A WHITE PAPER

FASTENER STRENGTH ANALYSIS

REVISION TWO

NUCLEAR SAFETY CONCERN 93-11

SAN ONOFRE NUCLEAR GENERATING STATION

A WHITE PAPER
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Revision Two

NUCLEAR SAFETY CONCERN 93-11

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EXECUTIVE SUMMARY

This paper has been prepared to address specific concerns about the ability of Safety-Related fasteners to perform their function at the San Onofre Nuclear Generating Station. A recent upgrade in fastener dimensional inspection technology has revealed that previous inspection methods may have allowed minor deviations from specified tolerances for a specific fastener thread characteristic, to pass receipt inspections. A sample of fasteners from Warehouse stock was inspected in response to this information. A number of fasteners were found to pass the previous inspection methods, but were found to be out-of-tolerance with the new inspection methods. Strength calculations were performed on the items with the greatest observed deviations from tolerances. Minor strength reductions were noted. Comparison of the reductions in thread strength to ASME Code design requirements were made and it was demonstrated that the strength reductions were on the order of 3 to 6%, with the Code strength margin being on the order of 200 to 300%. Independent Laboratory analysis and testing was performed which correlated closely with these calculated results. It was further calculated that thread attributes would have to be grossly out of tolerance (approximately 15 times the greatest observed deviation) for the threads to fail at installation. Threads this grossly out-of-tolerance would have failed previous inspection methods and would fail visual examination at receipt inspection or during installation.

This evaluation clearly demonstrates the significant strength margin inherent in ASME Code fasteners. In addition to the strength margin of the individual fastener, installation configurations would require multiple fastener failures prior to joint failure. The nature of the design of fastener preload requires the maximum fastener load to occur at installation, thereby identifying fasteners which may fail at the time of installation instead of in-service. If a fastener should loosen due to loss of preload, leakage would occur before joint failure. This paper discusses the various leakage monitoring and corrective action systems in place at SONGS, including an effective Root Cause evaluation program.

In addition to the fastener strength analysis, this paper documents a review of nuclear industry experience with fasteners. No documented cases of fastener failure due to dimensional non-conformance were identified at SONGS or elsewhere in the industry. This information is consistent with NRC and EPRI findings summarized in Generic Letter 91-17, "Bolting Degradation or Failure in Nuclear Power Plants", which closed Generic Safety Issue 29 on the same subject.

This paper does not attempt to encompass the current national controversy with respect to System 21 vs System 22 dimensional acceptance of fasteners, nor is it intended to provide a statistical dimensional characterization of all fasteners in the SONGS Warehouse. Nor does this paper analyze the effect of variation of all 14 characteristics of thread form, many of which are not measurable by even SONGS current improved inspection technology, but require implementation of System 23 level inspections. What this paper does clearly demonstrate is that current industry practice provides adequate fastener strength and reliability for the safe operation of nuclear power plants. The uncertainties revealed by advanced dimensional inspection technologies play an important role for applications having relatively small safety factors (such as aerospace), but do not have a direct link to fastener failure in heavy industrial applications where strength margins are orders of magnitude higher. In conclusion, plant nuclear safety is not compromised by current industry fastener inspection practices.

Revision Two of this paper has been prepared to include the analysis of data obtained during NRC inspection of fasteners at SONGS. Also included in this revision is an evaluation of a "combined-case" which calculates the effect on fastener strength for hypothetical conditions where both internal and externally threaded fasteners would exhibit the maximum (proportionally equivalent) out-of-tolerance conditions from this study. The hypothetical nature of the combined-case analysis is required since the maximum observed out-of-tolerance conditions for internal and externally threaded fasteners in this study, occurred in items of different sizes. Combining them requires evaluating proportionally similar out-of-tolerance conditions for the same size items. The results of this combined-case evaluation revealed that when fastener ultimate tensile strength is not considered, the ultimate thread strength of the threaded connection would be reduced by 10.45% due to the combined out-of-tolerance pitch diameter conditions. Thread strength of this combined-case was shown to exceed maximum fastener preload specified for ASME Section III, Class 1 service by 240%, and Class 2 and 3 service by 337%. Therefore, it was concluded that there is no impact on the ability of the combined-case threaded fastener condition to withstand maximum applied loadings associated with service conditions at SONGS.

The NRC inspection of fasteners involved obtaining a sample of items from the SONGS Warehouse stock utilizing a classic random sampling method and the CGI Sampling Procedure. 519 items from 44 material codes were inspected using the Johnson Indicating Gage process and the Go/No-Go gages. The sample included some items which had been excluded from the White Paper sample including items of a small nominal diameter (less than 1/2") and also set screws. Analysis of the inspection data for this sample revealed the following conclusions[1]:

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[1] "Statistical Evaluation of NRC Audit Data for SONGS Fastener Stock", Tetra Engineering Group, INC. Report 94-SCE-005, Authors: Dr. Frank J. Berte', David S. Moelling PE, and Fredrick C. Anderson PE. (Attachment 5)

1) The NRC data is consistent with the White Paper data. This confirms that the sampling procedures performed for the White Paper data set were sufficiently random to allow vaiid statistical results to be drawn.

