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Superseding AIR5926

White Paper to Support Supersession of MIL-S-8879C With AS8879C

RATIONALE

This document has been determined to contain basic and stable technology which is not dynamic in nature.

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FOREWORD

Introduction

The Department of Transportation Inspector General (DOT IG) issued an audit report in October 2000 (report number: AV-2001-003) that identified a disparity in test data of fasteners used in commercial aviation. As a result, the Federal Aviation Administration (FAA) conducted a thorough Fastener Audit to ascertain the safety of fasteners used in commercial aircraft and to investigate the cause of variation in measurement.

The FAA's audit (report number: FAA-IR-01-02) found that all fasteners meet their metallurgical and mechanical property requirements and concluded that there are no flight safety concerns involving aircraft fasteners. However, the audit did reveal that there were some deficiencies relating to industry practices regarding the use of fastener design specifications, thread inspection techniques, and quality assurance systems.

The FAA identified several non-safety related dimensional discrepancies involving fasteners. These discrepancies, often involving less than 1/30th the thickness of a sheet of paper, are the result of differing interpretations of existing standards for measuring screw thread dimensions. These standards are designed to promote the interchangeability of threaded parts, which are fabricated internationally and are not related to safety-critical functions.

The FAA recommended that industry develop consensus design and inspection specifications for threaded components and improve various elements of the quality assurance system for threaded components.

Screw Thread Conformity Task Force Formed to Improve Industry Practices

To respond to concerns, the General Aviation Manufacturers Association (GAMA), National Aeronautics & Space Administration (NASA), International Aerospace Quality Group (IAQG), Industrial Fasteners Institute (IFI), and the Aerospace Industries Association (AIA) held a meeting of industry stakeholders to review the FAA Fastener Audit report's conclusions and recommendations. Industry committed to form a working group to take up the issues raised by the FAA Audit and to improve industry practices regarding the design, inspection, and quality assurance of threaded components used in commercial airplanes. The working group was designated the Screw Thread Conformity Task Force (STC-TF).

The Screw Thread Conformity Task Force is comprised of engineering and quality representatives from each stakeholder group, including commercial airplane and engine manufacturers, general aviation manufacturers, fastener manufacturers, FAA, Department of Defense (DOD), and thread gage manufacturers.

The STC-TF developed a program plan and established two working groups to focus on items in the FAA report.

1. The Engineering working group will develop a clear screw thread specification that is acceptable to all industry-wide stakeholders to supersede the inactive MIL-S-8879C specification. The superceding document(s) shall provide for clarified design and acceptability requirements, compatibility/interchangeability with products that are manufactured to MIL-S-8879C, maintenance of intended thread form function, minimization of required part standard/drawing changes, and clarified inspection methods.

2. The Quality working group will develop, publicize, and propose a standardized quality system audit process for aerospace manufacturers of threaded components. This document will highlight best practices regarding quality assurance systems for production of threaded components such as control of suppliers, statistical sampling inspection plans, in-process controls, final inspection processes, and control/calibration of thread gages (see Section 1).

Purpose

The Aerospace industry established the STC-TF to address and respond to the findings and recommendations from the FAA Fastener Audit report:

- Standards/Engineering
  - Develop a single aerospace thread specification for UNJ thread form
  - Resolve language disparity: acceptable vs. conforming
  - Resolve min material measurement issue (cone & vee vs. best wire size)
  - Determine if 40% differential is still applicable
  - Address the net fit issue for screw thread pitch diameter

- Quality system findings
  - Supplier control
  - Flow-down of technical/quality requirements and use of standards
  - In-process, receiving, and final inspection/acceptance plans (sampling)
  - Inspection, maintenance, calibration and proper use of thread gages

Objectives

The aerospace industry (IAQG, AIA, GAMA, and IFI) has committed to FAA and DOT IG to evaluate FAA's findings and recommendations, and to develop effective corrective and preventative actions. The STC-TF has determined that the following products are needed:

1. Consensus thread specification to supercede MIL-S-8879C
2. Standardized criteria for the use and calibration of various thread inspection gages
3. Standardized quality system audit criteria for the manufacture of threaded components

Schedule

The industry has agreed to implement these corrective actions by publishing a nongovernmental standard by July 2002. The intent is that this standard will be a replacement for MIL-S-8879C. MIL-S-8879 is owned by DOD, but it is a defacto international standard, and DOD support is imperative for success. This aggressive schedule requires all stakeholders to focus on the issues identified in the FAA report and to participate constructively toward the development of consensus products. To avoid potential process delays, the STC-TF decided to adopt the "ground rules" developed by the engineering team:

The project timeframe was determined to be one of the most stringent issues. In an attempt to minimize influences that would delay the process, the following ground rules were set and agreed upon by the members of the team:

- Not returning a response to a ballot is an indication of complete approval.
- All comments and disapprovals must be accompanied by rationale for the comment and proposed alternate language.
- Resolution of comments and disapprovals shall occur by a 3/4 majority vote of the team members present. There is no quorum requirement, but only one vote per company in the case of a dispute.
- Communication of the importance of these decisions is key to the continued success of the project.

Development of Mission

The team discussed items important to development of a mission for the team. The following mission statement was created for the team to review:

This Task Team has been created to develop a clear supersession for screw thread specification MIL-S-8879C that is acceptable to all industry-wide stakeholders. The superceding document(s) shall provide for clarified acceptability requirements, compatibility/interchangeability with product that is manufactured to MIL-S-8879C, maintenance of intended thread form function, minimization of required part standard/drawing changes and clarified inspection methods.

The Task Team will develop recommendations regarding supersession path, implementation methods, and implementation schedule to facilitate ease of transition for all stakeholders. The Team shall also prepare communications to outline the reasoning and philosophy of recommended design changes should they be needed.

1. SCOPE:

This paper was prepared to support supersession of MIL-S-8879C with Screw Thread Conformity Task Force selected industry standard AS8879C, published by the Society of Automotive Engineers (SAE). Other documentation changes will be covered by separate papers. Separate papers are anticipated for thread gaging issues, and thread gage calibration procedures. The STC-TF decided that the thread design standard needed to be completed before thread gage definition could be addressed. Thread gage definition has to be known before calibration procedures can be addressed.

2. BACKGROUND:

2.1 History of Screw Threads:

Screw threads are a geometric shape that provides a boundary surface that is generally pictured as a formed helix. The screw thread shape is independent of material, heat treatment, plating, coating, tensile strength, etc. Screw thread geometry is never purchased as an entity. Screw threads are a geometric surface component of a part that may be manufactured or purchased, like a bolt, nut, or fitting. As such, screw thread standardization documents primarily fall into the class of documents known as design standards, rather than into the class known as product standards or specifications.

Threads are nothing more than an inclined plane wrapped around a cylinder. Archimedes is thought to have used a screw form to raise water over 2000 years ago. Joseph Whitworth applied himself to the standardization of the mechanical screw thread form in about 1833. His results came to be known as the British “Whitworth Standard”. In about 1864, William Sellers proposed standardization of screw threads in the United States. Sellers’ efforts resulted in a standard that was adopted by the U.S. government (the thread form became known as the American National thread form). World War II standardization efforts brought about adoption of the Unified thread form by America, the British Isles, and Canada (hence the term ABC standards).

Individuals, companies, and even countries may agree to use certain definitions for various products, geometries, sizes, and such, in order to ensure interchangeability. These definitions become standards, and published standards are the record of the agreements made. Screw thread standards exist for many kinds of threads, e.g., tapered pipe, Whitworth, buttress, acme. A bolt and a nut that are intended to mate generally need to have compatible thread forms. For aerospace applications, the primary thread form used in fastening has been the 60° included angle Unified Form.

Fastener designers discovered that providing a root radius on external Unified threaded bolts would improve fatigue properties when compared to Unified threaded bolts without root radius. This led to the UNR thread form where an external thread root radius within the established Unified envelope is mandatory. Increasing the size of the thread root radius beyond established Unified limits provided additional fatigue improvement. This eventually resulted in establishment of the UNJ thread form. The UNJ thread form is the Unified thread form with an increased minor diameter for both the external and internal thread, and includes a mandatory root radius on the external thread. This thread form is currently used as standard throughout the aerospace industry.

2.1 (Continued):

The UNJ thread form was first published as a military standard by the Aeronautical Standards Group (ASG) as specification MIL-S-8879 (ASG), dated 21 September 1960. By the time MIL-S-8879 revision C was published in 1991, this standard had wide spread use not only in the United States, but also worldwide. It was for all intents and purposes, an international standard. Several other specifications or standards have been published internationally that provide the identical tabulated geometry. The FAA did not request changes to these other standards.

