threads formed by rolling have far greater fatigue strengths than machined threads, whether they be turned, milled or ground. Thread rolling tends to produce a maximum thread diameter which is greater than the diameter of the original bolt blank. However, many aircraft bolt specifications forbid thread diameters which exceed the shank diameter. Hence, the length of the blank which is to be threaded is frequently reduced in diameter prior to thread forming.

If gross decarburization develops during the heat treating process, the decarburized layer must be ground off prior to thread forming. If the heat treating process involves a quench hardening, a tempering treatment is desirable to relieve the quenching stresses. With carbon steel bolts, it is often more desirable to substitute work hardening for the heat treatment prior to thread forming. Heat treating after thread rolling should be avoided since this relieves the beneficial compressive residual stresses.

It is interesting to note that thread rolling cold works the surface layer in addition to creating compressive stresses. In a sense, this is undesirable since it raises the effective stress concentration factor. The net effect, however, is decidedly beneficial. In some materials the rolling process produces fine cracks in the surface. If the compressed layer is thicker than the crack depth, and if the material has a low rate of crack propagation, then the fatigue resistance is nevertheless improved. This, however, limits the degree of rolling that is beneficial for a given material.

Most aircraft bolts are manufactured with rolled threads despite the fact that most bolt procurement specifications do not require this. The reason is that rolling is a rapid means of thread forming and also an economical one when the volume of production is high enough to warrant the special equipment and tools needed. In the event that a small quantity of special bolts is required, the specification of rolled threads can be omitted unless fatigue failure in the bolt threads is anticipated. Shear bolts for specific pure shear applications, for example, do not necessarily require rolled threads since, with the proper grip length, failure will occur in the unthreaded shank. Although rolling increases the fatigue strength of threads, it does not increase the static strength, and hence thread rolling does not influence the preload capacity.

Other methods for producing residual compressive surface stresses are available when roll-forming is impractical. One method, of course, is to roll the thread roots alone after the machining process. Other methods are carburizing, nitriding and cyaniding. Special care must be taken with carburizing since, under certain circumstances, the residual surface stresses achieved with this technique may be tensile, rather than compressive. Also, the quenching process which follows carburizing may form cracks. Nitriding, on the other hand, virtually always produces a compressed crack-free surface. The disadvantage of case-hardening methods, regardless of the depth of the case, is that they increase the brittleness of the surface of the part.

Irregularities in thread surfaces are eventually smoothed out by alternating loads and result in a slackening of the preload. For this reason, smooth thread surface finishes are desirable for maintaining preload. Most aircraft bolt specifications which call for rolled threads, such as MIL-B-7838A, require that the surface roughnesses of the thread flanks and roots not exceed 32 microinches (measured according to MIL-STD-10A). Certain other aircraft bolt specifications, such as NAS 583, also cite this roughness value, and therefore implicitly require that the threads be rolled, since lathe-cut threads commonly yield roughnesses of the order of several hundred microinches. While grinding can also produce very smooth thread surfaces, careful techniques are required to avoid introducing residual tensile stresses at the same time.

Regardless of the thread forming process used, careful inspection is necessary to detect flaws. A systematic program of tests conducted in a direct stress fatigue machine showed that it is important to detect longitudinal flaws as well as transverse flaws. Longitudinal seams, laps and inclusions in aircraft bolts reduce fatigue life as much as thirty percent.

The fatigue strengths of stress-relieving grooves (see Section 3.3) are also improved by rolling. Alternatively, shot peening can be used to develop compressive stresses in the groove surfaces.

3.3 Improving Fatigue Strength at the First Thread

(Supplementary references: 31 and 33)

In terms of fatigue strength, a continuous thread is superior to a series of circumferential grooves having the same pitch. Furthermore, the series of grooves is superior to a single circumferential groove because the interference of adjacent stress concentrations results in a cancelling effect.

In spite of this effect, threads are notoriously fatigue sensitive. In one series of investigations it was found that the shank diameter of bolts could be reduced to three-fourths of the root diameter of the threads (57 percent of the area) and fatigue failures still took place in the threads. In fact, the result was an improvement in fatigue strength for reasons which are discussed later.

The first thread on a bolt does not benefit from the presence of interfering stress concentrations on both sides of it. It is, therefore, one of the three most common sources of bolt fatigue failures. Special efforts must be made to reduce the stress concentration at the first thread, and it is usually imperative to keep this thread away from the nut, which introduces stress concentrations of its own.

A common practice for reducing stress concentrations at the first thread is to run the thread out gradually. This process generally raises fatigue strength to satisfactory levels but may not be sufficient if the bolt is subjected to bending. When bending of the bolt is expected, it is often desirable to undercut the shank with a smooth groove just beyond the threads to a depth slightly greater than the thread root. The smooth groove has an inherently lower stress concentration factor than a thread and exerts a mitigating stress interference effect on the first thread. Furthermore, bending energy is now absorbed over the length of the undercut, whereas it is otherwise primarily absorbed in the threads. Some investigators report that in the presence of a stress-relieving groove it is permissible to engage the first bolt thread with the nut so long as this thread is kept away from the seating face of the nut. This is done by torquing the nut beyond the first thread so that it projects over the groove.

When severe bending is anticipated, a number of undercuts should be employed to provide flexibility. Sufficient spacing should be allowed between the undercuts to permit the bolt to bear on the bolt hole surface properly, and to provide a full cross section at the faying surface. This type of bolt thus becomes a specialized part, suitable only for its particular application. See, for example, Fig. 1.

A procedure which has been suggested as an alternative to undercutting involves drilling down the center of the shank from the head to a point just before the first thread.

Most standard aircraft bolts do not employ stress-relieving grooves or drilled shanks, but rely rather on gradual thread runout. The only notable exceptions are the NAS 563 series and shoulder bolts such as NAS 1297. The slightly reduced shank diameters available on some engine bolts such as the MS 9033 series can hardly qualify as undercuts. Nevertheless, fatigue failures of aircraft bolts at the first thread have become a rarity; this attests to the suitability of the thread runout techniques.

3.4 Effect of Nut Design

(Supplementary references: 14, 17, 21, 31, 34 through 40)

The fatigue strength of engaged threads in tension is only about two-thirds of that of unengaged threads. This situation, is due, in part, to the fact that engaged threads do not share the load uniformly. In a bolt-nut combination, the first engaged thread carries the greatest share of the load, and following threads carry smaller and smaller shares. The result is that most fatigue failures of bolts occur in the first engaged thread.

It is obvious that a nut design which will provide a more uniform distribution of load over the working bolt threads will generally increase the fatigue strength of the bolt. Hence, while a thorough discussion of nuts is beyond the scope of this report, the subject must be included to some extent because of its importance in bolt fatigue.

Many nuts designed to distribute bolt load more evenly have been produced and many more have been proposed. Most of these involve reducing the effective stiffness of the nut relative to the bolt. It will be recalled that, to obtain maximum benefits from preload, the nut should be stiff. However, the improved load distribution obtained with a less stiff nut generally outweighs the reduction in preload benefits which accompany it. Some alternative principles of nut design and selection which have been set forth to improve load distribution among threads are listed below. These may be used individually or in combinations.

(1) Use nuts having a lower elastic modulus than the bolt.

(2) Use nuts having a lower yield strength than the bolt.

(3) Undercut the working face of the nut (see Fig. 2a).

(4) Taper the nut threads.

(5) Use a slightly greater pitch for the nut threads than for the bolt threads.

(6) Use a tapered lip on the nut (see Fig. 2b).

(7) Use a concave or convex working face on the nut with a suitably matched washer (see Fig. 2c).

A commercially available nut having a so-called "Equa-stress thread form" incorporates items (4) and (5), above, and reportedly increases aircraft bolt fatigue life by more than one hundred percent.

Another patented product employs a collar around part of the nut to exert a radial pressure. According to the manufacturer, this diffuses the stresses in the nut and the bolt more uniformly and permits the application of greater preloads without exceeding allowable stresses in the threads.

The addition of a second or even a third nut to a bolt-nut combination is desirable. Tightening the second nut against the principal nut reduces the stress at the first working thread. The fatigue benefits so derived increase with the internut torque applied.

The thread length of the nut should be at least half the bolt thread diameter to prevent stripping of the threads. Increasing the thread length beyond this point may increase the static strength of the bolt, but only adds to its fatigue strength in that its preload capacity may have been raised. Available evidence on the question of how far the bolt should protrude beyond the nut is contradictory.

Under-tightening of aircraft bolts frequently occurs because a castellated nut is adjusted simply to enable the split-pin to be inserted. This error would be much reduced if two suitably oriented holes were drilled in the bolt end instead of one.

It was formerly believed that forcing materials, such as nylon pellets or hemp packing, into threads increased fatigue life. Recent evidence, however, shows that non-metallic locking elements generally disturb the load distribution and reduce fatigue life. The sole advantage of a non-metallic locking element is that it resists complete loss of the nut by vibrations when the preload has completely disappeared.

In spite of all the work that has gone into nut development, most tensile fatigue failures of aircraft bolts occur in the engaged threads. It is clear that further increases in the tensile fatigue properties of bolts will have to come from advances in materials, thread shapes, thread processing techniques and/or nut design.

3.5 Bolt Head and Shoulder Design

(Supplementary references: 21, 29, 41 through 44)

Fatigue failures of aircraft bolts occasionally occur at the juncture between the bolt head and the shank. This danger is minimized by using as large a fillet radius as possible, and by properly machining the underside of the bolt head so that preloading is truly axial.

Where a large fillet radius cannot be used, several alternatives are available. One is the use of a reentrant fillet. This has the added advantage of providing a more flexible bolt for greater preload benefits and also reduces offset loading tendencies. A second alternative is the use of an elliptical fillet which is sometimes superior to a circular fillet of equivalent size. As a last resort, the shoulder can be flame hardened; this results in a stress distribution similar to that obtained with a fillet.