2) The limits computed from the White Paper data bound the pitch diameter deviations in the NRC Data set.

3) The NRC data shows similar behavior in bin-to-bin and supplier-to-supplier variation as the White Paper Data set. This confirms previous conclusions that there is no systematic supplier-related trends in thread deviations.

4) NRC data for smaller items not considered by the White Paper show a slightly wider range of deviation than the larger items. Otherwise, they are consistent with observations for the White Paper data.

5) Thread Strength Margins for the smaller items are adequate, even with the slightly larger deviations from the ANSI B-1.1 thread limits.

6) Thread Strength Margins for the larger items are consistent with those computed for the White Paper data. These demonstrate large margins to ASME Code service requirements.

The data and evaluations included in Revision Two of this paper support the previous conclusion that plant nuclear safety is not compromised by current industry fastener inspection practices. Additionally included in Revision 2 of this report is a recent ASME Code Interpretation which provides clarification of fastener inspection requirements. The NRC has reviewed and concurred with this Interpretation.

BACKGROUND

In early 1993, the systematic dimensional overcheck of fastener threadform during Receipt Inspection at SONGS was upgraded from the use of Fixed Limit (Go/No-Go) Gages, to the Johnson Indicating Gage process which provides a direct digital readout of thread attributes for comparison with the appropriate criteria. The Johnson Gage system was found to provide improved inspection accuracy through the measurement of specific thread characteristics, and faster inspection times when compared with the Go/No-Go gages.

Since its first issuance in 1924, the ANSI Standard for threaded fasteners, B-1.1, has provided dimensional criteria for standardized thread forms. The dimensions and tolerances have been presented to the 1/10,000 of an inch, and have remained constant over the years for the standard thread forms. Early dimensional inspection methodology was unable to assure exact fastener thread conformance to the fine tolerances stated in the standard. Modern fastener industry practice has included the usage of indicating gages for the setup and maintenance of the manufacturing machining processes, with the fixed limit gages providing final dimensional acceptance. In 1978, framework for the use of indicating gages for final dimensional acceptance of threaded fasteners was provided with the issuance of ANSI B-1.3. This standard provided definition of System 21 inspection requirements (which are met with the Go/No-Go gaging), and System 22 inspection requirements (which include the measurement of Pitch Diameter and can be met with an indicating gage system).

Licensing requirements for the final dimensional inspection of safety related fasteners dedicated for use at SONGS are tabulated in Attachment 1. Briefly stated, all safety-related pressure boundary fasteners are required to meet System 21 requirements, except for class 3 fit bolting which must meet System 22 requirements. These are requirements for final dimensional inspection by manufacturers and are reflected in the respective Procurement Engineering Packages (PEPs). SONGS has elected to exceed these requirements through the use of Johnson indicating gages to overcheck the System 22 attribute of thread pitch diameter at Receipt Inspection. The usage of the Johnson Gage system at SONGS was initiated for a twofold purpose, improved inspection productivity and the trending of supplier performance. An independent report by Nuclear Oversight on the usage of the Johnson Gaging system in October 1993, stated the following advantages over the previous system of inspection:

- Increased accuracy in measurement of thread attributes
- Faster inspection process
- Minimum gage wear
- Less frequent and easier calibration
- Direct readout of thread dimensions precludes dispute with suppliers over accuracy of Go/No-Go gages

The Johnson Gage system at use in the Commercial Grade Item Laboratory for Quality Control Receipt Inspection is connected to a computer database into which results of each inspection are entered. The computer can then process the information and produce a Supplier Quality Index for each supplier. From this information, it can be determined if a specific supplier is providing consistent bolting product, which is a good indicator that the supplier is utilizing statistical process controls. Analysis of the inspection data has allowed Procurement Engineering to recommend consistent high-quality suppliers over the less consistent suppliers.

The nuclear industry is governed by construction standards which routinely provide a margin in the range of 200 to 300% of ultimate strength in bolting materials over the maximum operating service load. Nuclear industry oversight organizations and interest groups routinely provide information on industry equipment failure and trend performance of specific categories of components, such as fasteners. Most recently, the NRC issued Generic Letter 91-17, "Bolting Degradation or Failure in Nuclear Power Plants", which closed Generic Safety Issue 29, on the same subject, based on studies conducted by EPRI, MPC and AIF which analyzed many fastener failures. In closing this issue, the NRC found no evidence to indicate that failures were directly attributable to dimensionally nonconforming fasteners. Thus, careful evaluation of cumulated nuclear operating experience has shown that no safety issues exist with current industry fastener inspection practices.

NUCLEAR SAFETY CONCERN

Return To Stock Inspections

In December of 1993, the inspection of threaded fasteners with the Johnson Gage system was expanded from an overcheck during Quality Control Receipt Inspection, to the inspection of fasteners which had previously been examined and accepted with Go/No-Go gages.