In the mid 1990s, the DOD implemented acquisition reform policies and converted a huge amount of military standards and specifications to industry standards and specifications. As a consequence of acquisition reform, DOD made MIL-S-8879C inactive for new design without naming a replacement for this defacto international standard. The goal of the STC-TF team is to have a replacement standard designated by the DOD.

2.2 Determining Thread Dimensions:

The Unified Form thread is based upon the trigonometric relationships of a 60° equilateral triangle. For Unified Form threads, the height of the thread, H, is equal to 1/2 the thread pitch times the Tangent of 60°. Trigonometric relationships are indeterminate. This means that there is no fixed number of decimal places where all following digits are zero for all angles. Desktop computer calculators will provide the Tangent of 60° to 31 decimal places.

The number of decimal places used to calculate thread dimensions has an impact on the theoretical accuracy of the geometry that results. Historically, thread dimensions were calculated, long hand, using 6 decimal place values. (Desktop computer spreadsheets will make dimension calculations to at least 14 decimal places.)

Thread dimensions calculated to 6 decimal places were then rounded, by hand, to four decimal places for tabulation in standards. (Desktop computer spreadsheets provide a dozen ways to round, or truncate, numbers. ASME B1.30 rounding rules or recommendations do not necessarily agree with computer spreadsheet rounding choices.) The long hand calculation method provided a sufficient number of decimal places for most engineering purposes. For example, plotting four decimal place thread form on paper, to a scale such as 100X, provides illustrations pleasing to the eye. An example spread sheet plot is provided in Figure 1.

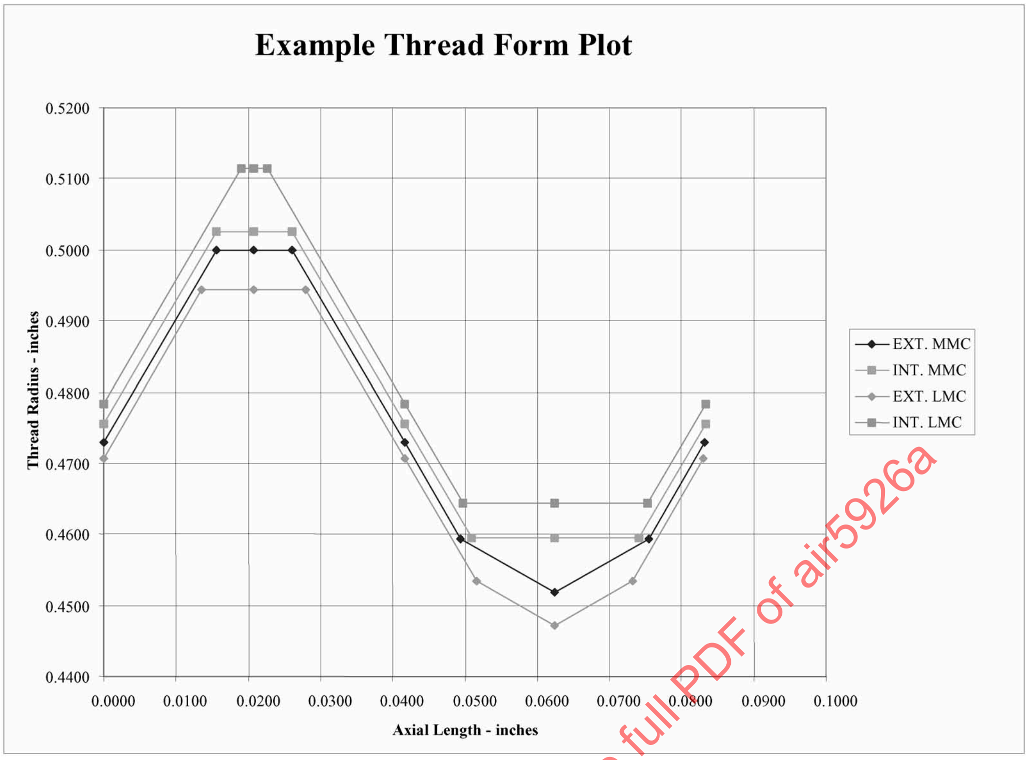

FIGURE 1 - Spread Sheet Thread Form Plot Using 4 Decimal Place x and y Coordinates

2.2 (Continued):

Plots using one decimal place would not be so pleasing to the eye, and plotting problems would be very noticeable as the pitch diameter points would appear out of line with the crest and root and the thread flank would not look straight, flat, or linear. If the one decimal place plot omits the pitch diameter data point, then the drawing appearance would be more acceptable. The point of the foregoing discussion is that the tabulated thread dimension values, in all the 60° screw thread specifications known to man, are not accurate – strictly speaking. Never the less, the results are adequate for engineering purposes.

2.2 (Continued):

Understanding the implications of this rounding issue is important in establishing an understanding of one source of variation seen in measured hardware threads. Let the intersection of the thread crest truncation cylinder and the thread flank be taken as one mandatory plot point, point “A”, and the intersection of the thread root radius tangent point with the thread flank be taken as the other mandatory plot point, point “B”. If this is done, then plotting the pitch diameter point, point “C”, in the graphic is unnecessary to the generation of the thread graphic. However, because pitch diameter is so ingrained in the definition of the thread it will necessarily remain part of the definition. Figure 2 illustrates a 4 decimal place plot of a thread flank. Note that points “A”, “B”, and “C” have individually calculated x and y coordinate values. If the graphic is constructed to provide an absolutely straight line between point “A” and point “B”, then the pitch diameter point “C” may be placed in the graphic at the appropriate x and y coordinate location. What you will notice is that point “C” will not be on the line connecting point “A” and point “B”. It will be slightly above or slightly below the line depending on the thread size, and depending on how the 6 decimal place calculation dimensions were rounded. Drawing scale will have a big impact on how noticeable this anomaly will be. It is noticeable at 100X if you are looking for it.

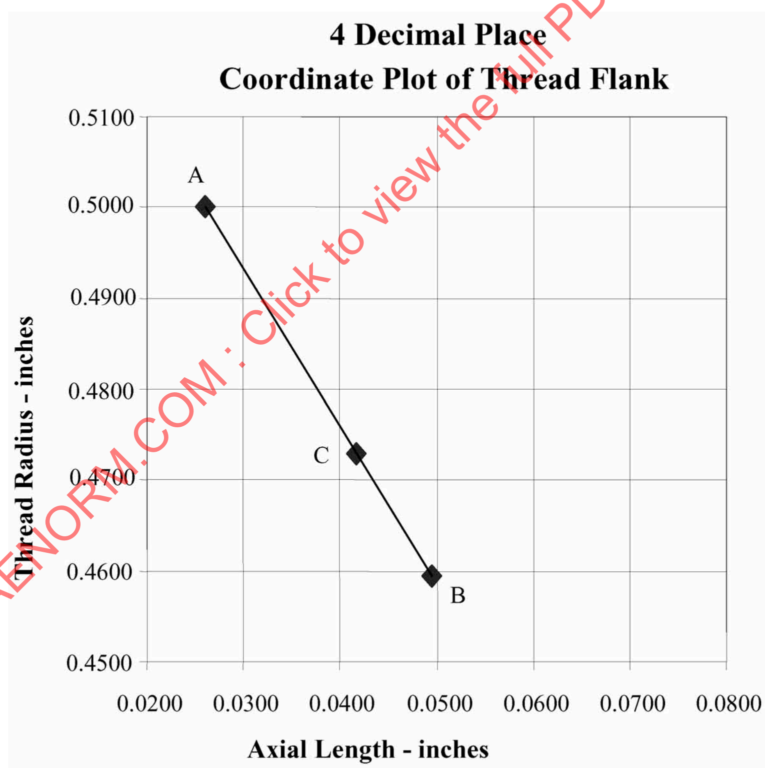

FIGURE 2 - Thread Flank Plot Using 4 Decimal Place Dimensions

2.2 (Continued):

Figure 3 illustrates a 2 decimal place plot of points A, B, and C. In this illustration the out of line condition of the pitch diameter plot point, C, is obvious.

