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FATIGUE ACCEPTANCE TEST LIMIT CRITERION FOR LARGER DIAMETER
ROLLED THREAD FASTENERS

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REFERENCE: Kephart, A.R., 1), "Fatigue Acceptance Test Limit Criterion for Larger Diameter Rolled Thread Fasteners."

ABSTRACT: This document describes a fatigue lifetime acceptance test criterion by which studs having rolled threads, larger than 1.0 inches in diameter, can be assured to meet minimum quality attributes associated with a controlled rolling process.

This criterion is derived from a stress dependent, room temperature air fatigue database for test studs having 0.625 inch diameter threads of Alloys X-750 HTH and direct aged 625. Anticipated fatigue lives of larger threads are based on thread root elastic stress concentration factors which increase with increasing thread diameters. Over the thread size range of interest, a 30% increase in notch stress is equivalent to a factor of five (5X) reduction in fatigue life. The resulting diameter dependent fatigue acceptance criterion is normalized to the aerospace rolled thread acceptance standards for a 1.0 inch diameter, 0.125 inch pitch, Unified National thread with a controlled Root radius (UNR). Testing was conducted at a stress of 50% of the minimum specified material ultimate strength, 80 Ksi, and at a stress ratio (R) of 0.10. Limited test data for fastener diameters of 1.00 to 2.25 inches are compared to the acceptance criterion.

Sensitivity of fatigue life of threads to test nut geometry variables was also shown to be dependent on notch stress conditions. Bearing surface concavity of the compression nuts and thread flank contact mismatch conditions can significantly affect the fastener fatigue life. Without improved controls these conditions could potentially provide misleading acceptance data. Alternate test nut geometry features are described and implemented in the rolled thread stud specification, MIL-DTL-24789(SH), to mitigate the potential effects on fatigue acceptance data.

KEY WORDS: fasteners, fatigue acceptance tests, rolled threads, thread size effects, Alloy X-750 HTH, Alloy 625 (direct aged), Test Nut effects, fatigue test procedures, MIL-STD-1312-11A, MIL-DTL-24789(SH).

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The rolled thread process has been shown to be an effective means of improving the Stress Corrosion Cracking (SCC) initiation resistance of threaded fasteners, References (1), (2) and (3). Rolled threads have an extensively documented improved fatigue resistance over cut threads, particularly for aerospace, petrochemical and transportation industry applications. As a means of supporting improved SCC resistance, fatigue acceptance testing of rolled thread studs is performed to demonstrate the presence of beneficial residual compressive stresses in the rolled thread roots.

Initially the specification for rolled thread studs contained process related quality assurance requirements consistent with the 40 year aerospace rolled thread experience, typified by aerospace fastener specifications for IN-718 bolts, References (4) and (5). Test procedures complied with MIL-STD-1312, Test 11, tension fatigue, Reference (6). Use of the aerospace acceptance practice and AMS/SAE experience would assure that the inherent benefits of the thread rolling process to fastener properties would be linked to commercial process quality controls.

A significant finding of the fatigue acceptance testing of in-feed rolled threads is the occurrence of fatigue fracture in large diameter threads (2.0 to 2.25 inch basic thread diameter) at cycles well below the aerospace minimum acceptance criteria of 65,000 cycles, average. Also, large diameter cut threads exhibited reduction in fatigue life compared to smaller diameter threads.

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Review of these events led to the conclusion that large diameter threads with more severe notch stresses, when tested at the same net-section stress (50 % of Ultimate Tensile Strength, UTS), were responsible for the reduced lifetime, rather than a reduced fatigue benefit due to the thread rolling process. This contention is consistent with fundamental notched fatigue behavior in which the principal driving force of fatigue life is the local notch root stress, and not the net-section stress which is independent of notch sharpness.

To assure consistency of thread root applied notch stress effects, two approaches were considered. In one, the nominal (net-section) test stress was decreased to account for the higher notch concentration factor present in larger diameter threads. This would sustain the same thread root stress and provide a controlled measure of thread rolling process consistency. The other approach was to sustain the same nominal stress, as in smaller aerospace fasteners, but reduce the fatigue acceptance lifetimes for larger diameter high notch stress threads.