For greatest fatigue resistance, the bolt head should be forged to dimensions closely approaching the final dimensions in order that the favorable alinement of the metal fibers may be retained after machining. Also, the fillet should be cold rolled. The following data compare fatigue lives for several bolt head manufacturing processes.

Process	Average cycles to failure
Hot forged, with rolled fillet	100,000
Machined from bar stock	25,000
Cold upset with machined socket	20,000
Hot forged complete	17,000

The fatigue strengths of conical head bolts generally are lower than those of bolts with conventional heads, particularly when deep wrenching recesses, such as Phillips or Reed and Prince recesses, are involved. The head-to-shank juncture is notably fatigue sensitive, and careful machining of both the head and the countersunk hole is required to insure that the cone axis is alined. This minimizes bending stresses in flexible joints and provides proper bearing surface.

A recent commercial development is the self-alining nut which is a ball-and-socket type of nut-and-washer arrangement. The combination is reported to correct offset loading tendencies up to eight degrees when bolting non-parallel surfaces.

When high strength bolts are used on soft materials, it is desirable to employ hardened steel washers to permit greater preloading without brinelling. Thick washers should also be used under bolt heads and nuts when the holes are oversized, in order to better distribute the clamping forces. Experimental evidence shows that hardened steel washers do not reduce fatigue strength. Nevertheless, in the interest of simplifying assemblies, at least one aircraft bolt manufacturer now produces a line having extra-wide heads so that washers are not required.

3.6 Bolt Materials

(Supplementary references: 24, 28, 45 through 47)

The desirable properties for bolt materials are: Low notch sensitivity, medium damping capacity, high stress relaxation resistance, high endurance limits, and high tensile strength. Examples of classes of materials which meet most or all of these requirements are: Low carbon steels, nickel steels, Cr-Ni and Cr-Ni-Mo steels, nitrided steels, and 2017 and 2024 aluminum alloys.

Another important consideration is that notch sensitivity generally increases with material hardness. Hence, heat treating to higher strength levels is not beneficial if the notch sensitivity is increased as well. As recently as 1946, it was believed by some that no fatigue advantage could be gained by using steels with strengths in excess of 140,000 lb/in^2. Today, however, by maintaining a proper ratio between the yield and tensile strengths, and by using thread configurations which minimize the stress concentrations at the thread roots, manufacturers are producing fatigue resistant bolts having strengths up to 300,000 lb/in^2.

Most aircraft bolts continue to be made of steel although in recent years greater numbers of non-ferrous bolts have been produced. Low alloy steels such as AISI 4340 and 8730 predominate with stainless steel types 302, 347, 431 and others being used for corrosion resistance.

Titanium alloy bolts often show greater fatigue resistances than comparable steel fasteners even at temperatures up to 500° F. Some of the more highly regarded titanium alloys for this purpose are 7Al-4Mo, 6Al-4V and 13V-11Cr-3Al. Additional advantages of titanium alloy bolts are that they do not rust and that they weigh one-third less than steel. Indeed, if titanium alloy bolts were substituted for all of the steel bolts in the Northrop F-89, the overall weight would be reduced by about 100 lb while, on an eight-engined heavy jet bomber, the saving would be more than 1200 lb (ref. 45). The deterrent to large scale use of titanium alloy fasteners is their high cost which results not only from the cost of the material but also from the fact that manufacturing techniques must be tailored to each individual lot of material in order to achieve a uniform product. The value of weight-saving must be high to justify wholesale use of titanium alloy bolts.

The same economic considerations apply, only more so, in the case of beryllium bolts. While not yet available except in limited quantities, beryllium bolts have demonstrated a superiority to both steel and titanium bolts in terms of static and fatigue strength-to-weight ratios. In comparison with stainless steel bolts, beryllium bolts maintain a superiority up to 850° F. The atmospheric corrosion resistance of beryllium is similar to that of aluminum.

Further information on bolt materials is presented in Section 5.1.

3.7 Platings

(Supplementary references: 29, 31, 48 through 52)

Plating of aircraft bolts is required to resist oxidation and corrosion, and also to act as a lubricant to resist galling. The important considerations are to avoid platings which deposit with residual tensile stresses and plating processes which induce embrittlement.

Electrodeposits of cadmium, tin, lead, zinc, copper or silver will improve the corrosion-fatigue resistance of steel bolt threads without seriously impairing the normal fatigue properties, providing the necessary precautions are taken during the plating process to minimize hydrogen embrittlement.

Most aircraft bolts for normal environments are cadmium plated in accordance with QQ-P-416. The susceptibility of this process to hydrogen embrittlement increases with the thickness of the plate and the hardness level of the bolt. While dense platings provide greater corrosion resistance, they also prevent hydrogen from being effectively outgassed. Hence, the thickness of cadmium platings on aircraft bolts is generally kept below 0.0003 inch. With steel bolts heat treated to hardnesses above about Rockwell C46, vacuum deposition of the cadmium per MIL-C-8837 is recommended to avoid embrittlement.

The effectiveness of cadmium plating as a lubricant is controversial. One source reports a fifty percent reduction in driving torque as a result of the plating, while another insists that it actually creates galling. In any event, it is generally agreed that all bolts which will require periodic removal or retightening should be given a supplementary oil or grease lubrication. Molybdenum disulfide and graphite dry film coatings in suitable binders are occasionally added for increased lubrication and prevention of galling.

In certain types of moving-structure joints, such as wing folds, aircraft bolts serve as shafts. Cadmium platings and dry film lubricant coatings are frequently bypassed for these applications since they tend to deteriorate under the fatigue action of oscillatory motions accompanied by high bearing loads. Chromium platings of bolt shanks are more durable, and are often specified for these applications because of their low friction coefficients. Commercial and industrial hard chromium platings should be avoided since these are normally deposited in a relatively crack-free state of residual tension. Special chromium plating techniques have been developed in recent years which leave a network of many fine hairline cracks in the plating. These do not act as fatigue nuclei but rather prolong fatigue life by relieving the otherwise-present residual tensile stresses. They also provide reservoirs for the lubricant. A certain amount of hydrogen embrittlement is normally associated with chromium plating, but this has not been found to be excessively deleterious. Baking at temperatures to about 400° F to eliminate it generally results in a lowering of the fatigue strength. Baking at 750° F is desirable, however. The thermal expansion of chromium is about half that of steel, and the high temperature bake stretches the chromium beyond its elastic limit, thereby providing further stress relief. Residual compressive stresses in chromium platings have even been achieved in this way.

Two methods are available for obtaining satisfactory chromium platings using ordinary plating processes. The plating can be etched to provide stress relieving cracks, or the bolt shank can be peened prior to plating per QQ-C-320. Apparently, the peening raises the fatigue strength and the plating reduces it. The net effect has been reported as an increase in fatigue life of seven to ten percent over unpeened, unplated specimens.

Even with the most up-to-date techniques, however, the fatigue strengths of chromium plated bolt shanks are limited by the fatigue strength of the chromium. Failure of the plating creates a notch effect which leads to rapid failure of the steel shank. There is little advantage, therefore, in using very high strength steels in these applications. Incomplete bonding of the chromium plating should also be avoided since this, too, acts as a crack nucleus.

Bright nickel platings behave similarly to chromium platings in most respects but are not widely used on bolt shanks for moving-structure joints because of their relatively poor fatigue and friction properties. Dull nickel platings generally embody residual tensile stresses.

Further information on platings is given in Section 5.3.

3.8 Civil Engineering Bolts

The so-called "high tensile" or "high strength" steel bolts used in civil engineering applications should not be confused with aircraft quality fasteners. These civil engineering bolts are made of medium carbon steel with tensile strengths of 90,000 to 120,000 lb/in.^{2} and yield strengths of 77,000 to 88,000 lb/in^2. The threads may be either cut or rolled.

In civil engineering applications, these bolts require hardened steel washers under the head and the nut to prevent brinelling. Most studies show that when properly installed, they give higher fatigue strengths than similar riveted joints.

For further information see ASTM Designation A325-58T and the collection of articles on the subject in Fasteners, Vol. 15, Nos. 2 and 3, Fall-Winter 1960.

3.9 Cap Screws

(Supplementary references: 12, 29, 35)

When a tensile force is applied to a cap screw which is threaded into a tapped hole in a structural member the stress pattern in the structural member is much more favorable than the pattern which develops in a nut (see Fig. 3). The distribution of load among the threads is more uniform with the cap screw than with the bolt-nut combination. For this reason greater fatigue lives are normally attained with properly used cap screws than with comparable bolts. When the cap screws are used in light alloys which have lower elastic moduli, the fatigue strengths of the screws approach twice those of bolts.

As with a bolt, cap screws should be designed so that the first working thread is not located at the face of the seating. Also, the shank should be undercut or a stress relieving groove provided when bending is present. When these precautions are taken, fatigue failure frequently occurs at the thread closest to the bottom of the tapped hole. The fatigue strength at this location may be improved by axial drilling of the threaded end of the screw to decrease its stiffness and further distribute the load.

Cap screws should not be permitted to touch bottom in the tapped hole because no preload can then be developed.

4. FATIGUE OF BOLTED JOINTS

The effective stress concentration factor for a bolted joint may vary from about 3 to as high as 13 depending upon the detail design. This is a tremendously important consideration affecting the fatigue life of a joint. Relatively simple changes in detail design may improve life by a factor of 10 to as much as 100. As general rules, wherever possible, eccentric loadings should be eliminated in order to minimize tension and bending on shear bolts, and symmetrical designs should be utilized.

Experience has shown that, for a joint material with a given ultimate strength, better fatigue properties are obtained with lower values of yield strength. Preliminary results (ref. 53) suggest that the ratio of yield strength to ultimate strength should not exceed about 85 percent.

4.1 Joint Configuration

(Supplementary references: 11, 13, 54 through 57)

a. Single shear joints

In aircraft structures, the most fatigue-sensitive parts are generally those single shear (lap) joints in which rivets or bolts must resist a load, as distinct from those where the rivets or bolts merely hold an unbuckled skin on the frame. For conventional spacings, the stress concentration in the former case is usually about 5 1/2, while in the latter case it approaches 3.