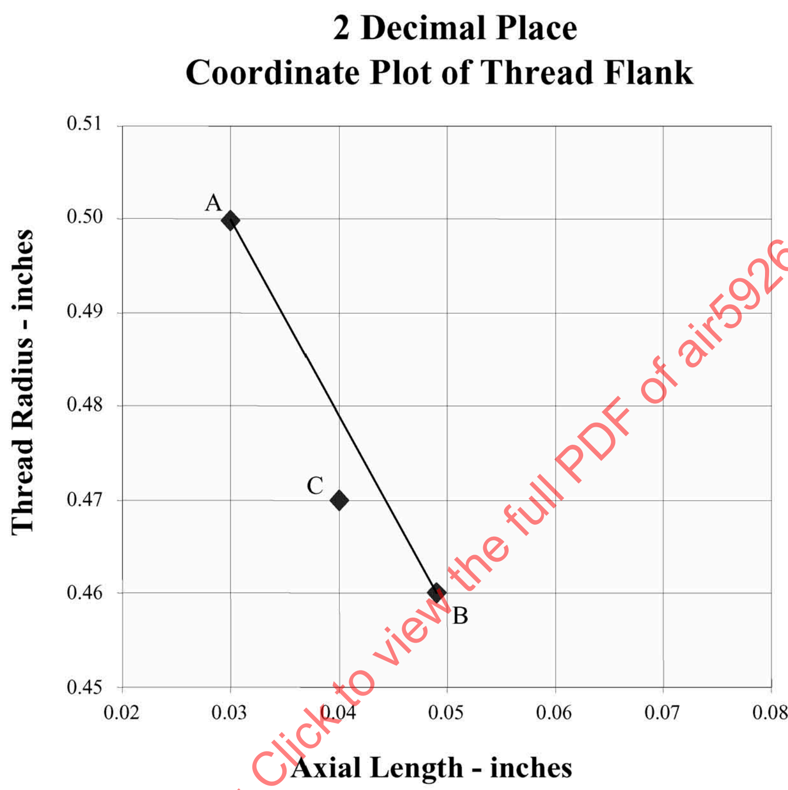

FIGURE 3 - Thread Flank Plot Using 2 Decimal Place Dimensions

2.3 Are The Number of Decimal Places Specified in the Standard An Issue?:

How important is the fourth decimal place value in the tabulated thread dimensions? The STC-TF examined this issue. After all, the previous specification values were determined by rounding 6 place computations. How big is 0.001 inch? A sheet of copier paper is about 0.003 inch thick. If a cylinder diameter varies by 0.003 inch it is easy to relate that variation to the thickness of a sheet of paper, and it is easy to visualize the impact on the measured diameter. It is a little harder to visualize when the diameter being discussed is thread pitch diameter. How much material would need to be removed from each thread flank in order to result in a pitch diameter change of exactly 0.003 inch? The answer is 0.00075 inch. Can you visualize it? Is the 5th decimal place answer necessary? Can you round it up (0.0008) or down (0.0007) to 4 decimal places and still have an exact pitch diameter change of 0.003 inch? What if you round it up or down to 3 decimal places?

2.3 (Continued):

A question posed to the STC-TF regarded the logic of retaining 4 decimal place dimensions for screw threads. It was suggested that the team reduce the number of decimal places for internal threads to 3 decimal places. A comparison relating methods of manufacture, thread inspection methods, and thread mechanical strengths was presented. External thread manufacturing techniques that include centerless grinding of the thread roll diameter, rolling of the threads, and inspection that uses three roll or segment style indicating gages yields tightly controlled product threads with excellent surface texture properties. External threads tend to be the weaker component in a nut-bolt combination, and design allowable loads are determined and published in MIL-HDBK-5 for bolts, but not nuts. Retaining 4 decimal place dimensions for external threads makes sense. Internal threads that are manufactured using drills for pretap hole preparation, thread taps that cut the thread form, the resulting relatively poor surface texture, and indicating gages that are very sensitive to operator technique do not support retention of 4 decimal place dimensions for internal threads. The STC-TF discussed the foregoing and voted to retain 4 decimal place dimensions for externally threaded product. The STC-TF supported the reduction of internal thread dimensions to 3 decimal places as a good engineering decision, but considered that doing so would be met with considerable industry resistance. As a result, the thread standard prepared by the STC-TF retained four decimal place dimensions for all threads.

2.4 Thread Dimensions When Coated, Plated, or Dry Film Lubricated:

During discussion of thread dimensions, the subject of coating and plating threads came up. AS8879 Revision B did not provide any before finish adjustment for major diameter, minor diameter, or root radius. Finish thickness is measured perpendicular to the surface upon which it is applied. For major diameter, and minor diameter the finish thickness will affect the diameter by 2 times the thickness (once on each side). For pitch diameter, which is determined on a thread flank, the finish thickness will affect the diameter by 4 times the finish thickness ($\delta d = 2 \times \delta t/\text{Sine } 30°$). Minimum finish thickness choices vary from finish specification to finish specification. The limitation for pitch diameter has always been "no more than 0.001 inch for threads with pitch diameter tolerance not exceeding 0.0035 inch" (where the pitch diameter tolerance is greater than 0.0035 inch the allowance has been larger). The implication is that the thread specification is over coding (contradicting) the finish specification. Notice that for Class 1 Cadmium plate, the maximum permitted major diameter or minor diameter adjustment would be 0.001 inch (2 times 0.0005). For pitch diameter, the adjustment should be 0.002 inch (4 times 0.0005), but is limited by the thread specification to 0.001 inch. Even Class 2 Cadmium plate should require a pitch diameter reduction of 0.0012 inch – not 0.001 inch. In order to meet both the thread specification maximum allowed reduction, and the minimum plating thickness requirement, the thread manufacturer must increase his acceptable minimum after finish pitch diameter to ensure conforming product by the difference - 0.0002 inch (for Class 2 thickness). The only change to AS8879C dimensions that the STC-TF considered acceptable was to incorporate recognition that finishes affect all thread dimensions, not just pitch diameter.

2.4 (Continued):

The minor diameter may be reduced by 2 times the minimum plating or coating thickness. Root radius is not mentioned. In order to maintain a “mathematical model” it would be logical to adjust the root radius. Plating or coating thickness in external thread roots will only impact the radius dimensions in a 1 to 1 ratio. In other words, plating or coating thickness will only impact the radius by the thickness of a single layer. Consequently, adjusting the external thread root radius by the equivalent of the minimum plating or coating thickness would be logical. Adding a coating or plating to an external thread root radius tends to make the radius smaller. Therefore, the STC-TF recommends that the manufacturer "do the math". However, it is necessary to observe the minor diameter requirements, both before and after application of coating or plating, at the same time. Note that the thread specification uses the term "may be" reduced so that it does not mandate the reduction. The paragraphs in AS8879C mandate dimensional conformance after plating and coating.

2.5 Conformance and Performance:

Inspection verifies conformance. In a manner of speaking, inspection and measurement of thread dimensions is inspection, measurement, and verification of technically imprecise requirements.

Ultimately, the question is: will the threaded product perform its intended function? Testing verifies performance. There are multiple intended functions. Will the threaded product assemble with its mating part? Will the threaded product hold parts together? Will the threaded product resist loads in service?

2.6 Assembly:

Will the mating threaded products assemble? There are two ways to answer this question. First, you may inspect both threaded products for dimensional conformance. Assembly is generally assured when product threads conform to the stated requirement – that they not exceed maximum material limits. Second, you may attempt assembly of mating products.

Assembly is an issue as mating product thread definition in MIL-S-8879C, at maximum material condition, is net fit. Oversize external or undersize internal threads will not assemble in a desirable manner. However, threaded product that is substantially outside of size limits (i.e., undersize external or oversize internal threads) will assemble without difficulty. For example, a .1640-32 threaded screw will assemble with a .1900-32 threaded nut. This combination assembles, but has no strength capability as the threads may be disengaged, without rotation, using your fingers.

2.6 (Continued):

Verification that product threads will assemble using the measurement method is labor intensive and requires inspection of four external thread dimensions, and inspection of three internal thread dimensions. The inspection of thread dimensions is important to the threaded product manufacturer because there is no profit in producing scrap. If the threaded product will not assemble because it is oversize, the customer will certainly find out and the manufacturer will incur the loss. Similarly, if the threaded product is significantly out of tolerance, it will not sustain loads induced when the parts are installed. Consequently, the threaded product manufacturer needs to utilize thread inspection techniques that will assure dimensional conformity. The threaded product customer does not want to run out of product on the assembly line. Consequently, the threaded product customer needs product procurement and acceptance procedures that will assure acceptance of suitable products and prevent receipt of products with potential problems. Aerospace industry threaded product manufacturers do have procedures in place that ensure high quality in-tolerance product will be produced. Aerospace industry customers do have procurement and quality procedures in place that ensure high quality intolerance product will continue to be made by their suppliers.