Accordingly, MIL-DTL-24789(SH), Reference (7), contains an allowance for reduced fatigue life when the thread diameter exceeds 1.00 inches. This paper describes the reference procedure for determining the reduced fatigue cycle lifetime for large diameter threads. The procedure is based on a combination of fundamental notch fatigue behavior and stress dependent test data from laboratory scale in-feed rolled threads of alloys X-750 HTH and direct aged 625 (DH).

AEROSPACE ROLLED THREAD FATIGUE ACCEPTANCE TESTING

Aerospace fasteners and tension fatigue testing include, depending on the specific fastener alloy, the following requirements or features:

- 1) test procedures that comply with MIL-STD-1312, Method 11A (tension fatigue) which describes the compression nuts and facility/ fixture requirements for thread testing;
- 2) a ratio of minimum to maximum tension test stress (R ratio) of 0.10;
- 3) a test load that is normalized to thread section area providing the same net-section test stress for all sized fasteners, independent of thread diameter or thread root radius;
- 4) tests are conducted in the load controlled mode;
- 5) nearly all fastener alloys must meet a minimum fatigue lifetime of 65,000 cycles based on the average of all fatigue tests;
- 6) nearly all fastener alloys must meet a minimum fatigue lifetime of 45,000 cycles based on the minimum individual fatigue test result;
- 7) tests are terminated when fatigue exposure exceed 130,000 cycles without fracture;
- 8) the test result is invalid if the threads fail in shear;
- 9) a typical test stress is 50% of the minimum allowed ultimate tensile strength (UTS) of the fastener alloy; however, some special high strength alloys allow up to 60% UTS test stress for extreme duty service and some other alloys allow lower percentages of UTS because of their limited ability to meet the minimum fatigue cycle lifetime of 65,000 cycles when tested at 50% UTS;
- 10) use of fine pitch UNJ thread form with a larger root radius than UNR for most applications;
- 11) and most aerospace fasteners are bolts having thread diameters of about 0.5 inches or less with infrequent maximum diameters of 1.5 inches.

EVIDENCE OF FASTENER SIZE EFFECTS ON FATIGUE LIFE

In addition to the observations of apparent thread size effects on fatigue life from experience with rolled and machined (sharp tool cut) fasteners, summarized in Table 1, size effects have also been reported from aerospace fastener experience, Reference (8). This test data from SPS Technologies

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is displayed in Figure 1 for high strength low alloy steel fasteners of sizes ranging from 0.25 to 5.0 inches in diameter.

This report shows that large diameter threads have much lower fatigue cycle lifetimes. Interpretation of the cause of the findings by the author of Reference (8) focuses on microstructure and more likely weak-link materials inclusion related degradations (Weibull sources) for larger diameter fasteners, without reference to notch effects on thread root stresses.

Review of fastener "size" effect findings and theories shows inconsistent views. A more complete list of the possible fatigue life influencing factors is provided below for controlled test stress conditions; i.e., same normalized test load (net-section stress) based on test load divided by the thread section area, and fatigue test data determined by thread section fatigue fracture.

- 1) Notch sharpness or bluntness: normally expressed as a stress or strain concentration factor and typically elastic for simplicity.
- 2) Thread fabrication method: surface effects due to altered material properties or manufacturing induced residual stresses or surface roughness when residual stresses are controlled to similar levels.
- 3) Fatigue crack growth path: increases with large size; therefore, implying longer life for larger threads in contrast to the observed shorter life.
- 4) Statistical effects of material inclusions: can act as local stress risers to reduce lifetimes in larger threads with more critical surface exposed, and more likely inclusion content or influence.