Single shear joints are poor in fatigue because of their inherent eccentricity which leads to bending of the bolts. Under this condition, the bearing loads are restricted to a small area of the hole wall and this results in increased stress concentrations at the holes.

Bending can be minimized by exercising extreme care in the design of the splice (or doubler) plate. Obviously, bending is reduced if the doubler is made as thick as possible. This, however, exaggerates the unequal distribution of load among the bolts which is a function of the difference in stretch of the upper and lower sheets between the first and successive rows of bolts. This unequal distribution can be reduced, in turn, by spacing the bolts closer together and by proper use of bolts of different moduli. Further improvements are possible by using a tapered doubler, by using a stepped laminated doubler, or by using an auxiliary thin doubler to take most of the bending at the first bolt.

Bonding the splice plate to the main plate is also helpful in that bending then starts at the edge of the splice rather than at the first row of bolts. Still another device involves the addition of a bolt just outside of the splice so that the first loaded bolt benefits from the mitigating stress interference effects of bolts on both sides of it.

b. Double shear joints

Double shear (or double strap) joints have the advantage of involving no eccentricity and bending. Further improvements in the fatigue strengths of these joints can be obtained by using a transverse stress relieving groove (Fig. 4a) or reduced plate thickness (Fig. 4b). With these modifications no special care need be taken with hole preparation (see Section 4.5) since, if the bolts are properly prestressed, fatigue failure generally takes place at the edge of the groove or fillet. As with lap joints, however, it is desirable to use a high concentration of bolts in a small area to minimize unequal bolt loading.

c. Scarf joints

Little experimental evidence is available in the literature on the fatigue properties of single scarf joints. The quality of this configuration is apparently very sensitive to detail design, material, and assembly procedure.

Double scarf joints, on the other hand, are the strongest joint configurations in fatigue because they involve no eccentricity and because they permit the bolts to be preloaded in shear.

4.2 Joint Proportions

(Supplementary references: 10, 11, 17, 18, 20, 21, 41, 58)

Having chosen a joint configuration, the next step in the design problem (aside from materials selection) is to select a bolt diameter, sheet thicknesses, and the number of bolts to be used. Insofar as fatigue strength is concerned, the criteria which govern these selections are as follows:

(1) Per unit of cross sectional area, smaller bolts are stronger in fatigue than larger ones. This is primarily a matter of stress concentrations rather than a metallurgical size effect, and the differences are small for near or adjacent sizes.

(2) Larger bolt diameters permit larger preloads and greater clamping forces. This leads to greater proportions of the joint load being transferred by plate friction and less fretting of the joint.

(3) Per unit of joint thickness, thinner joints are stronger in fatigue than thicker ones because, in thinner joints, greater proportions of the total load are transferred by plate friction.

(4) Higher ratios of bolt diameter to sheet thickness result in less bending of the bolts. Bolt bending reduces the fatigue strength of the sheet as well as of the bolt by creating a non-uniform distribution of bearing stress through the sheet thickness.

(5) When a considerable portion of the bolt load is in tension, the grip should be as large as possible to minimize bending stresses resulting from possible nonaxial bolt loading.

(6) Increasing the number of bolts in a joint permits the clamping force to be distributed more uniformly over the entire joint area. This reduces both the bearing stresses and the tendency to slip.

Of these six considerations, the first one is generally of secondary importance and may usually be disregarded. The second, third and fourth considerations are compatible with one another. The problem is thus ordinarily reduced to employing as high a ratio of bolt diameter to sheet thickness as possible, and as many bolts as possible. Space limitations, however, would generally require that bolt diameter be reduced if space were needed for more bolts, or that the number of bolts be reduced if space were needed for larger ones. Achieving the most favorable balance between number and size of bolts, then, is often a test of the designer's experience.

It should be noted, however, that if the tensile fatigue strength of the bolts is the critical factor, the bolt diameter should never be made greater than the grip. In certain cases it is even desirable to increase the grip by inserting washers or collars.

4.3 Bolt Pattern

(Supplementary references: 11, 41, 55, 57)

Adding more bolts to a joint raises its fatigue life because the increased clamping reduces slippage, and because the bearing stresses are reduced.

For a bolt row which is transverse to the direction of load, widening the joint and adding proportionately more bolts increases the fatigue strength of the joint approximately proportionally. If the bolts are spaced closer together, the ratio of fatigue strength to static strength increases. The optimum spacing is about 2 1/2 diameters.

For a bolt line which is parallel to the direction of load, adding more bolts raises the fatigue strength of the joint but lowers the average fatigue strength per bolt because the load is not uniformly distributed among the bolts. The first bolt carries the greatest share of the load. If the bolts are spaced closer together, the load distribution becomes more uniform and the fatigue strength is increased.

Another method of reducing the peak loads taken by the first bolts involves the use of aluminum bolts on the outside and steel bolts on the inside. Still another technique which has been suggested for achieving the same effect calls for smaller bolts on the outside and larger ones on the inside. Tests have demonstrated the latter suggestion to be invalid, however. In any case, it is undesirable to employ a mixture of fasteners in a joint, such as rivets and bolts, or tension bolts and shear bolts.

As a general rule, best results are obtained when each row contains the same number of bolts, and the different rows are in line with one another.

4.4 Preloading of Joints

(Supplementary references: 58 through 62)

In Section 2, it was shown how the preloading of bolts increases the fatigue life of both the bolts and bolted joints. This section will discuss further improvements in the fatigue life of joints which can be attained by preloading the joints themselves. In a sense, the use of the term "preloading" for both processes is unfortunate, since the two processes are quite different. When referring to bolts, the preload is a state of tension in the bolt which is created by torquing the bolt; every effort should be made to prevent the tension from diminishing during the subsequent fatigue loading. In the case of joints subject to tensile fatigue, the preload is a tensile load which is applied to the joint and removed immediately thereafter, prior to the fatigue loading.

The application of a high tensile preload to a joint creates local stresses in the vicinity of the bolt holes which exceed the yield strength of the material. When the preload is removed, local permanent tensile deformations remain, and a state of residual compressive stress is created in the vicinity of the holes. These residual compressive stresses reduce the effective mean stress which is actually "felt" by the material in the vicinity of the holes during the ensuing tensile fatigue loading. The fatigue properties of most structural aluminum alloys are quite sensitive to mean stress so that substantial increases in the fatigue life of joints fabricated from these materials are obtained when the effective mean stress is reduced by preloading. The fatigue properties of some steels, on the other hand, are relatively insensitive to mean stress. Joints fabricated from these materials are, therefore, relatively unaffected by this kind of preloading.

Note that this effect is, to a certain extent, opposite to that produced by the preloading of bolts where the mean stresses are raised, albeit with a simultaneous reduction in the alternating stress amplitude.

It is worth mentioning that preloading cannot significantly alter the fatigue properties of an ideally homogeneous specimen which does not have any stress concentrations since no macroscopic residual stresses can be created. This is one reason why such poor correlation is frequently encountered between the fatigue properties of materials and those of structures.

Preloading also increases the fatigue life of joints in another way. The plastic flow which takes place in the vicinity of the holes permits the subsequent fatigue loads to be more uniformly distributed among the various bolts. It is this effect that produces somewhat greater preload benefits with joints than with other types of structural stress concentrations.

Obviously, no benefits can be obtained unless the preload is high enough to create some localized plastic deformation. Raising the preload raises the fatigue life until some optimum amount of deformation is reached. Preloads greater than this apparently introduce excessive hole elongation or other damage. The actual magnitude of the life increase obtained through preloading depends on the nature of the subsequent fatigue loading as well as on the magnitude of the preload and the joint design. It has been conjectured that the residual compressive stresses which are created by tensile preloading tend to fade away during the ensuing fatigue loading, particularly when the fatigue stresses are high with respect to the yield strength. This would explain why the relative increase in fatigue life for a given preload decreases as the maximum load of the fatigue cycle is increased. This would also explain why spectacular improvements in fatigue life are obtained when the residual stresses are periodically restored by the application of single overloads at long intervals during the fatigue process, even when the overloads are of only moderate intensity. On the other hand, the application of a series of preloads prior to the initiation of fatigue loading results in fatigue life benefits which are only sightly greater than those obtained with a single preload.

Thus far, this discussion has been concerned with tensile preloads which are followed by tensile fatigue. The case of compressive preloads followed by compressive fatigue is qualitatively similar. Tests have substantiated the intuitive conclusion that preloads which are applied in the direction opposite to the subsequent fatigue loading create residual stresses which raise the effective mean fatigue stresses and thereby reduce fatigue life.

The proper choice of a joint material is all-important if there is a desire to employ the benefits of preloading. Unfortunately, this choice is not easy to make with the limited fatigue data available for various materials at various mean stress levels. It has been found, for example, that preload benefits are greater with joints fabricated from an extruded aluminum alloy than with identical joints fabricated from sheets of the same material. The reasons for this are not yet clear.

While structural preloading appears to be a powerful tool, it cannot be accepted as a design parameter as yet because it is not fully understood nor have all its implications been fully explored. Like bolt preloading, the best use for structural preloading may be as a means for achieving reliability through the attainment of fatigue lives beyond the design requirements.

One interesting corollary of the preload phenomenon should be emphasized. It is a questionable practice to evaluate the fatigue life of a structure using a specimen which had been previously tested to static failure and then repaired. This procedure may give an unrealistically high fatigue life and, perhaps, fatigue failure in the wrong place.

4.5 Hole Preparation

(Supplementary references: 11, 20, 35, 58, 63 through 70)

The fatigue strength of a strip with a hole is about 10 percent less for a threaded hole than for a smooth hole. If the hole is loaded with a bolt, the difference becomes much smaller.

The remarks made earlier regarding plating of bolts apply equally well to the plating of holes in thick lugs. Proper cadmium plating of holes was found to cause no serious reduction in fatigue properties, but chromium plating caused an eight-fold reduction in fatigue life, presumably due to the residual tensile stresses which frequently accompany this process.