2.7 Performance:

A manufacturer, or customer, of threaded product would do well to consider dimensional inspection as only a portion of a complete product inspection process. A complete product inspection process should include visual inspection, dimensional inspection, mechanical property tests, metallurgical tests, and interaction or installation tests when applicable. For threads, an initial visual inspection is important. Does the product have a thread form that looks like you expect it to look? A visual inspection can reveal gross defects. A threaded product, like a bolt, that is shipped with missing threads is an example of a product with a gross thread defect. It happens. Visual inspection can reveal malformed or poorly formed threads, burrs and slivers in threads, and similar defects. Visual inspection will find these kinds of defects faster than known measurement methods will. Mechanical property tests, like tensile tests, will also find product defects, such as low thread stripping strength. Low tensile property test results may indicate some problem with the product thread other than dimensional nonconformance. Low strength is generally due to improper heat treatment. Low strength may occur for any length of thread engagement if the part heat treatment is wrong. Low fatigue test results can indicate improper heat treatment, or missing head-to-shank fillet roll, or improperly rolled threads, or metallurgical defects, like laps in the thread. Low fatigue life is never due strictly to a tiny dimensional nonconformity. Sustained load tests, like stress rupture testing, can reveal embrittlement problems, bad heat treatment, and even wrong material. Sectioning the part axially and conducting a metallurgical examination may also find defects in the product thread. Metallurgical grain structure defects, laps, seams, cracks, bursts, voids, and inclusions are all revealed in a metallurgical examination. Interaction between mating product threads can be tested by using a known acceptable mating product to check the product being inspected for acceptance. Mechanical property defects and metallurgical defects are of much greater concern to engineering than tiny dimensional nonconformities that are on the order of a few ten thousandths of an inch. The effect of tiny dimensional nonconformities on mechanical property tests is lost in the natural scatter exhibited by the test results.

2.7 (Continued):

Recently there has been some concern expressed in technical literature regarding the susceptibility of threads to loosen in a vibration environment. Some conclusions expressed in the literature have blamed the loosening on out of tolerance threads. It is a known fact that given the correct conditions all threads will loosen. It does not depend on an out-of-tolerance condition. Individuals that believe they have discovered a new phenomenon may have never studied the mechanics of a mass on an inclined plane. A mass sliding on a sloped surface is an elementary textbook Engineering problem. See for example: Mechanical Engineering Design, by Joseph Edward Shigley, published by McGraw-Hill Book Company. When the forces acting on a mass, that tend to move the mass along the slope, exceed the forces tending to resist motion, then the mass will slide on the slope. It should be noted that Shigley shows that screw threads will not self-loosen when the coefficient of friction is greater than the tangent of the lead angle. Shigley goes on to say that the thread flank angle generates a wedging action that further increases the frictional force resisting self-loosening. Figure 4 illustrates the forces on the thread pressure flank for a small element of the thread form. In Figure 4, F_{NF} represents the component force normal to the pressure flank, F_{NL} represents the component force normal to the lead angle, and F_{BP} represents the component force projected in the plane of the nut's bearing surface. The lengths of the arrows, representing the forces acting on the thread pressure flank, are proportional to the actual loads. The scale used to generate Figure 4 is so small that an enlarged view is necessary to show the force acting in the loosening direction down the slope of the thread helix. The non-pressure thread flank does not contribute to vibration loosening. By extension, this illustration could be modified and used to explain why threads that have an intentional interference fit are not in general use. Interference fit between both pressure and non-pressure thread flanks would substantially increase the radial forces in the mated threads.

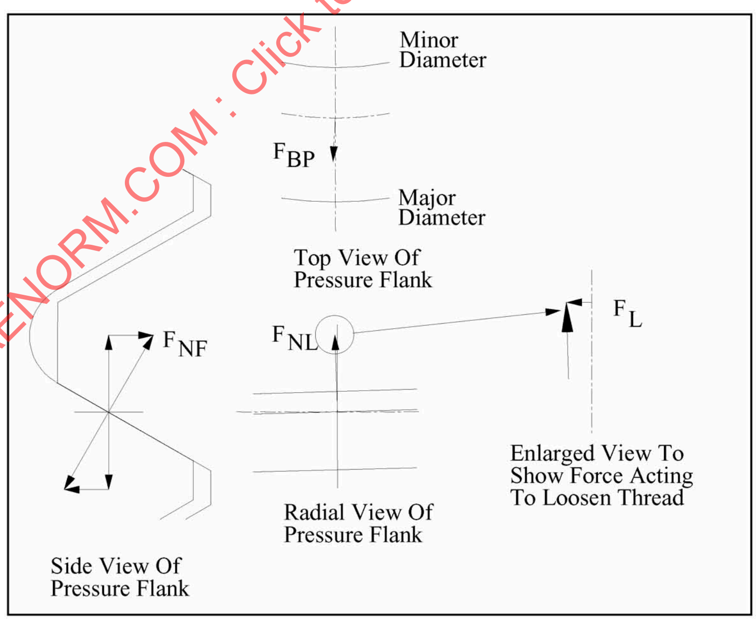

FIGURE 4 - Forces on Thread Pressure Flank

2.7 (Continued):

At least five industry standard test methods exist for evaluating vibration resistance of threaded products. The oldest is the Stanley Hammer Test, MS26531, dated 3 January 1956. At least one new test method is described in current technical literature. All six methods will loosen mated threads. These test methods have verified the results of the textbook mathematical analysis. Using a simple vibration test to assess threaded product performance in service is rather simplistic. None of the existing standard vibration test methods simulate real world aircraft structures, mechanisms, or service conditions. If you have ever seen any of these vibration tests in action, you will have no trouble imagining how undesirable it would be if a passenger seat shook like that. Aircraft manufacturers intentionally design structures that minimize vibration.

It should be pointed out that threaded components used in constructing an aircraft, for example, are subjected to a lot of different kinds of “vibration tests” before the aircraft is certified for use. Individual structural fasteners may be subjected to “standard” vibration tests as part of their qualification process. For example, the self-locking nut specification MIL-N-25027 contained a standard vibration test as part of the qualification process when it was first issued in 1954. Since then, MIL-N-25027 has been converted to industry specification NASM25027 that still contains a vibration test. The tension-tension fatigue test that bolts are required to pass is conducted using non-self-locking nuts. This test is discontinued at 130,000 load cycles. There has never been a report of the test nut coming off the bolt during this test. A product that successfully passes all the inspections and tests in the part procurement specification becomes a “qualified product”.

Threaded product qualification for use in aircraft design only begins with passing qualification inspections and tests as described in the product specification. Once that has been accomplished, an aircraft structural fastener, for example, must be tested and pass joint fatigue tests. Joint fatigue tests simulate structural joints that have proved to be suitable for design use, and poor joint designs are discarded. There are typically two kinds of joint fatigue tests. Low load transfer fatigue tests and high load transfer fatigue tests. Low load transfer fatigue tests simulate a structural joint as might be found in the center of a large panel where it would attach to supporting structure. High load transfer fatigue tests simulate a structural joint as might be found at the edge of a large panel where it attaches to supporting structure and abuts an adjacent panel. For use in either situation, a new fastener must meet or exceed existing fastener design fatigue test results. These joint fatigue tests are a form of vibration test where known loads are imposed on the joint materials and transferred through the fastener between the joint materials. New configuration joint designs are fatigue tested and some tests have revealed joint configurations where fasteners do tend to disassemble. Such designs are discarded. For designs that are unique, entire panels or major pieces of structure may be constructed and subjected to fatigue tests. These fatigue tests simulate service loads. When a passenger aircraft design has been completed, a fatigue test airplane is built, and subjected to multiple life cycle fatigue tests. Life cycle fatigue tests consist of mounting the entire aircraft in a custom fixture that applies service loads to all parts of the aircraft. All the important parts of the aircraft are instrumented in order to keep track of actual loads and to warn of imminent failures. These tests are full spectrum fatigue in that they simulate taxi, take-off, climb, cruise, decent, landing, and taxi to terminal. These tests are conducted until the number of flight cycles exceeds multiple life times for the aircraft. Similar tests are conducted on the aircraft engines. Figure 5

2.7 (Continued):

illustrates the test fixture(s) necessary to fatigue test a commercial jet airplane. Testing does not stop at this point. Extensive flight tests are conducted. Flight tests include flutter testing to evaluate sudden onset large amplitude vibration. Lastly, it is common practice to cut test panels out of old retired aircraft and conduct further fatigue tests to determine how much life might still be in the structure after it is taken out of service. The reason for testing retired aircraft structure is to provide data for operators of similar aircraft that are still in service. Consequently, design Engineers routinely consider vibration effects, and structural aircraft joints are designed so they will resist coming apart.