Referring to the more fundamental structural analysis work of Peterson and Neuber, it is clear that the principal factor affecting the survival of a notched and tensile stressed component is the influence of notch geometry on local notch strain or stress. Since both the thread root radius and diameter of constant pitch threads mutually affect the fundamental notch stress effects, they cannot be treated independently. Therefore, fatigue acceptance criteria for rolled threads should address this notch stress factor.

Using elastic finite element notch stress analysis consistent with notch size scaling concepts, the SPS data of Figure 1 was reevaluated to determine if the reported size effect would remain after appropriate notch factors were included. The SPS reported thread section stress was taken at a fatigue fracture lifetime of 100,000 cycles which is similar to the aerospace acceptance limit. Figure 2 shows that incorporation of notch stress effects, using the simple "linear elastic" rule, would completely compensate for the apparent thread size effect since there is no remaining notch stress trend for either cut or rolled threads over the size range reported.

NUMERICAL REPRESENTATION OF THREAD NOTCH STRESSES

Finite element analysis provided an estimate for notch stress effects which increase with diameter when the thread pitch and notch radius are held constant. The concept of scaling provides a simple visualization of this size effect. For example, for 8 thread per inch (0.125" pitch), the elastic stress concentration factor (SCF) for the first nut engaged 1.375-8 UNR-2A thread with a 15 mil root radius is 7.0; whereas, doubling the diameter to 2.75 inches results in a SCF of 8.5 for the same thread pitch and root radius. However, if the 1.375 diameter thread were scaled up by a factor of 2X, the pitch and thread root radius would also increase by 2X to 0.25 and 30 mils, respectively. The resulting SCF for this 2.75-4UNR thread would remain the same (7.0) as the 1.375-8 UNR thread.

SCF calculated values represent the average elastic stress concentration in the first compression nut engaged thread. Here the SCF is largest since succeeding threads have smaller SCF's due to decreasing load sharing distribution, Reference (9). While these notch stress factors are based on elastic modeling, the resulting effects are also related to notch strain which is considered to be the principal influence on fatigue life.

It is also noted that the coarse thread series with variable pitch has a smaller sensitivity of SCF to increasing size; thus, larger thread diameters in the coarse series would be expected to reveal longer fatigue lives. Likewise, use of the UNJ larger thread root radius series (25% larger than the UNR series) would have a slightly lower notch stress (8%) with a resulting longer fatigue (about 2X) life for the same thread pitch.

EXPERIENCE WITH FATIGUE TESTS OF LARGE DIAMETER THREADS

All of the large thread fatigue test results have been obtained from Alloy X-750 HTH or 625 DH fasteners. Figure 3 shows the decreasing fatigue life trend with increasing thread diameter for both cut and rolled (in-feed) thread fasteners tested at maximum thread section stresses of about 50% UTS, or 80 Ksi. Table 1 lists the existing large thread data and associated fastener features.

While it can be conjectured that the reduced life of the large diameter rolled threads is due to degraded rolling process and controls, the similar trend from controlled machining process (sharp tool cut) threads implies that the rolling process quality is not the principal cause of the reduced fatigue life.

REDUCED LIFE ACCEPTANCE FATIGUE TEST CRITERIA

Since nearly all of the large diameter thread acceptance tests were performed only at the required test stress of about 50% of the UTS of the fastener material, direct measurement of a stress effect on fatigue life was not available for large threads. Thus, data from the smaller diameter laboratory stud threads (0.625-11 UNR-2A) were used to determine the local thread root stress effect. Table 2 and Figure 4 list the fatigue fracture data and illustrate stress dependency trends, respectively. Figure 4 shows that, for the stress ratio R of 0.10, the data trend is linear over the range of 40 to 70% of the UTS. Thus, a simple linear numerical representation is facilitated. Statistical analysis shows excellent reproducibility (small data variability) where the lower one standard deviation occurs at 75% of the mean data. This small data scatter is well within the often quoted factor of two (2X) data range, or a lower life of 50% of the mean data.