An interference fit between a bolt and the unthreaded hole boundary increases fatigue life by reducing fretting and impacting between the bolt and the hole boundary and by changing the stress conditions at the boundary. Under externally applied fatigue loading, an interference fit effects a rise in the mean stress level at critical points on the hole boundary, together with a simultaneous fall in the alternating stress amplitude. This phenomenon is entirely analogous to the beneficial changes in stress conditions which are created in bolts by preloading.

The beneficial effects of interference fits do not become significant until the ratio of the bolt diameter to the hole diameter exceeds a certain threshold value. This value depends, to some extent, on the materials involved, but is generally about 1.003. (Small changes in the modulus of elasticity of the bolts affect the fatigue properties only slightly as compared to the effects of very minute changes in the degree of interference.)

As the bolt-to-hole diameter ratio is increased, the mean stress level also increases, but the alternating stress amplitude decreases only until a certain limiting value of interference is reached. Beyond that point, the alternating stress amplitude remains essentially unchanged. The magnitude of this limiting value of interference depends on the ratio of plate width to hole diameter. The most desirable degree of interference, however, depends on the relative influences of mean stress and alternating stress on the fatigue resistance of the plate material involved and may be less than this limiting value.

In any case, of course, the degree of interference employed should not be so great as to hamper the proper preloading of the bolts, nor to create plate stresses exceeding the yield strength. For materials subject to stress corrosion cracking or hydrogen embrittlement, the degree of interference must be reduced further. BuWeps specification SD-24J requires, in this case, that the interference stresses at the hole boundary not exceed 50 percent of the material's yield strength in the longitudinal grain flow direction, 35 percent in the long transverse and 25 percent in the short transverse direction.

An alternative means of achieving an interference fit involves the use of an interference bushing. In this case, the bolt need have only a sliding fit in the bushing. Excessive clearance between the bolt and the bushing, however, tends to produce early fatigue failure of the bushing.

The full benefits which are available through the use of interference fits are secured only if the degree of interference is maintained by regular inspection and repair in service. This is particularly true when hardened steel bolts are used in softer plate materials. The bolts tend to brinell into the plate material and reduce the tightness of the fit. This brinelling action can be minimized by using high clamping forces which are achieved, in turn, by high preloading of the bolts and by the use of more bolts.

A relatively new technique for improving fatigue strengths around bolt holes in aluminum alloys is represented by the proprietary development known as "coining." In this technique* a die or rolling wheel is used to form a concentric groove around the hole. The metal adjacent to the hole is forced inward, creating a residual compressive stress field, and it is usually necessary to ream out the hole to its original diameter to permit insertion of the bolt. The residual compressive stresses can produce rather dramatic improvements in fatigue life by reducing the effective mean stresses under cyclic loading. This phenomenon is thus seen to be akin to structural preloading and opposite to the stress redistribution effects which result from the use of interference bolts.

In order to obtain the most favorable increases in fatigue properties it is necessary to employ the correct coining depth, coining diameter, groove radius, and spacing between holes. With thin sheet materials coining of only one side has been found best while with stock thicker than about 0.1 inch both sides should be coined.

It is of interest to note that it is not generally advisable to preload a structure containing coined holes, presumably because the combined effect of the preload and the coining can easily exceed the optimum amount of localized plastic deformation mentioned in the previous section.

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*Not to be confused with the ordinary coining process in which images or characters of a punch and die are impressed onto a plane metal surface.

4.6 Fretting

(Supplementary references: 13, 20, 40, 71, 72)

Fretting is a phenomenon which is brought about by a combination of (1) a high stress parallel to a surface which is in contact with a second surface, and (2) relative movement at the point of contact. When both of these conditions exist, a process of local welding and direct adhesion takes place, followed by a tearing apart of the temporary connection so formed. Some investigators believe that eventual failure results from the adhesion and its accompanying stress concentrations rather than from the surface disruption.

In the fatigue loading of a bolt and nut combination, relative movement is most pronounced on the thread flanks but the maximum stresses are in the thread roots. Hence, tests show that fretting is not ordinarily a contributor to fatigue failures in bolts.

On the other hand, many fatigue failures of joints are initiated by fretting. The elimination of relative movement between the faying surfaces by tight clamping is, therefore, very important. This represents another argument for distributing bolts all over the contact area, using a close pitch and preloading the bolts with a high torque. Indeed, to achieve greater clamping, it has been suggested that the fatigue strength of a joint can be raised by increasing the bolt size even though failure does not occur in the bolt and even though the cross sectional area of the joint is thereby reduced.

When relative movement cannot be entirely eliminated, it is necessary to lubricate the faying surfaces to delay fretting failure. Grease is a good preventative but not permanent. Molybdenum disulfide, either alone or in an epoxy resin base, is more effective. Still another practice involves the insertion of a thin wafer of a non-metallic material between the contacting surfaces.

An extreme case of fretting has been observed in the repeated impacting of two stressed surfaces, such as a bolt shank and a hole wall. Lubrication was found to be ineffective here since a protective film could not be maintained under the repeated impacts.

5. EFFECTS OF TEMPERATURE ON THE FATIGUE STRENGTH OF BOLTS AND BOLTED JOINTS

(Supplementary references: 32, 45, 73 through 82)

Most specifications for aircraft fasteners are written around room temperature properties in spite of the fact that aircraft operating temperatures have been rising steadily over the past two decades. That this procedure has not, perhaps, caused serious difficulty except in exploratory vehicles may be attributed to the fact that materials have been available which are capable of withstanding the temperature levels attained by conventional military and commercial airplanes in the past without excessively deleterious effects. In the next decade, however, airframe temperatures may be expected to severely try both the capabilities of materials and the ingenuity of designers.

Figure 5 presents an estimate of the maximum equilibrium skin temperatures which will be reached by conventional manned military aircraft during the next ten years. Transient peak temperatures attained will be even higher.

The problems introduced by operation at these temperatures are multifold. For the purposes of this report, however, the most important is the fact that the fatigue properties of materials do not vary with temperature in any generally systematic or theoretically predictable manner, particularly when stress concentrations are present. Furthermore, the results of fatigue tests at elevated temperatures are dependent upon the cycling speed employed. Hundreds of tests must be conducted on each material before its fatigue properties as a function of temperature and time can be considered defined. Unfortunately, this can only be considered a beginning toward the solution of hot fatigue problems. Even a full set of elevated temperature fatigue properties for fasteners and structural materials is insufficient, by itself, to properly design a joint for a hot fatigue environment because, when temperature is introduced as a variable, structural interactions take place between a bolt and its nut, and between the bolt-nut combination and the material being fastened. These interactions are enumerated in the subsections below. Further discussion of temperature effects is presented in Section 6.

5.1 Bolts

Under elevated temperature exposure three structural interactions take place between a bolt and the bolted material, all of which change the amount of preload in the bolt. The first, and most significant of these, is stress relaxation. Stress relaxation of bolts under creep environments is a well-known, though perhaps not so well-understood, phenomenon. Frequently ignored, however, is the fact that relaxation of the bolted material also reduces the bolt preload. Temperature dependent changes of the elastic moduli of the bolt and the bolted material also change the magnitude of the preload in the bolt. For elevated temperature applications, this change is almost always a reduction. Finally, the development of differential thermal expansions between the bolt and the bolted material causes changes in the bolt preload.

The latter interaction can be minimized by choosing bolt and bolted materials which have reasonably similar thermal expansion properties. Failure to do this may result in either an immediate loss of preload upon exposure to high temperature or an immediate increase in preload to stress levels which may be high enough to cause rapid relaxation and the eventual loss of preload upon return to normal temperatures.

Thermal expansion compatibility will also reduce the danger of thermal fatigue of bolts in applications where temperatures are cycled either frequently or severely.

Some bolts for aircraft engines are now made with the thread diameters reduced, usually 0.003 in. from standard, for applications over 550° F. It is claimed that this facilitates retightening, removability and reusability which might otherwise become impossible due to the galling created by elevated temperature oxidation and breakdown of the plating.

Figure 6 shows current thinking with regard to suitable bolt materials for elevated temperature use. The temperature capabilities represent a compromise between fatigue, creep and short-time properties, and therefore are not directly applicable to any specific application.

Thread rolling may be difficult with some of these materials, and the costs of this process may be high if special equipment is needed and if production quantities are small. On the other hand, tests indicate that the residual stress benefits of rolled threads decrease with temperature, so that requirements for rolled threads may, perhaps, be eliminated in special cases if a more economical means of obtaining a smooth surface finish is found.

5.2 Nuts

When self-locking nuts are used, a further reduction of bolt preload occurs at elevated temperature from the relaxation of the locking element regardless of whether it is of the insert, beam or elliptical type. This relaxation may be reduced somewhat by increasing the nut thread length so that the unit stresses in the locking device are reduced. Relaxation of the locking element limits the number of times a nut may be satisfactorily retightened on a bolt to restore the desired preload. Using a torque test similar to that described in MIL-N-25027, one investigator found that self-locking nuts exposed to 900° F could be reused at least 15 times, while those exposed to 1200° F had practically no reusability. The periodic replacement of self-locking nuts is, therefore, a necessity.

A mitigating effect of elevated temperature exposure is that creep in the threads, if not excessive, acts to distribute the bolt load among the threads more uniformly, and thereby tends to prolong fatigue life, other things being equal.

When applying nuts to reduced diameter bolts (Section 5.1), it must be recognized that different torques are required to produce given amounts of preload because the friction characteristics are changed.

A graphic presentation of the approximate temperature capabilities of nuts fabricated from various materials is given in Fig. 7.