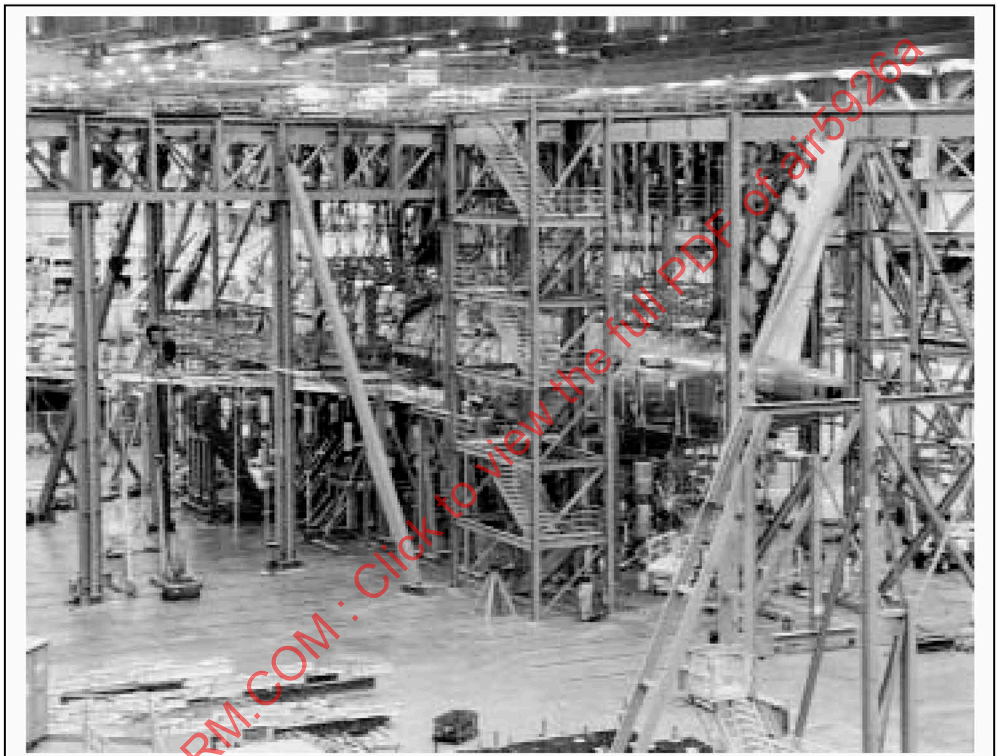

FIGURE 5 - Typical Fatigue Test Setup for Wide Body Jet Transport Aircraft

It should be noted that standard fatigue tests do not necessarily introduce the frequency and vibration amplitude that is seen in service. Some fatigue tests are typically run at approximately 5 cycles/second whereas vibrations seen in service typically cover a wide range of frequencies including much higher frequencies which are more likely to loosen a threaded fastener. It is apparent that fatigue tests do not obviate the need for fastener vibration tests. Though traditional fastener vibration tests may be much more severe than the actual service environment seen by the fastener, the duration of the test is very limited. (MIL-N-25027 requires 30,000 cycles at 1800 cycles/min or 17 minutes total duration.)

2.7 (Continued):

In general, fastener preload cannot be counted on to provide frictional forces that will prevent the free running threads from vibrating loose. Fastener preload in service can be minimal for a variety of reasons such as:

1. Failures to apply correct torque to fastener during production or maintenance.
2. Relaxation of joint materials such as form in place gasket or liquid shim.
3. Varying coefficient of thermal expansion between fastener and parent material.
4. Wear or corrosion.

All of the foregoing items constitute reasons that all threaded fasteners should have general means of safetying (safetied). For aerospace applications, the use of these safety devices is controlled by NASM1515, Requirement 111 (formerly MIL-STD-1515, Requirement 111). Applications where preload must be maintained require a closely controlled installation torque, verification, torque striping of the fastener, and a joint design that will maintain preload. These kinds of applications are a small percentage of all threaded fasteners on airplanes.

2.8 Dimension Check:

The tabulated dimensions in MIL-S-8879C may be recomputed using current desktop computer spread sheet software. Once the spread sheet is set up to simulate the long hand calculation method, the computed results agree with the existing requirements, although a few dimensions are different by 0.0001 or 0.0002 inch. The dimensional anomalies that are found are likely attributable to manufacturing needs that were applicable in the 1950s. Although some thought was given to correcting these tiny discrepancies, there was no overriding reason to do so. Consequently, the STC-TF proposal is to maintain tabulated thread dimensions on a one for one basis. Major errors in tabulated dimensions that were found in MIL-S-8879C were previously corrected in AS8879B.

2.9 Changes Proposed for Incorporation in AS8879C and STC-TF Team Decisions:

2.9.1 Deletions: The first proposed change that was identified was the elimination of thread "application categories". Thread application categories was an invention of revision C of MIL-S-8879. The two categories that were specified are "Safety Critical" and "Other". Designating a screw thread as safety critical implies that the threaded part application has critical thread performance requirements when in service. Historically, parts that have critical performance requirements have been defined using a control drawing. DOD mandated these requirements on their subcontractors by imposing drawing requirements in accordance with the rules in DOD-STD-100. No such definition requirements were imposed for thread category safety critical. The DOD did designate about 800 parts, (AN, MS, NAS, and OEM) as having safety critical applications, but did not issue control drawings for any of them. These parts are purchased to special instructions that are specified as part of a purchase order. For these applications, DOD uses National Stock Numbers (NSN) to control parts. The problems associated with this method should be obvious. First, the part marking remains the same as part numbers that have not been designated as having safety critical threads. Once separated from its paper work there is no known way to tell the safety critical thread category part from the non-safety critical thread category part. The potential for installing a category "other" threaded standard part in place of one made to thread category "safety critical" requirements is great. Once installed, a standard part purchased as a safety critical thread category part looks identical to one without special procurement provisions. None of the STC-TF team members were able to identify any standard part drawing that specifies thread category "safety critical". DOD has indicated that there are no standard part drawings that specify thread application category "safety critical", but they do impose thread application category safety critical lot acceptance inspection by purchase order requirement. DOD also stated that they would continue their current procurement practices. The STC-TF team agreed that discussion concerning safety of threaded parts is appropriate in a design manual, or design guidance specification, such as NASM1515, requirement 119. The STC-TF team agreed that DOD should provide instructions on the supersession notice that states which new requirement replaces which old one. The STC-TF team agreed that application categories "safety critical" and "other" should be discontinued. The revision of AS8879 includes two requirement "categories", and two conformance "categories". The default for design and conformance will be Category 1. Category 1 applies to the majority of all parts manufactured. Where increased dimensional scrutiny is required, Category 2 may be specified. Category 2 must be specified on the part drawing or procurement documents when it is needed.

Leaving the Length of Engagement and Tolerances paragraph in Section 3, Requirements, implies that a conformity check would be needed. The STC-TF team could not conceive of any conformity check that made sense. As this information is more or less background to explain the basis for dimensions and tolerances listed in AS8879, it is reference information, rather than a requirement. Hence, it was moved to Section 6, Information for Guidance Only.

2.9.2 Form Variation: Evaluation of form variations was necessary in order to address concerns regarding the 40% differential rule. MIL-S-8879 in paragraph 4.4.2.1, flag note 1 stated: If the differential between "GO" functional size and pitch diameter size does not exceed 0.4 of the pitch diameter tolerance, inspection of flank angle and lead (including helix variations) is not necessary.

For Category 1 threads, the STC-TF agreed that if a product external thread fully enters a threaded ring gage and a product internal thread allows full entry of a threaded plug gage, then inspection of product thread for form deviations was unnecessary. None of the STC-TF team members could provide a reason for determining geometric form deviations for a product thread. Even if the values were measured and recorded, what do you do with them once you have them? What value are these values? Surely, there must be a valid use for knowing the geometric form deviation values of a product thread. Academic interest is nice, but adds no beneficial value to a product thread. The magnitude of the form deviations is, of necessity, tiny. Suppose an inspector does determine all of the form deviations. There are two possible out comes. First, the form deviations are nearly zero, and the individual form deviations meet standard recommendations. The threaded part would be accepted. Second, some or all of the form deviations are relatively large, and do not meet standard recommendations. If the product thread passes inspection with the GO gage, the part would be accepted. If the thread does not pass inspection with the GO gage, then the part will be rejected regardless of the magnitude of the individual form deviations. From a final product thread standpoint, the STC-TF could see no value in inventing a use for knowing product thread form deviation values. From the above definitions and rationale, the STC-TF concluded that conforming product thread to the 40% differential rule is not required for Category 1 threads.