Figure 4 displays the data in the standard S-N form (logarithm of Stress amplitude (Sa) vs. Number of cycles to fracture). However, to address thread root notch stresses, the Sa is reflected in terms of surface elastic stress (SES) by multiplying the section stress by the elastic stress concentration factor of 5.7 for the 0.625-11 UNR-2A thread with a root radius of 11.5 mils. This is consistent with the linear notch stress rule. Statistical analysis of the data of Table 2 and Figure 4 showed no significant difference between the two fastener alloys for either cut or rolled threads.

The log-linear stress amplitude vs. fatigue fracture life trend for the same stress ratio (R = 0.10) is of the form:

$$Sa (SES) = A (N, \text{ cycle life})^{-n}$$

Where A, n are fitting constants

Eq. 1

Statistical analysis of this data resulted in fitting constant values (A / n) of 1395 / -0.1606 for the average life and 1293 / -0.1606 for the 95% lower bound. The predicted average and 95% lower bound lives were 148,313 and 92,492, respectively, for a surface elastic stress amplitude of 206 Ksi.

Based on this stress dependency and increased thread root stress for larger diameter threads, the equivalent fatigue lifetimes for larger diameter threads can be estimated. Table 3 lists the features of the larger threads and expected reduced fatigue life, assuming that the "size" is dominated by thread root stress determined by root radius and thread diameter based elastic notch stress concentration factors.

Thread sizes of Table 3 cover the range of testing experience obtained from large (2.0 inch diameter) studs in addition to the laboratory test studs of 0.625 inch diameter. These estimated lives for large diameter rolled threads are derived directly from the testing of nearly packed in-feed rolled 0.625 inch diameter threads.

Figure 4 also shows the higher stress amplitudes calculated for the larger diameter threads, all of which are based on thread root surface elastic stresses listed in Table 3.

Using the 95% lower bound life of 58,058 for the 1.00 inch thread normalized to the aerospace acceptance of 65,000 cycles, the thread size dependent fatigue life limits are as shown in Figure 5 for 8-UNR rolled threads. Note that the size-lifetime trends are different for the constant pitch series (8 threads per inch) and the variable pitch (coarse) thread series because of the difference in thread root radii. Also, acceptance limits for the 8-UNJ threads would be larger (100,000 cycles for the 1.0 inch threads), because of the larger root radii (1.25 times that of 8UNR) and lower notch root surface elastic stress.

The minimum individual fatigue test acceptance life for the 1.00-8 UNR thread of 40,730 cycles is normalized to the standard aerospace lifetime of 45,000 cycles. From the log-normal distribution statistics for the 0.625 inch diameter thread database, the 40,730 cycle life is equivalent to the 99.8% lower bound, or about 3 standard deviations (SD) from the mean. The 95% lower bound is equivalent to 1.645 SD. Thus, the aerospace minimum individual test life limit can be treated as a very unlikely occurrence (0.2%) based on the reference 0.625 inch diameter database variability.

The rolled thread size dependent fatigue life relationships, shown in Figures 5, can be numerically expressed in the form of:

$$N \text{ (Number of cycles)} = B \times (\text{Basic Thread Diameter, BTD})^m \quad \text{Eq. 2}$$

Where B, m are fitting constants

The fitted constants for the minimum acceptable fatigue lifetimes of 8-UNR and 8-UNJ thread forms of diameters larger than 1.00 are represented by (B / m) of 65000 / -2.21 and 100000 / -2.06, respectively.

Figure 5 shows the available fatigue fracture test data for each diameter thread, all of which fall to the acceptance side of the 95% lower bound curve. Thus, all large diameter threads made by controlled in-feed rolling that have been tested with fatigue lives less than 65,000, would be acceptable to these limits. Using the formulation of eq. 2, the minimum average fatigue life acceptance limits for 2.0 inch diameter threads would be 14,000 and 24,000 cycles, for 8-UNR and 8-UNJ, respectively.