5.3 Platings

Cadmium melts at 609° F and the peak useful temperature of cadmium platings for short-time service is about 550° F. For continuous duty the limiting temperature is reduced to about 450° F to prevent diffusion of the cadmium into the base metal. Diffused nickel-cadmium platings, on the other hand, are satisfactory up to about 900° F. For still higher temperatures, silver plating must be employed. Silver is a very satisfactory plating material except for its high cost because it acts as a good lubricant as well as a protective coating without the addition of a supplementary dry film lubricant. The usefulness of silver, however, ends at about 1300° F. Beyond that temperature, a natural finish is most common although a hard chrome finish is occasionally employed up to 1600° F, despite its deficiencies in terms of fatigue strength. A proprietary finish called KAF is also reported by its manufacturer to be usable to 1600° F, and two other products, Chromalloy W-2 and Durak MG, have seen service at even higher temperatures.

5.4 Joint Materials and Designs

The selection of a configuration for a particular joint is generally dictated by the structural application, rather than by the environment. Hence, skin-to-spar cap connections, for example, are necessarily required to be lap type joints, whether for ordinary or elevated temperature use. Similarly, the principles governing the design of the bolt pattern and the joint proportions, which were outlined earlier, apply here.

If the two materials being joined do not have comparable thermal expansion properties, the already unequal distribution of load among the various bolts becomes aggravated, and the peak loads taken by the end bolts becomes greater. This is an important consideration in the selection of doubler materials. However, just as creep tends to distribute a bolt load more uniformly among the threads, it also tends to distribute the total joint load more uniformly among the various bolts. Excessive creep, of course, produces sufficient hole elongation to create undesirable slippage under alternating loads. Under these conditions, an interference fit between a bolt and its hole, which might otherwise be used to combat slippage, may be difficult to maintain.

Figure 8 presents an estimate of the capabilities of various structural materials for supersonic airframes. As in the case of Figs. 6 and 7, the capabilities shown represent compromises of various mechanical properties. The optimum material for a particular application can only be selected after a detailed study of the requirements. Also to be considered is cost. The tabulation below indicates the approximate costs of some structural materials. Actual costs vary with the requirements of the material specifications involved.

Alloy	Relative Cost
302	x
17-7 PH	2x
A286	3x
Inconel X	6x
Titanium	20x
Rene 41	25x
S-816	30x
Beryllium	700x

6. DISCUSSIONS OF THE FATIGUE OF BOLTS

6.1 On the Evaluation of Bolts for Various Applications

Bolts occupy an unusual place in the field of aircraft components. The gradual development of many aircraft components has been the result of efforts on the part of both the aircraft industry and the component manufacturers. By contrast, the tremendous amount of specialized equipment and fabricating know-how that is required to produce top quality bolts has relegated their development primarily to the bolt manufacturer, albeit with a view toward satisfying aircraft requirements. Of necessity, therefore, many aircraft bolt procurement specifications are patterned after the bolt maker's own fabrication specifications with the addition of some acceptance tests. The acceptance tests, in turn, are designed to guarantee uniformity in accordance with some standard of quality, but usually do not serve to verify the mechanical properties which the purchaser expects the bolt to have.

The ever-increasing cost of high performance military aircraft, however, has forced the bolt user to reappraise the situation, and it is inevitable that he must tighten his specifications in order to insure reliability. At the same time, the specifications must be realistic; they cannot exceed the bolt maker's capabilities. They must, therefore, be based upon critical evaluations of available products.

The remainder of this subsection deals with the evaluation of bolts in terms of fatigue requirements. Section 6.2 extends the discussion to the specifications themselves.

a. Bolts for tension fatigue applications

The evaluation of the capabilities of a bolt for a tension fatigue application is based on many considerations but the most important is, of course, the S-N curve for the bolt. For this reason, the author was mildly surprised to find that there does not exist (or there is, at least, not readily available) a compilation of S-N curves for the various types and sizes of aircraft bolts. Clearly, the existence of such a manual would have gone a long way toward making the present report unnecessary. It is strongly recommended that the preparation of such a manual be initiated immediately, and that it be supplemented and revised regularly to keep it up to date. The procedure for preparing such a manual could take the following form:

(1) Compile the data on fatigue of aircraft bolts that are already available in the literature. A considerable amount of such data does exist and a reference list for this was started during the course of the survey reported herein.

(2) Request the manufacturers and large users of aircraft bolts to make available the data of this type which they have secured for their own purposes.

(3) Establish standard fatigue testing methods for evaluating aircraft bolts. In the long run, this is the most important step. An excellent precedent has been set by the National Aircraft Standards Committee of the AIA with the issuance of specification NAS 1069, "Tension Fatigue Test Procedure for Aeronautical Fasteners" (see also ref. 47), which is more complete than the "comparable" sections of Fed. Test Method Std. No. 151.

(4) Set up a low-cost, open ended test program to evaluate promising new bolts, new bolt features and new bolting accessories as they become available. Examples of the evaluations that would be immediately desirable might include aircraft quality bolts with threads having 90° flank angles and with truncated threads, and nuts having the so-called Equa-Stress thread form.

Naturally, the availability of S-N curves for aircraft bolts is not the whole answer. The effects of preload should also be known, at least as to whether it is beneficial to the particular bolt or bolt material in question. This, incidentally, points up a shortcoming in NAS 1069, which calls for the minimum load to be 10 percent of the maximum load in all cases, and thus furnishes no clue at all regarding the effects of preload.

b. Bolts for non-tension fatigue applications

In the past, tensile fatigue problems in bolts have overshadowed shear fatigue problems, shear fatigue being more critical in the sheet than in the bolt. However, a fairly uniform characteristic of most high temperature bolting materials is their relatively low shear strength. Consequently, it appears wise to prepare now for a potential shear fatigue problem in bolts.

Virtually nothing is known about fatigue in bolts loaded in "pure" single or double shear. It is even debatable whether fatigue failure can result from such loading. This question is purely academic, however, since in actual practice a certain amount of bending almost always accompanies so-called shear loading of bolts. Bending loads, of course, are very damaging to threaded members in terms of fatigue life. Just how damaging they are cannot be estimated from tension fatigue data with present knowledge.

Clearly, research is needed to provide additional knowledge on this subject. This additional knowledge may demonstrate a relationship between shear/bending fatigue and tension fatigue properties. However, it is more probable that no such relationship will evolve, in which case it will be necessary to establish standard shear fatigue test methods for evaluating aircraft bolts comparable to the tension methods already discussed. The importance of this research and development cannot be overemphasized. There already exists a dire need for interaction curves describing the fatigue properties of aircraft bolts under combined loads.

c. Bolts for elevated temperature fatigue applications

Most of the comments made in the preceding paragraphs also apply to the evaluation of bolts for hot fatigue applications. The compiling of S-N curves, in this case for various temperatures, is particularly important. There is no dependable way to analytically evaluate the fatigue properties of a new bolt at a given temperature from the S-N curve at room temperature. The feasibility of experimentally evaluating the fatigue properties of bolts at elevated temperatures, on the other hand, has been demonstrated, ref. 32, at least for tension loadings.

Having determined the elevated temperature fatigue properties of a bolt, it is then important to examine its stress relaxation properties. It is entirely possible that certain potential bolt materials may exhibit excellent basic fatigue properties at elevated temperature, but at the same time have such poor stress relaxation properties that rapid loss of preload and premature fatigue failures in service would be unavoidable. Standard bolt stress relaxation test methods, therefore, must also be developed.

6.2 On the Specification of Fatigue-Resistant Bolts

a. Bolts for use at ordinary temperatures

Most procurement specifications for aircraft bolts, such as MIL-B-6812, do not call for any acceptance fatigue tests. It goes without saying that such bolts should not be employed in fatigue-sensitive areas.

A few of the higher strength bolts, such as the MS20004 series, do require acceptance fatigue tests. The test conditions for these, which are prescribed in MIL-B-7838, call for tests under a single specified stress amplitude. The question is, "Are the results of tests under a single stress amplitude sufficient to substantiate an entire S-N curve?" The answer is a qualified "yes", at least insofar as tension fatigue alone is concerned. The qualifications are as follows:

(1) Bolts fabricated in accordance with each of the permissible variations given in the specification should all have S-N curves which are similar to, or superior to, an arbitrary S-N curve which it is desired to substantiate. (The permissible variations in most specifications extend to a choice of several materials and fabrication processes.)

(2) The prescribed fatigue stress amplitude for the acceptance tests should be chosen after careful study of the S-N curve, to be sure that it is in a region near the middle of the curve where the dS/dN slope is relatively high and the scatter in N is relatively low.

As mentioned earlier, the present state of knowledge precludes the use of acceptance tests in tension fatigue as a substantiation of shear fatigue properties.

b. Bolts for elevated temperature use

Separate procurement specifications for fatigue-resistant bolts will have to be prepared for each temperature range (e.g. 900, 1200, 1500° F) because the materials involved for each range have different characteristics.

The procedure of using a single fatigue stress amplitude to substantiate an entire S-N curve is probably as applicable to the elevated temperature case as it is to the room temperature one. However, the tests will have to be conducted at the temperature in question because it is highly unlikely that bolts fabricated from a number of materials will have similar S-N curves both at room temperature and the temperature in question. An alternative procedure would limit each bolt specification to a single material (or, perhaps, two similar ones) and a single sequence of manufacturing operations. This procedure has been adopted in specifications such as AMS 7476 and AMS 7478. In this case, room temperature fatigue tests can provide reasonable substantiations for elevated temperature S-N curves providing the material does not experience significant metallurgical instability over the range of temperatures involved. The addition of an elevated temperature test would furnish greater reliability.

This is only the beginning, however. As pointed out in Section 5, the elevated temperature fatigue properties of a bolt, per se, do not define its fatigue properties in a joint at elevated temperatures because of the interactions which take place between a bolt, its nut, and the bolted material. Limitations on the usage of the bolt, such as permissible bolt-nut-material combinations, dimensional ratios, and temperature ranges should be called to the attention of the user. These limitations could be presented in the form of an AND standard, or included in a document such as MIL-HDBK-5 (formerly ANC-5). The disadvantage of this approach is that it is too restrictive, at least in the case of MIL-HDBK-5. Each deviation from the suggested usage would require approval from the procuring agency. A more fitting approach, at least for the present, would be to include these criteria in the respective bolt procurement specifications. The inclusion of usage limitations in bolt procurement specifications is common, and several now specify the appropriate nuts to be used.