ASME Y14.5M-1994, Dimensioning and Tolerancing, states in multiple places that form variations combine to provide a functional (virtual) size larger than single element (actual) size. The team agreed that functional size would be acceptable if the product thread was accepted by the appropriate GO thread gage, or maximum material indicating gage. Indicating gages may be used to determine where the thread form is between minimum and maximum limits when that information is needed.

2.10 Discussion of Screw Thread Size Limits and Form Variations:

The STC-TF determined that the differences between tabulated thread feature diameters and geometric form errors needed clarification. ASME B1.7M, Nomenclature, Definitions, and Letter Symbols for Screw Threads, provides a comprehensive description of screw thread terminology. Additionally, to aide in understanding the differences between feature dimensions and geometric form errors, the following definitions were expanded and/or discussed. As a first step to understanding the differences between feature dimensions and geometric form errors the following definitions were put together.

2.11 Definitions (see 2.10):

2.11.1 Size: ASME Y14.5M-1994 does not provide a definition of this term. Note that size may mean different things to different people. For example, a thread "size" of .2500-28 UNJF-3A means something entirely different from a pitch diameter "size" of 0.2243 to 0.2268 inch. MIL-S-8879C uses the terms "GO" functional diameter size and pitch diameter size, but does not use the term "NOT GO".

2.11.2 Limits of Size: Unless otherwise specified, the limits of size of a feature prescribe the extent within which variations of geometric form, as well as size, are allowed. This control applies solely to individual features of size defined in paragraph 1.3.17. (From ASME Y14.5M-1994, paragraph 2.7.)

2.11.3 Individual Feature of Size: One cylindrical or spherical surface, or a set of two opposed elements or opposed parallel surfaces, associated with a size dimension. (From ASME Y14.5M-1994, paragraph 1.3.17.)

2.11.4 Limits of Size for Screw Threads: With respect to screw thread limits of size, no portion of the complete thread should be permitted to project beyond the envelope defined by the maximum-material limits on the one hand, or beyond that defined by the minimum-material limits on the other, and thus be outside of the tolerance zone specified.

2.11.5 Individual Element Variation: The diameter equivalent of the variation of any given element, except pitch diameter and flank angle, should not exceed 0.4 of the pitch diameter tolerance. The diameter equivalent of the variation of flank angle should not exceed 0.3 of the pitch diameter tolerance for flank engagements of .5625H. The full pitch diameter tolerance should not be used as pitch diameter manufacturing limits unless deviations in all other elements are zero. (Individual element variation determination is only applicable to Category 2 threads).

2.11.6 Form Variation: A form variation is a deviation from perfect form. The recognized form variations include lead variation, flank angle variation, circularity variation, cylindricity variation, helix variation, and straightness variation.

2.11.7 Dimensioning Screw Threads: Methods of specifying and dimensioning screw threads are covered in ANSI Y14.6 and ANSI Y14.6aM. (From ASME Y14.5M-1994, paragraph 1.8.19.) Each tolerance of orientation or position and datum reference specified for a screw thread applies to the axis of the thread derived from the pitch cylinder. Where an exception to this practice is necessary, the specific feature of the screw thread (such as major diameter or minor diameter) shall be stated beneath the feature control frame, or beneath or adjacent to the datum feature symbol, as applicable. (From ASME Y14.5M-1994, paragraph 2.9.)

2.11.8 Form and Orientation Tolerances: Form tolerances control straightness, flatness, circularity, and cylindricity. Orientation tolerances control angularity, parallelism, and perpendicularity. (From ASME Y14.5M-1994, paragraph 6.2.) Form and orientation tolerances critical to function and interchangeability are specified where the tolerances of size and location do not provide sufficient control. (From ASME Y14.5M-1994, paragraph 6.3.)

2.11.9 Form Tolerances for Individual Features:

2.11.9.1 Circularity: Circularity is a condition of a surface where, for a feature other than a sphere, all points of the surface intersected by any plane perpendicular to an axis are equidistant from that axis. (From ASME Y14.5M-1994, paragraph 6.4.3.) The circularity tolerance must be less than the size tolerance. (From ASME Y14.5M-1994, paragraph 6.4.3.1.) For screw threads, circularity is a roundness condition of the screw thread pitch diameter cylinder. Thread pitch diameter circularity should be limited to one-half the pitch diameter tolerance, where the total pitch diameter tolerance is less than 0.004 inch, and round within 0.002 inch where the total pitch diameter tolerance is 0.004 inch or larger. For screw threads, the pitch diameter cylinder is imaginary, i.e., the actual physical pitch diameter only exists at discreet points on any plane perpendicular to the thread axis. (This is only a conformance requirement for Category 2 threads.)

2.11.9.2 Cylindricity: Cylindricity is a condition of a surface of revolution in which all points of the surface are equidistant from a common axis. (From ASME Y14.5M-1994, paragraph 6.4.4.) The cylindricity tolerance must be less than the size tolerance. (From ASME Y14.5M-1994, paragraph 6.4.4.1.) For screw threads, cylindricity is a condition of the pitch diameter cylinder. For screw threads, taper is a conical condition of the pitch diameter cylinder, based on a specified length of engagement, and should be limited to 0.4 of the pitch diameter tolerance. (This is only a conformance requirement for Category 2 threads.)

2.11.9.3 Straightness: Straightness is a condition where an element of a surface, or an axis, is in a straight line. (From ASME Y14.5M-1994, paragraph 6.4.1.) For screw threads, straightness is a condition of the screw thread axis. The straightness tolerance must be less than the size tolerance. Since the limits of size must be respected, the full straightness tolerance may not be available for opposite elements in the case of waisting or barreling of the surface. (From ASME Y14.5M-1994, paragraph 6.4.1.1.1.) For screw threads, the straightness of the thread axis cannot be determined directly, but a drunken helix is one indication of an out of straight condition. (This is only a conformance requirement for Category 2 threads.)

2.11.9.4 Flatness: Not applicable to screw threads.

2.11.10 Form Tolerances for Related Features: Form tolerances for related features control position and orientation of features one to another. (Paraphrased from ASME Y14.5M-1994, paragraph 6.6.)

2.11.11 Lead Deviation: Lead deviation is a location variation. Lead is the number of thread starts divided by the number of threads per inch. For single lead threads, lead is the same as thread pitch. Both thread lead and thread pitch are basic dimensions and are listed in AS8879, Table 1. The diameter equivalent of variations in lead (including helix deviations) should not exceed 0.4 of the total pitch diameter tolerance. Deviation in thread lead, or pitch, is referred to as lead variation. For aerospace practice, for threads rolled after heat treatment, a small compressed lead may provide some enhancement of bolt thread fatigue properties. (This is only a conformance requirement for Category 2 threads.)

2.11.12 Flank Angle Deviation: Flank angle deviation is an orientation variation. Flank angle is measured between the thread flank and a plane perpendicular to the thread axis on an axial plane. The flank angle for Unified threads is 30°. Flank angle is a basic dimension. Flank angle is also known as basic half angle of the thread. The diameter equivalent of deviations in flank angle, for flank engagements of .5625H, should not exceed 0.3 of the total pitch diameter tolerance. Flank angle deviation is also referred to as flank angle variation. (This is only a conformance requirement for Category 2 threads.)

2.11.13 Formulas: Formulas for lead deviations equivalent to 0.4 of the pitch diameter tolerance and the flank angle deviations equal to 0.3 of the pitch diameter tolerance are included in this white paper, but not in AS8879C.