Compared with the fatigue life of machined (cut) threads, the rolled thread studs 95% lower bound minimum acceptance limit has a minimum fatigue life improvement of about 5X for 1.0 inch diameter threads, and about 2.5X for the 2.0 inch diameters. For a typical fatigue life, rolled thread studs are expected to have a 10X to 5X improvement, respectively, for the same thread diameters. This improvement factor over cut threads is due to the presence of residual compressive stresses in the thread roots of rolled threads. Rolled threads from small and large diameter threaded studs, with reduced fatigue acceptance lives, have exhibited significantly improved resistance over cut threads when exposed to aggressive aqueous environments prone to stress corrosion cracking, Reference (3).

Limited fatigue testing of studs having both small and large diameter thru-feed rolled threads also show equivalent or better lifetimes than in-feed threads.

RESULTS FROM FATIGUE TESTS OF NUT GEOMETRY VARIABLES

Undocumented commercial experience from fatigue tests employing various compression nut geometries has noted significant effects on threaded fastener fatigue test life. This led to limited tests to evaluate the principal cause of fastener fatigue life sensitivity to test nut geometry. Again, using the principle of local thread root stress concentration, two primary variables were investigated:

a) Nut bearing surface flatness, ranging from flat to the maximum concavity allowed by the nut dimension tolerance required by MIL-STD-1312, Figure 2 and Table II of test 11A, and

b) Nut thread flank angle mismatch between external and internal threads.

A simple cylindrical external nut shape was compared to the flanged type shape allowed by MIL-STD-1312, 11A, in addition to a range of nut diameters ranging from 1.3 to 2.0 times the basic thread diameter (BTD). For evaluation of these nut geometry variables, the 0.625-11UNR-2A thread form was tested using Alloy X-750 HTH for both the rolled thread test studs and the compression nuts.

Because of the smaller SCF of 5.7 for the 0.625 inch diameter threads and the longer fracture fatigue lifetime of about 200,000 cycles, when tested at 50% of UTS, the testing was conducted at 60% of UTS (95 Ksi, maximum stress) at a stress ratio of $R=0.10$. Baseline fatigue life data for flat bearing surfaced nuts of 1.6 X BTD and 2.0 X BTD exhibited no difference in trends between nut diameters and averaged 66,000 cycles. The smaller diameter nut (1.3 X BTD) had a shorter fracture fatigue life, averaging 40,000 cycles from two test studs. Flanged nut geometry testing, comparing a flat bearing surface to a concave surface, showed a factor of 2 to 3 longer fatigue cycle life for the concave nuts. The flat nuts had a bearing surface dimensional runout of less than 0.001 inches, whereas, the concave surfaced nuts had a runout of 0.003 to 0.004 inches, near the maximum allowed concavity of 0.004 inch per MIL-STD-1312, 11A, for the 0.625 inch diameter thread test nut.

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Figure 6 shows the fretting surface contact footprints for bearing surfaces including the observed fatigue lives. In one test of studs and mixed geometry nuts (one flat and one concave), fracture occurred in the flat nut threads. Figure 6 shows nut-to-tensioner cup contact only at the outer nut diameter, whereas, the flat surface nut had a relatively uniform contact footprint. Application of the test load at a location further from the stud thread root with the concave nuts is believed to lower the local stress concentration in the first nut engaged screw thread, thus increasing the fatigue lifetime.

Figure 7 shows the data trends from concave and flat bearing surface flanged nuts in addition to those with mating flanks which had contact both above and below the screw thread pitchline. When the nut contact is located above the pitchline flank cracks can occur which, if sufficiently deep, can relieve the first engaged thread root stress; thereby, extending the fatigue life. Both thread root cracking and fracture occurred in the set with nut flank contact above the pitchline. The average fatigue lifetime observed was similar to the set with nut flank contact below the pitchline. Thus, from this test, the most sensitive nut geometry feature affecting fatigue life is the bearing surface concavity.