The task of charting these criteria is primarily a matter of computation at this point, since much of the required data on stress relaxation and elastic and thermal properties are available in the literature. These criteria, with little modification, would also be applicable toward minimizing the possibility of thermal fatigue failure.

Eventually, of course, material of this nature should go into MIL-HDBK-5. It may be a while, however, since this document does not yet recognize the existence of such phenomena as stress relaxation and thermal fatigue.

6.3 On Errors in Fastener Replacement

In Section 1, it was pointed out that the variations of aircraft bolts commercially available are becoming more and more numerous. Under this situation, the possibility of using an incorrect bolt in a specific location must be recognized. The inadvertent use of an MS9033 bolt instead of an NAS 624 bolt, for example, could lead to disastrous consequences. The likelihood of such an error occurring is apparently greater when fasteners are being replaced than when they are being initially installed. Thus, elevated temperatures multiply the possibilities of danger in the case of nuts which require regular replacement. The accidental use of the wrong nut in a high temperature region could lead to rapid loss of preload and premature fatigue failure.*

The straightforward solution to this problem, namely, the indoctrination of repair, maintenance and inspection personnel to exercise particular care in this respect has not been entirely successful and a need for supplementary measures is indicated. Three approaches of a supplementary nature are outlined below. These may be implemented either singly or simultaneously, by directive from the procuring agency or voluntarily by the organizations involved.

a. Decrease the chances of installing a wrong bolt

The problem is not concerned with the inadvertent use of bolts with the wrong head style, or of the wrong size, but of "look-alike" bolts having different mechanical properties. Hence, this approach calls for a reduction in the number of "look-alike" bolts which shall be acceptable for use in assemblies fabricated for the procuring agency. The pruning of the undesirables could be effected after a comparative study of the mechanical properties, including the S-N curves, of the various types. As an example, MS20008-19 bolts can replace all AN149080 and NAS 148-32 bolts without any sacrifice of structural integrity. The penalty for this approach is that the user pays a premium for mechanical properties which he does not always need. In the example cited, the premium would be about seven percent. On the other hand, the resulting greater use of the MS bolt would tend to lower its price and, further, some savings would undoubtedly accrue to the user from a reduction in his inventories.

Another aspect of the problem deals with the wrenching recesses in countersunk head bolts (see Section 3.5). Research is required to establish the relative merits of the many recesses now available, and

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*It is recognized that a similar problem exists with the tools used for fastener replacement. Use of the wrong tools, particularly those of the impact type, has caused serious structural damage on occasion.

steps should then be taken to standardize on the more desirable ones.

b. Decrease the chances of failure resulting from the installation of a wrong bolt

This means upgrading the mechanical properties of the lower quality bolts insofar as this is economically feasible. One way to do this was brought out in this report. It was pointed out that rolled threads give fatigue strengths which are superior to those obtained with cut threads; that thread rolling is an economical process when large quantities are involved; and that most bolt procurement specifications do not necessarily require that threads be rolled. It is suggested that specifications not requiring thread rolling, such as MIL-B-6812 and MIL-B-7874, be amended to require it.

c. Increase the ease with which the presence of an improper bolt in an assembly may be detected

The usual technique of marking bolts and nuts with numbers and symbols requires too close an inspection for rapid identification. It is recommended, therefore, that the advantages of color coding be exploited more fully than heretofore. At present, this marking technique is used rather infrequently (cf., MIL-N-25027), despite its promise. Indeed, the enormous success which this technique has had in the electronics industry could well be a goal to strive for in the preparation and revision of fastener specifications.

Another suggestion is the promotion of an industry-wide practice of placing markings on assemblies adjacent to each bolt pattern, the markings to correspond with those on the proper bolts and nuts to be used.

* * *

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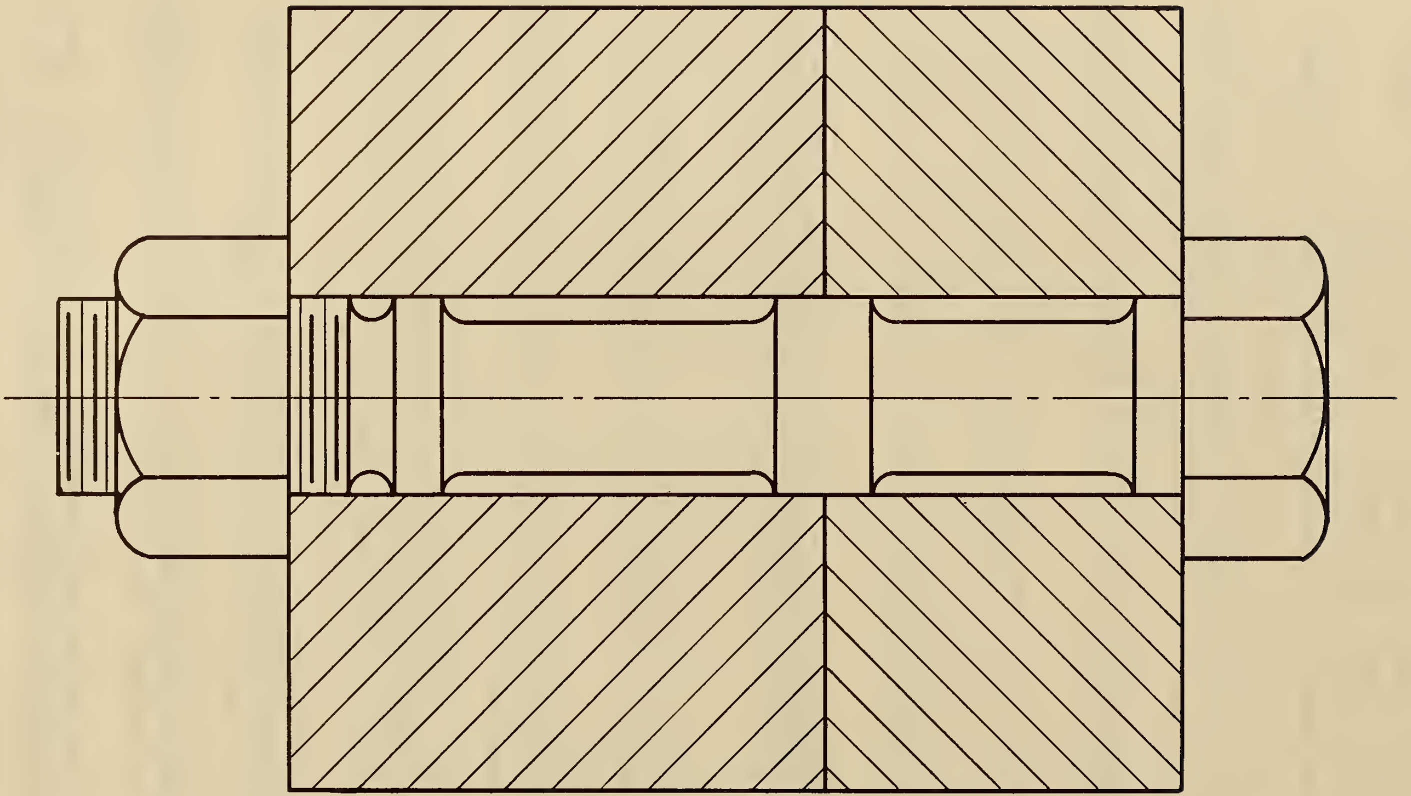


Fig. 1 Bolt with multiple undercuts

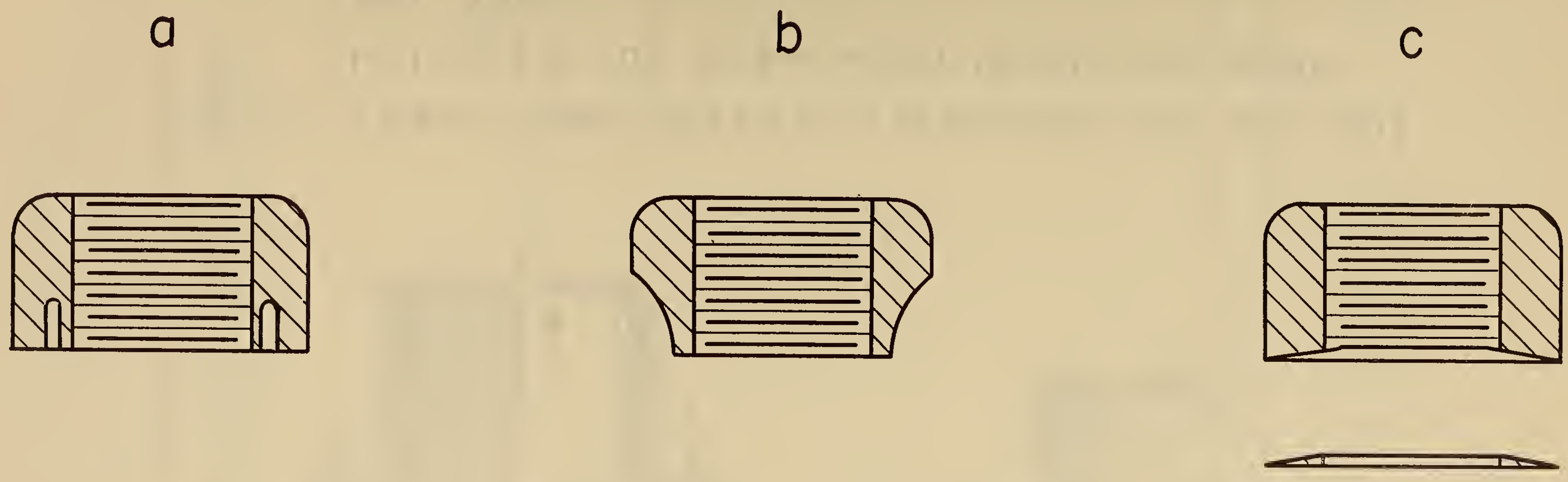


Fig. 2 Special nuts, (a) undercut with annular groove, (b) with tapered lip, (c) with concave working face and matching washer.