2.11.14 Diameter Equivalents of Lead Deviation: Lead deviation is only meaningful for an axial length of multiple pitches. Lead deviations equivalent to 0.4 of the pitch diameter tolerance may be determined from the following formula. Note that the pitch diameter increment may be determined using differential gaging techniques, and may be a selected percentage of the pitch diameter tolerance (0.4 times the pitch diameter tolerance is generally used). MIL-S-8879A included lead variation (for American National thread form) in the appendix, but the appendix does not appear in MIL-S-8879C. See FED-STD-H28/2, Revision None, paragraph 8.2.

$$\delta p = \frac{\delta E}{\cot\alpha} = \delta E \tan\alpha = \frac{\delta E}{1.7321} = \delta E \times .57735 \qquad \text{(Eq. 1)}$$

where:

δE = Pitch diameter increment due to lead deviation
δp = The maximum pitch deviation (lead variation) between any two of the threads engaged
α = Basic half angle of thread

2.11.15 Diameter Equivalents of Flank Angle Deviations: Diameter equivalents for flank angle deviations may be determined from the following formula. Note that the pitch diameter increment may be determined using differential gaging techniques, and may be a selected percentage of the pitch diameter tolerance. (0.4 times the pitch diameter tolerance is used for flank engagements of .75H, and 0.3 times the pitch diameter tolerance is used for flank engagements of .5625H.) See FED-STD-H28/2, Revision None, paragraph 8.1. Flank angle deviations for flank engagements of .5625H and equivalent to 0.3 times the pitch diameter tolerance are essentially the same values for the standard fine thread bolt sizes as were tabulated in MIL-S-8879A, Appendix A (which tabulated flank angle deviations equivalent to 0.4 times the pitch diameter tolerance for American National thread form).

$$\tan\delta\alpha = \frac{\delta E}{1.125p} \qquad \text{(Eq. 2)}$$

where:

$\delta\alpha$ = Variation in half angle of the thread
δE = Pitch diameter increment due to deviation in half angle (flank angle variation)
p = Thread pitch

2.11.16 Combined Variation: If every element that makes up the thread form is assumed to vary from perfect form, then the sum of all the individual element variations plus all the orientation variations plus the actual variation in size will generate the functional size of the thread. The combined effect of the variations is cumulative. If this condition were expressed mathematically for the pitch diameter, the relationship would be:

Actual Functional Pitch Diameter = Single Element Pitch Diameter Size + Circularity variation + Cylindricity variation + Straightness variation + Lead variation + Flank Angle variation.

Functional Pitch Diameter is the same as Functional Size, and the limit is the same as Maximum Material Condition (MMC) envelope. The upper limit of maximum material condition is equal to the "GO" gage size.

As an aside, it had been noted that the Appendix in MIL-S-8879 Revision A and the Appendix in AS8879 Revision B were both misleading. These Appendices provided Allowable Lead and Angle Deviations for tabulated diameter-pitch combinations equivalent to 40% of the pitch diameter tolerance. These Appendices were both based on the American National thread form. Neither one was computed based upon the UNJ thread form. Both listed thread diameter-pitch combinations that were not covered by the tables of standard thread dimensions. Flank angle deviations were tabulated for basic depths of .75H thread flank engagement, rather than .5625H as required for UNJ threads. For these and various other reasons, the STC-TF decided to delete the Appendix from AS8879C.

2.11.16 (Continued):

It should be noted that not all indicating thread gages detect the full extent of all form deviations. For example, 3 element indicating gages do not detect the full extent of an oval out-of- round condition, and 2 element indicating gages do not detect the full extent of a tri-lobular out-of-round condition.. This topic is addressed in ASME B1.3M-1992 paragraph 4d, and will be further explored by the STC-TF.

2.12 New Provisions:

In 1993, the AIA published a white paper on threads to justify revision of MIL-S-8879C. That paper made a strong point that net fit threads would not assemble. That paper concluded that the thread design specified by MIL-S-8879C should include a small clearance between mating threads at MMC. At the industry thread meeting in Philadelphia, on March 16, 1993, the DOD agreed to work with the aerospace industry to revise MIL-S-8879C to incorporate changes that the AIA considered necessary. Shortly thereafter, the DOD inactivated MIL-S-8879C, and so the needed changes were never addressed. The STC-TF team determined that providing a clearance in the external thread was considered a poor choice, as this would likely bring to question impact on existing bolt tension allowable loads. Providing a clearance fit in the internal thread was considered a better choice, but was not adopted because the STC-TF could not verify that all existing designs could accommodate a change to the internal thread dimensions.

The Unified threads defined in the National Bureau of Standards Handbook H28 include 3 different classes of thread fit. The class 1 and 2 external threads provide an allowance at the pitch diameter and each has a wider tolerance band than class 3. The class 1 and 2 internal threads simply provide a wider tolerance band than class 3. Logically, for typical tension fastener applications where the internal and external threads are made from similar materials and heat treated to similar levels, the allowance should have been applied to the stronger thread, i.e., the internal thread, not the weakest, i.e., the external thread. Whatever the rationale was at the time, that an external thread allowance was incorporated and an internal allowance was not, seems to be lost to history. Figure 6 illustrates the different thread fit classes. The class 3 threaded ring and plug gage limits are also shown in Figure 6.

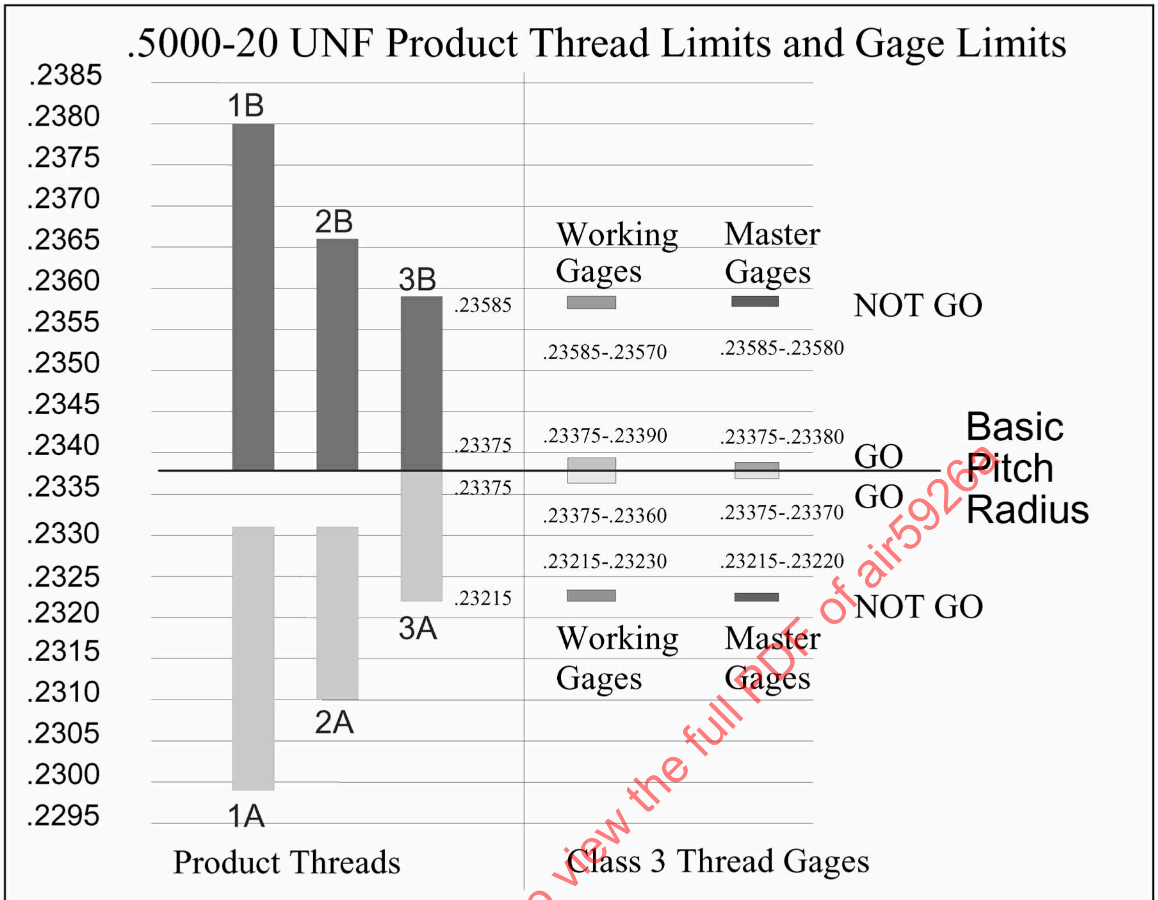

FIGURE 6 - Unified Thread Class Tolerance Bands and Thread Gage Tolerance Bands

2.12 (Continued):

MIL-S-8879A did provide class 3BG threads that provided an internal thread allowance as a constant clearance dependent on threads per inch (e.g., 0.003 inch for all pitches 32 threads per inch and coarser) at the pitch diameter for all thread sizes. MIL-S-008879B and MIL-S-8879C eliminated that option stating that thread “class 3BG for high temperature applications shall no longer be used”. As there has been no objection voiced concerning the elimination of class 3BG, the STC-TF team saw no reason to incorporate class 3BG in AS8879. The STC-TF team did consider adding provisions for a clearance fit to AS8879. Three possible methods for providing clearance have been identified.