While thread flank contact mismatch showed no significant effect in this test, other tests have shown significant increases in fatigue life due to contact above the pitchline with resulting large flank cracking without any thread root cracking and thread fracture. Figure 8 shows flank contact mismatch induced fatigue cracking in which fatigue runout (no thread fracture) resulted after 260,000 cycles of testing at 50% UTS for Alloy X-750 HTH. Similar testing with nuts of matched flank contact produced fatigue fracture lives ranging from 31,000 to 115,000 cycles. To avoid inaccurate fatigue lifetime measurements, thread flank contact above the thread pitchline must be controlled.

When fatigue acceptance or process qualification tests are required by MIL-DTL-24789, the required test nut geometry is different than MIL-STD-1312, 11A. MIL-DTL-24789 requires tighter tolerances on bearing surface runout of approximately 25% of the aerospace allowances on nut concavity. Also, a special feature has been added to the internal thread to avoid external thread contact above the thread pitchline. This feature (controlled flank contact, CFC) is a metal cutout of about 0.002 inch deep at the major diameter of the internal thread and extending over the upper one-third of the thread flank. Figure 9 illustrates the nut geometry features which help avoid potential fatigue lifetime effects from nut geometry. The simple cylindrical geometry and double ended test capacity of these test nuts offset the costs of the additional features, while providing more reliable test data.

CONCLUSIONS

1. Reduced fatigue lives in large diameter rolled threads, relative to existing aerospace standards for smaller diameter threads, can be explained by accounting for thread root notch effects, a principal factor not present in current aerospace fastener fatigue acceptance requirements.
2. Combining a fatigue database from a small diameter stud with thread diameter and pitch dependent elastic stress concentration factors can provide a practical formulation for thread "size" dependent fatigue acceptance limits.
3. Fatigue lifetime limits, which include thread diameter dependent stress concentration factors for 0.125 inch pitch threads, compare favorably to in-feed rolled thread test database for up to 2.25 inches in diameter.

4. Significant effects of test nut geometry and tolerances on fatigue fracture life of threads prompted adoption of additional control features in the rolled thread procurement specification MIL-DTL-24789.

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TABLE 1.

FATIGUE FRACTURE LIVES FOR CUT AND IN-FEED ROLLED THREADS of LARGE DIAMETER STUDS
and SMALLER BASELINE THREADS OF ALLOYS X-750 AND AGED 625 AND TESTED AT 75°F AND R = 0.1

Thread Size and Form	Alloy	Manuf. Process	Stress, Max, Ksi	Root Radius mils	Stress Conc. Factor	Stress Amplitude SES, Ksi	Fatigue Fracture Life, cycles	Fatigue Runouts* No. / Cycles
1.000-8UNR-2A	X-750 HTH	Cut	82.4	15	6.3	234	13,396; 14,638; 15,073; 14,730	0
1.375-8UNR-2A	X-750 HTH	Cut	82.6	14.5	7.2	268	13,293; 18,800; 16,800; 13,728; 15,323; 13,109	0
2.000-8UNJ-2A	X-750 HTH	Cut	80.1	21	7.35	265	9,349; 8,490	0
2.000-8UNJ-2A	625 DH	Cut	80.1	21	7.35	265	5,276; 5,804	0
2.250-8UNJ-2A	625 DH	Cut	80.1	21	7.50	273	7,534	0
0.625-11UNR-2A	X-750 HTH	Roll-REC	76.5	11.5	5.7	196	114,000; 140,000; 145,000; 339,000 (Test by SPS)	1 / 260,000
0.625-11UNR-2A	X-750 HTH	Roll-NP	83	11.5	5.7	214	192,475	1 / 200,000
0.625-11UNR-2A	X-750 HTH	Roll-NP	95	11.5	5.7	245	51,662; 54,308; 54,616; 67,878; 70,504; 82,212; 89,925	0
0.625-11UNR-2A	625 DH	Roll-UF	90	11.5	5.7	231	109,211; 111,331; 116,922; 132,685; 173,262; 175,944	2 / 260,000
1.000-8UNR-2A	X-750 HTH	Roll-UF	82.4	15	6.3	234	89,187; 84,215	13 / 200,000
1.375-8UNR-2A	X-750 HTH	Roll-UF	82.5	16	7.0	260	65,247; 67,838; 75,385; 85,045; 94,379; 99,857 100,719; 115,304; 116,919; 121,552; 125,657	4 / 200,000
1.375-8UNR-2A	X-750 HTH	Roll-UF	82.5	17	6.85	253	73,015; 74,946; 72,637; 69,116; 62,691; 54,543; 57,632; 70,427	0
1.375-8UNR-2A	X-750 HTH	Roll-UF	69.5	16	7.0	219	None	7 / 200,000
1.6875-8UNR-2A	X-750 HTH	Roll-NP	80.0	16	7.35	262	49,000; 50,000; 74,000; 66,000; 98,000; 102,000	0
2.000-8UNR-2A	625 DH	Roll-UF	80.6	19	7.5	272	22,116; 23,162; 40,675	0
2.000-8UNR-2A	625 DH	Roll-UF	60.4	19	7.7	204	175,055	1 / 260,000
2.000-8UNJ-3A	X-750 HTH	Roll-NP	80.3	21	7.35	264	39,281; 47,374; 39,117	0
2.000-8UNJ-3A	625 DH	Roll-NP	80.3	21	7.35	264	39,341; 38,807; 36,415; 44,412; 39,262; 40,144; 30,268; 26,042	0
2.000-8UNJ-3A	625 DH	Roll-NP	67.7	21	7.35	222	90,122	0
2.250-8UNJ-3A	625 DH	Roll-NP	81	19.5	7.60	277	23,089; 27,075; 27,161; 27,513; 28,090; 31,318	0
2.250-8UNJ-3A	625 DH	Roll-NP	66	19.5	7.60	226	89,029	0