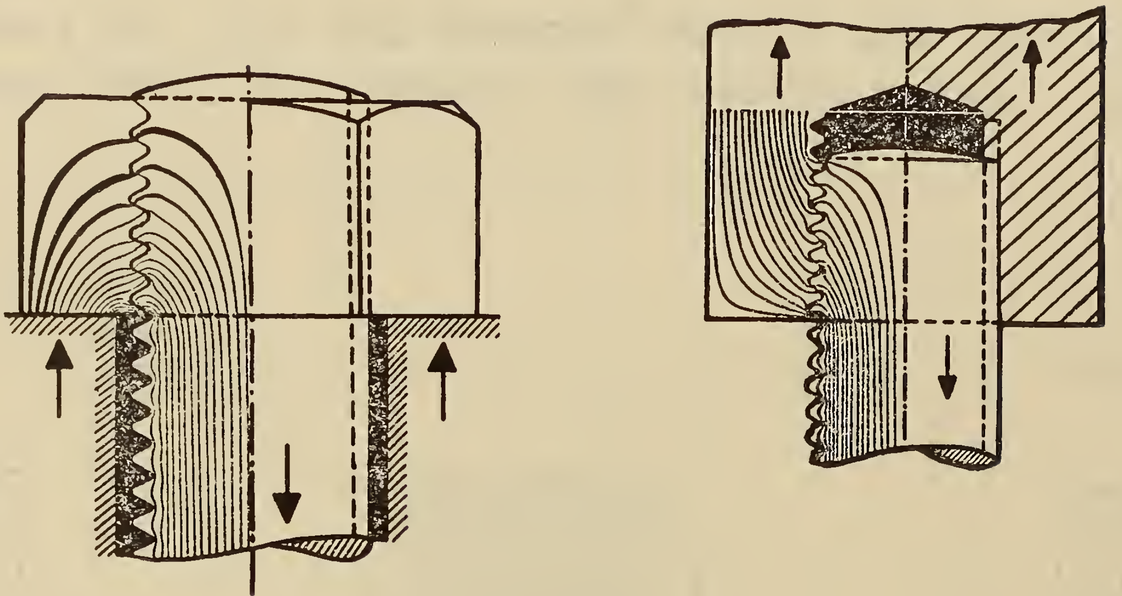
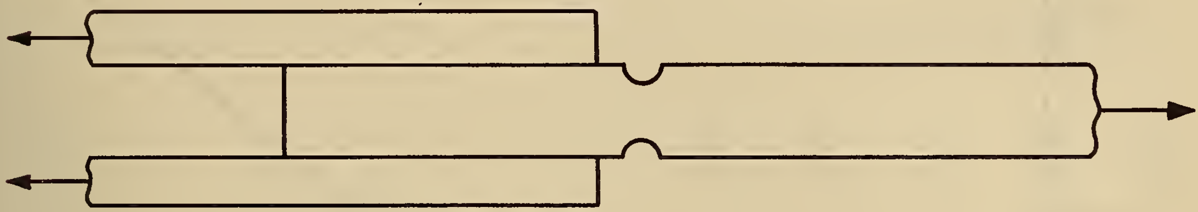
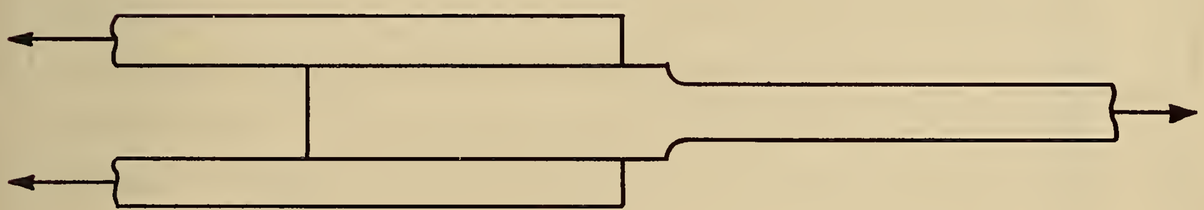


Fig. 3 "Stress-flow" lines for a preloaded bolt and nut (left) and a cap screw under tensile load (right). (Ref. 35.)



a



b

Fig. 4 Double shear joints with (a) stress-relieving grooves, (b) reduced plate thickness.

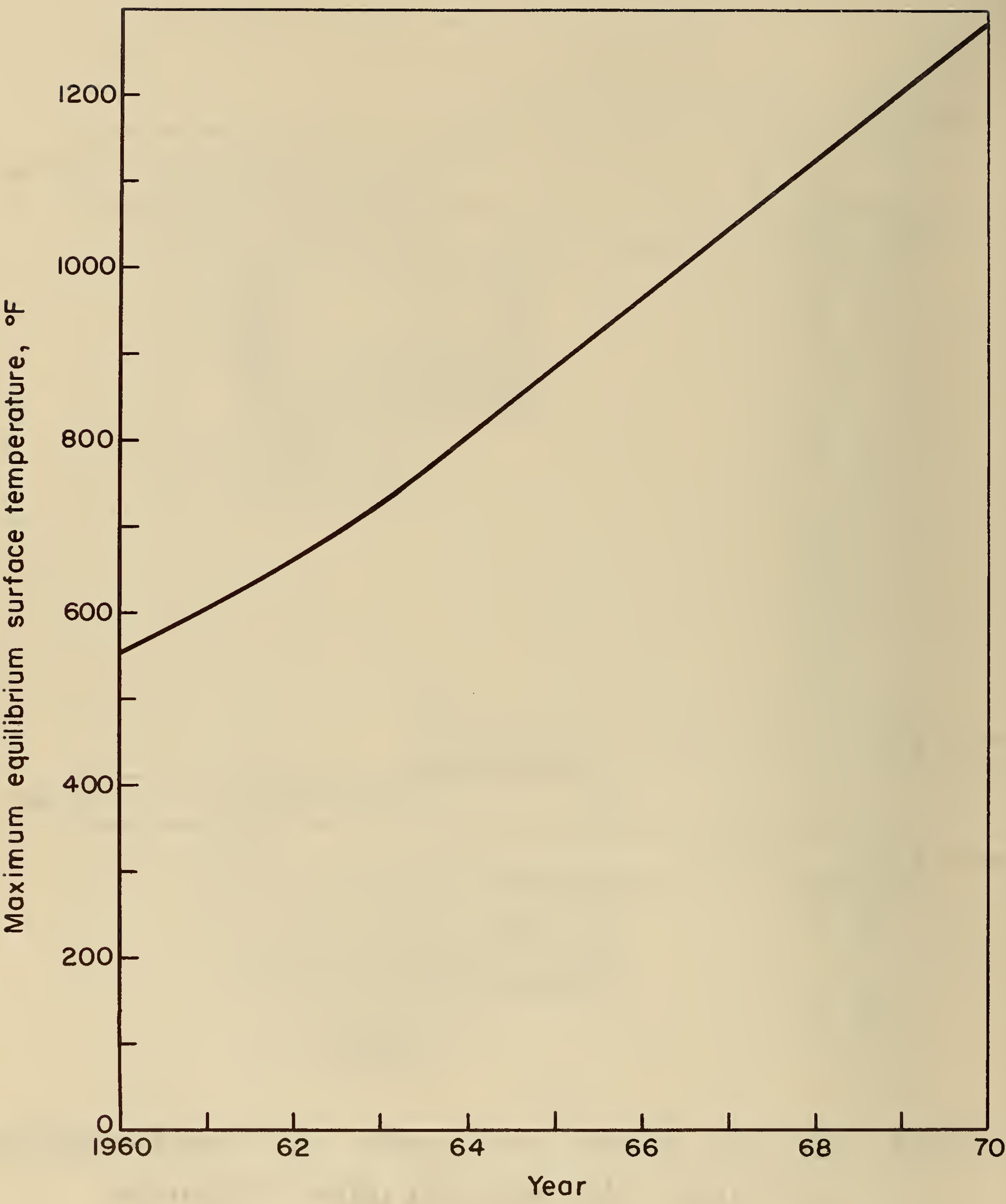


Fig. 5 Temperature trend for manned military aircraft in continuous flight.