Method 1, typified by MIL-S-8879A, class 3BG, is to simply add a constant small clearance to each thread size. The impact on thread dimensions for small size and large size extremes are not a consideration with this method. Small thread sizes lose a higher percentage of material in their thread form than large thread sizes, and mechanical properties of the smallest size will be affected the most.

2.12 (Continued):

Method 2 uses calculated length factors as described in FED-STD-H28/6, Revision None, beginning with paragraph 5.5.4. Length factors are calculated using standard GO gage thread lengths and actual lengths of engagement of mating threads upon which their dimensional limits are based. (MIL-S-8879C states, “The length of engagement for UNJC, UNJF, and 8UNJ series threads, upon which their specified tolerances are based, is equal to the basic major diameter”. The UNJEF, 12UNJ, and 16UNJ thread series in MIL-S-8879C use a length of engagement of 9 pitches.) This method has merit in that it considers thread diameter and length of engagement of mating parts and is strongly related to inspection gage definition. Results are generally proportional to the thread size and provide more clearance for larger sizes than for the smaller sizes. One problem with this method is that gage length is not precisely known.

Method 3 calculates allowance using the pitch diameter tolerance formula and applies clearance to the internal thread in a manner similar to that used for external thread class 2A. The application of pitch diameter tolerance to internal threads has historically provided 30% greater tolerance, than for external threads, due to the difficulty encountered in manufacturing internal threads. No clear reason for increasing the allowance in internal threads, as compared to external threads, to allow 30% more clearance presented itself. When pitch diameter tolerance formulas are used to calculate an allowance for internal threads the results are proportional to the diameter, length of engagement, and thread pitch. The pitch diameter formula may be found in FED-STD-H28/2, Revision None, paragraph 6. Application of the pitch diameter tolerance in determination of an external thread allowance is described in FED-STD-H28/2, Revision None, paragraph 5.

It should be kept in mind that there is no truly wrong way to compute an allowance for threads. It is possible to provide more theoretical clearance in the thread form than is necessary to ensure assembly of product threads. Excess clearance will affect mechanical properties. Conducting qualification tests on a new thread class should be considered prior to use. In any case, if current product internal threads are satisfactory, they will continue to be satisfactory. If new production internal threads will meet specified requirements, with a proposed clearance, then the product thread will be suitable.

A side benefit of providing a small clearance between the external thread and the internal thread would be seen in thread GO gages. Current practice of manufacturing GO gages inside product tolerance at maximum material condition limit would lead one to believe that the master threaded GO set plug should screw into the master threaded GO set ring. The maximum material condition of these gages is net fit. In reality, it is highly unlikely that they will assemble. Master pins and rings may be purchased as matched sets, and these GO gage sets will assemble if that is a requirement imposed as part of the purchase order. Master GO gages manufactured by different companies do not generally assemble. The implication is that threaded product manufactured at maximum material limit will not assemble, or that one of the GO gages does not conform to dimensional requirements.

2.12 (Continued):

One observation that was made concerned the relationship between GO and NOT GO gages and product thread manufacturing limits, and may be of some value. If the thread manufacturer subtracts gage size from product size manufacturing limits, then the loss of thread tolerance due to the gage tolerance may be considered as contributing a "guard band" to help reduce the likelihood that product threads could be out of tolerance. Using the .5000-20 UNJF thread size as an example, the product external thread pitch diameter limits are 0.4643 to 0.4675 inches.

Subtracting the working gage limits leaves manufacturing limits that would be 0.4646 to 0.4672 inches. This is a tolerance of 0.0026 inch, or a loss of 0.0003 inch from both minimum and maximum limits. Using this in practice would result in reduced product variation, and less chance of producing nonconforming product. Where indicating gages are used and gage uncertainty is known, subtracting gage uncertainty from product thread limits would also provide a "guard band". Refer to Figure 6 where diameter values were divided by 2 to show impact around the basic pitch radius used as a zero line.

2.13 Rationale for Assessing Strength Impact on Internal Thread:

Thread mechanical properties when loaded in tension were a concern. Initial evaluation of thread tensile strength was conducted using the H28/ASME formulas. Three modes of thread failure are recognized. The three modes are: (1) tensile failure of the external thread cross section, (2) external thread stripping, and (3) internal thread stripping. Other modes of failure, such as nuts that split from hoop stresses, that are due to overall part design were not considered. It is an important distinction to keep in mind that thread form design limits are a separate issue from threaded product design properties. Figure 7 illustrates the location of the section areas used to calculate thread strengths. The computed shear areas were multiplied by a shear strength value of 95,000 psi to determine the load carrying ability of each standard fine thread size. Tensile strength was determined using a 160,000 psi tensile requirement.

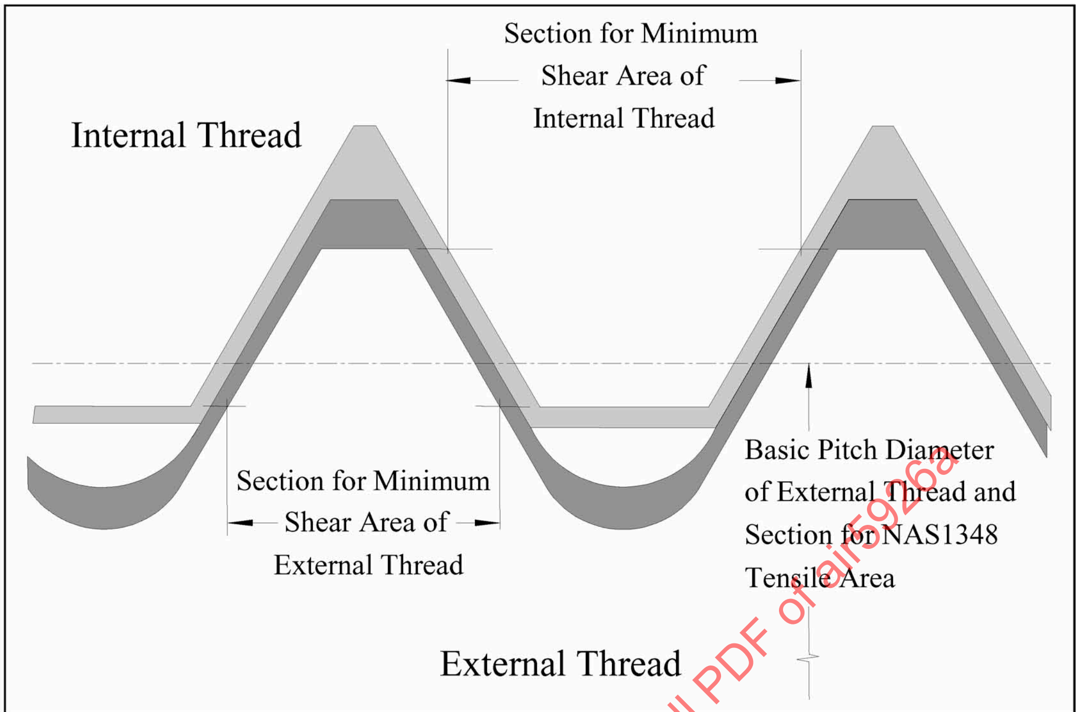

FIGURE 7 - Thread Section Areas

2.13 (Continued):

Impact on thread tensile strength and thread stripping strength was evaluated by incremental changes in calculation dimensions. It was recognized that external thread tensile area would always provide the lowest failure load where existing tension nuts provide a one-diameter length of engagement. It is also recognized that various tensile stress areas have been used over the years. External thread tensile area is empirical, i.e., based on test results. The largest tensile area that is in common use is computed at basic pitch diameter of the external thread. All other tensile areas are smaller, and resulting differences between computed tensile strength and thread shear strength would be larger for those areas. To state this in another manner, the calculated tensile strength at the basic pitch diameter is as close as calculated tensile strength can get to calculated thread shear strength for either internal or external threads. The impact of oversize nut threads on failure loads is shown in the example represented by Figure 8. In Figure 8 the circles indicate required bolt tensile strength at basic pitch diameter regardless of nut thread dimensions. The squares represent reduction in bolt thread shear strength as nut minor diameter increases beyond specified maximum, which is labeled as “design size”. The diamonds indicate the reduction of nut thread shear strength as nut pitch diameter increases beyond specified maximum. This figure illustrates that fastened threads are much less sensitive to oversize nut thread pitch diameter than to nut thread minor diameter.