NOTES: UF = Underfilled Thread Crests; NP = Nearly Packed Thread Crests due to slightly oversized blank diameters; Runouts are without thread section fracture

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TABLE 2. FATIGUE FRACTURE LIVES FOR CUT AND IN-FEED ROLLED THREADS (0.625-11 UNR-2A) OF ALLOYS X-750 AND AGED 625 TESTED AT 75°F AND R = 0.1

TEST LOAD Kips Max.	STRESS % of MINIMUM UTS	MAXIMUM TEST STRESS Ksi	ALTERNATING STRESS Ksi SES	Test Material X-750 HTH		Test Material Aged 625 (DH)	
				Cut Threads	Rolled Threads	Cut Threads	Rolled Threads
				Fracture Life, cycles	Fracture Life, cycles	Fracture Life, cycles	Fracture Life, cycles
34.5	98	157	406	750	1,321	1,202	2,088
29.9	85	136	352	2,370	4,796	2,360	5,186
26.4	75	120	310	3,841 3,860	11,055 9,423	3,994	13,189
22.9	65	104	269	7,524	22,455		35,318
21.1	60	96	248	7,796	35,617		
19.4	55	88	228	13,791	64,329	10,937	128,997
17.6	50	80	207	21,234	157,984	20,222 18,405	105,807 116,399
14.1	40	64	166	51,251 68,016	1,485,860 730,577	39,358	807,331
12.3	35	56	145		2,840,764		
10.6	30	48	124	165,508	6,190,642		10,213,625
8.5	24	39	100	803,527			
7.5	21	34	88			1,177,513	

NOTES: 1) For solid 0.625" diameter studs, the stress concentration factor (K_t) or SCF = 5.75 for the first nut engaged thread
This rolling process produced Nearly Packed Thread Crests due to supplier selection of slightly oversized blank diameters