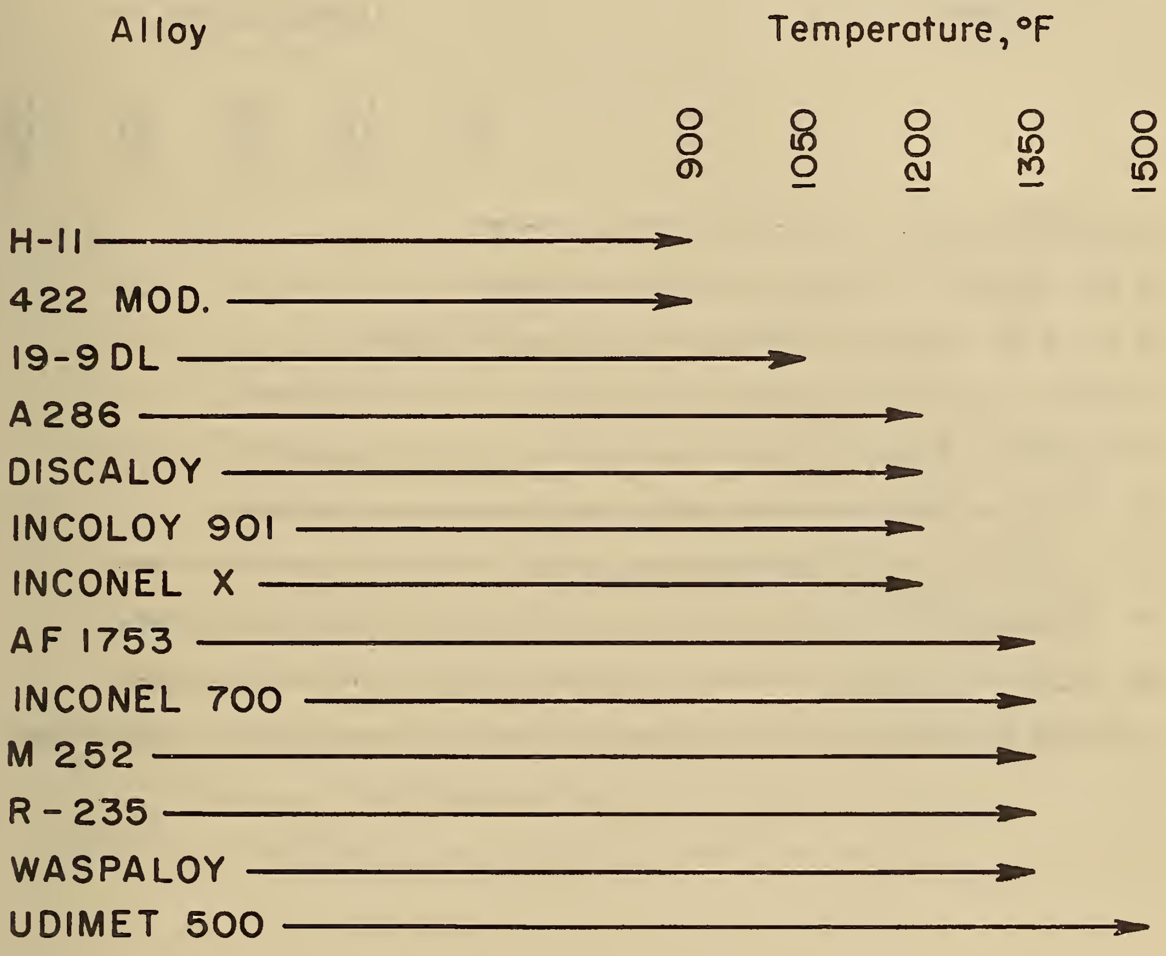


Fig. 6 Approximate temperature capabilities of commercially available aircraft bolts.

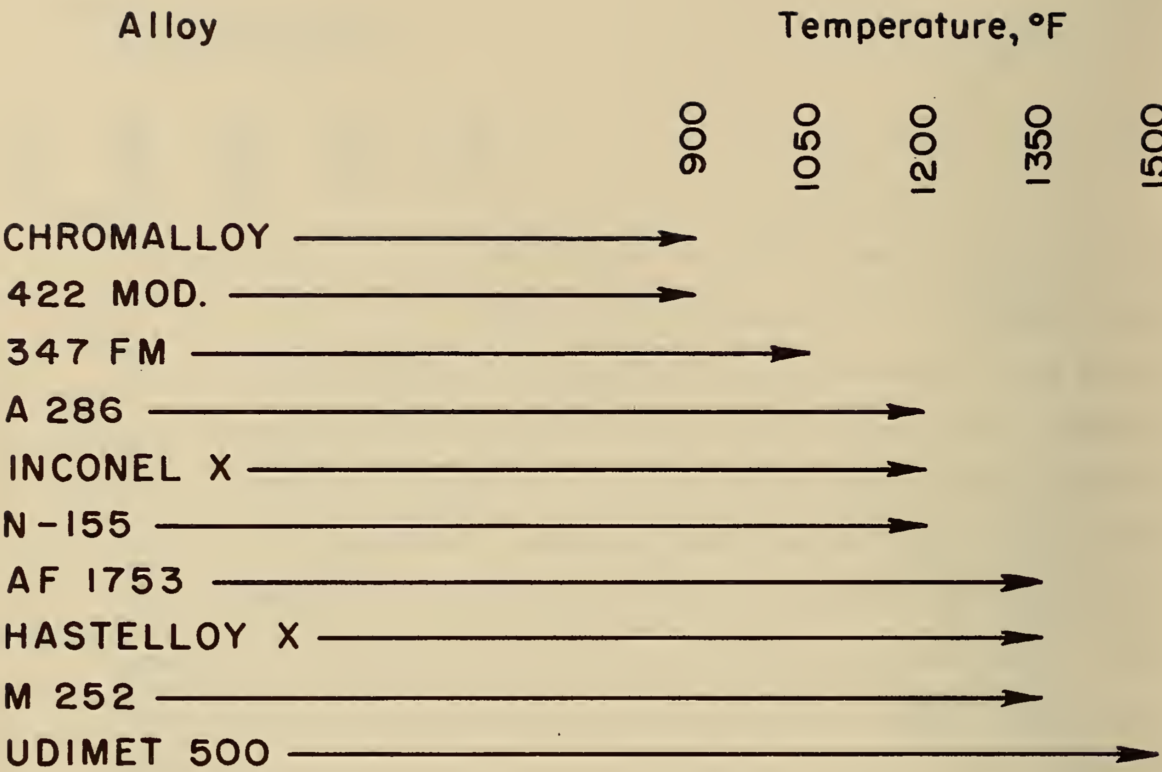


Fig. 7 Approximate temperature capabilities of commercially availible aircraft nuts.

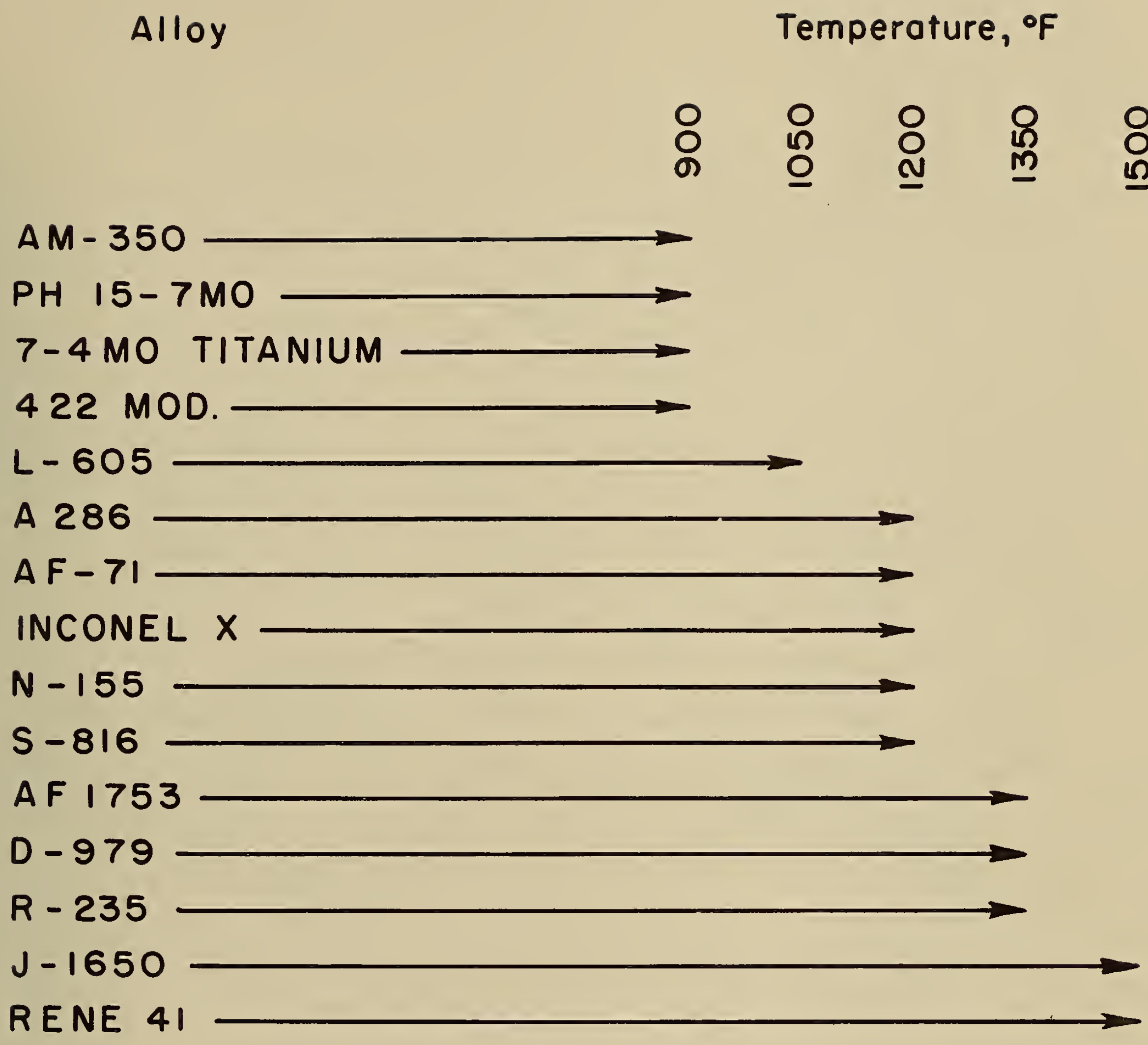


Fig. 8 Approximate temperature capabilities of several airframe materials.

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Radiation Physics. X-ray. Radioactivity. Radiation Theory. High Energy Radiation. Radiological Equipment. Nucleonic Instrumentation. Neutron Physics.

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Radio Propagation Engineering. Data Reduction Instrumentation. Radio Noise. Tropospheric Measurements. Tropospheric Analysis. Propagation-Terrain Effects. Radio-Meteorology. Lower Atmosphere Physics.

Radio Standards. High Frequency Electrical Standards. Radio Broadcast Service. Radio and Microwave Materials. Atomic Frequency and Time Interval Standards. Electronic Calibration Center. Millimeter-Wave Research. Microwave Circuit Standards.

Radio Systems. Applied Electromagnetic Theory. High Frequency and Very High Frequency Research. Modulation Research. Antenna Research. Navigation Systems.

Upper Atmosphere and Space Physics. Upper Atmosphere and Plasma Physics. Ionosphere and Exosphere Scatter. Airglow and Aurora. Ionospheric Radio Astronomy.