TABLE 3. ESTIMATED FATIGUE LIFETIMES OF ROLLED THREAD FASTENERS BASED ON THREAD ROOT STRESS EFFECTS

Thread Diameter, Form for 2A Class Thread Fit	Average Root Radius, mils	Elastic SCF	S _a (SES) for 50% of minimum UTS	Estimated Fatigue Lifetime Cycles to Fracture		
				Mean Life	95% Low Bounds	99.8% Low Bounds
0.625 - 11 UNR	11.4	5.73	206	148,313	92,492	64,885
1.000 - 8 UNR	15.7	6.16	222	93,099	58,058	40,730
2.000 - 8 UNR	15.7	7.90	284	20,093	12,531	8,791
1.000 - 12 UNJ	13.7	6.77	243	53,082	33,087	23,208
1.000 - 8 UNJ	20.7	5.79	208	139,655	87,093	61,098
2.000 - 8 UNJ	20.7	7.34	264	31,658	19,743	13,850

NOTES: Maximum Thread Section Stress is 50% of the minimum UTS of 160 Ksi, Stress Range R = 0.10

Figure 1a.

Air Fatigue Test Results for Cut Threads of High Strength Alloy Steel as a Function of Thread Diameter

Figure 1b.

Air Fatigue Test Results for High Strength Alloy Steel Thread Rolled After Heat Treatment as a Function of Thread Diameter

Figure 2.

SPS Fatigue Strength for High Strength Alloy Steel of Both Cut and Rolled
Thread Fasteners Revised to Account for Thread Root Surface Elastic Stress

Figure 3.

**Threaded Fastener Fatigue Lifetimes of Cut and In-Feed Rolled
Threads of Alloys X-750 HTH and 625 DH
Stressed to 50% UTS As a Function of Thread Diameter**

Threaded Fastener Fatigue Lifetimes of Cut and In-Feed Rolled 0.625 Inch Diameter Threads of Alloys X-750 HTH and 625 DH Stressed from 30% to 98% of the Minimum Acceptable UTS

Figure 5.

Air Fatigue Lifetime Acceptance Limits for Rolled Threads
Stressed to 50% UTS as a Function of Thread Diameter
and Compared to Limited Cut and Rolled Thread Test Data

Figure 6.

Fatigue Test Wear Marks on Bearing Surface of Flange Design Test Nuts
As a Function of Nut Bearing Surface Flatness

NUT DIAMETER
Inches and
Relative BTD

1.245" or
2.0 X BTB
FLANGED

Flank Contact
Below Pitchline

FLAT BEARING SURFACE,
0.000 TO 0.001 FIM

NUT ID: 5-6; Stud DD1-05
Thread Fracture 51,622

MIXED BEARING SURFACE TEST,
FLAT ONE END, CONCAVE ON OTHER

NUT 5-9; Stud DD1-18
Thread Fracture 89,925

CONCAVE BEARING SURFACE,
0.003 TO 0.004" FIM

NUT 5-13; Stud DD1-17
Thread Fracture 162,668

NUT ID: FTN-2; Stud DD1-10
Thread Fracture 53,052

NUT 5-5; Stud DD1-18
Thread Did Not Fracture

NUT FTN-1; Stud DD1-20
Thread Fracture 131,960

1.245" or
2.0 X BTB
FLANGED

Flank Contact
Above Pitchline

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Figure 7. Effect of Compression Nut Bearing Surface Flatness on Threaded Fastener Fatigue Life for Alloy X-750 HTH Tension Tested at Maximum Stress of 60% UTS, $R = 0.1$ and 75°F

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Figure 8. Fatigue Cracks in Thread Flanks of Alloy X-750 HTH Rolled Threads
With Engaged Nut Flank Contact Footprints Only Above the Thread Pitchline

First Nut Engaged Thread

25 X Magnif.

No Root Cracks Observed After 260,000 Cycles
With Maximum Stress at 50% UTS (80 Ksi)

0.625 - 11 UNR-2A Threads

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Figure 9. Illustrations of Flanged Fatigue Test Nut of MIL-1312, Test 11A and Cylindrical Test Nut Design Required by MIL-DTL-24789(SH)

Flanged Fatigue Test Nut Design Allowed by MIL-STD-1312, Test 11A

Cylindrical Shaped Fatigue Test Nut Design Required by MIL-DTL-24789

Controlled Flank Contact Feature (Cutout) of Test Nut Design of MIL-DTL-24789