- Bolting which had been issued for plant use, but not used, was subjected to restocking inspection with the Johnson Gage equipment. Inspection revealed several fasteners with dimensionally out-of-tolerance conditions. Two Warehouse Nonconformance Reports (WNCR) were written on 12/8 and 12/13/93 to document the inspection results. One WNCR documented the failure of threaded rod to pass the Go/No-Go inspection. The potential existed that out-of-tolerance fasteners had been installed in the plant.

  NOTE: Subsequent review of work documents revealed that the subject items failing the Go/No-Go inspection had not been installed in safety-related systems, but had been used as construction and maintenance aids, and on non-safety related equipment. No plant NCRs were required to be written.

Initiation of Nuclear Safety Concern
A Nuclear Safety Concern was issued on 12/27/93 stating the potential for non-conforming fasteners to exist in the warehouse.

Specific Bin Inspections
In response to specific information from the Submitter, samples of eight warehouse stock bins were inspected with Johnson Gage instruments, revealing dimensionally out-of-tolerance fasteners in three of the bins. WNCRs were written to document the observed conditions.

Response to Nuclear Safety Concern
The identification of these out-of-tolerance conditions was not unanticipated due to the nature of the two different (fixed limit and indicating) gaging systems. It was decided that a logical response to the Nuclear Safety Concern would include obtaining some examples of fasteners which had failed the System 22 inspection, but passed the System 21 test. The items falling in this category could then be analyzed for impact on thread strength due to the out-of-tolerance thread conditions and this reduced thread strength could be compared to the ASME Code requirements for thread strength.

WAREHOUSE SAMPLE INSPECTION

In order to obtain specific engineering data to address the Nuclear Safety Concern, it was decided to obtain additional examples of items from Warehouse stock which would pass the System 21 inspection method of Go/No-Go gaging, but indicate out-of-tolerance when examined with the Johnson Gages. From the entire Warehouse population of fasteners, sample Material Codes were identified; and from these, sample fasteners were chosen for inspection.

- A Sample Plan was devised which utilized the existing Receipt Inspection sampling procedure to the greatest extent practical. A computer listing of all fastener Material Codes, including item descriptions, was obtained from responsible Warehouse personnel. Obvious items which were not subject to the concern were eliminated from the listing (washers, metal screws, set screws). To further define the test sample, bolting smaller than 1/2 inch diameter was omitted. Additionally, items which had been previously examined with Johnson Gaging were eliminated from the listing. This left 286 fastener Material Codes subject to sample inspection.

- From Table 2 of Procedure S0123-XXXII-2.5, "Sampling Program for Assessing, Estimating, and Reporting Commercial Grade Item Quality", a sample size of 32 Material Codes was chosen from Table 2. It should be noted that 32 is the maximum sample size specified in the procedure. Given that a sample of 32 Material Codes needed to be chosen, from the 286, it was decided that a systematic sampling plan would be utilized in which every ninth Material Code from the listing would be chosen and would provide an adequate assurance of randomness. The fasteners had been sorted by item name alphabetically. Thus, every ninth Material Code would provide various samples of each item type. Systematic sampling plans for this type application, have been shown to be essentially equivalent to simple random sampling[2]. If a Material Code item proved uninspectable due to unavailability of Johnson Gage segments, the next Material Code, of the 286, would be chosen.

- The bins associated with the 32 Material Codes were provided in their entirety to Receiving Quality Control Inspectors. Fasteners were obtained from each Material Code in accordance with procedure SO123-XXXII-2.5, which specified a sample size, based on the total number of fasteners in each Material Code, and assured randomness in the sample selection. Samples were then examined with the Johnson Gages.

- Sample inspections were performed in accordance with routine requirements for fastener Receipt Inspection. The Johnson Gage thread attribute of Functional Diameter, and the ANSI B-1.1 characteristic of thread Pitch Diameter were measured for each sample fastener. Thread Functional Diameter is a measurement specific to the Johnson Gage system and provides a measure to verify that thread conditions are between the maximum and minimum material condition requirements. Thread Pitch Diameter is measured by point contact on essentially one thread at a time. Thread Pitch Diameter is the key element in the calculation of thread shear area, which is discussed later in this paper, and relates directly to the shear strength of threads. The measurement of Pitch Diameter is consistent with the requirements of System 22 as specified in ANSI B-1.3.

- Sample Data was obtained and evaluated. The 32 sample Material Codes contained 1542 fasteners, of which 356 were inspected using the Johnson Gaging system. A total of 96 fasteners were found to be out-of-tolerance when compared to the requirements of ANSI B-1.1. This represents a 27% rate of fasteners which indicated out-of-tolerance. The out-of-tolerance items were also checked with Go/No-Go gages. All but one of the out-of-tolerance items were found to be acceptable when checked with Go/No-Go gages. Many of the out-of-tolerance readings were only out by a few 1/10,000ths of an inch, and several were reported by the responsible inspector to be slightly out of round, with the worst deviation being the recorded value. Also noted from the review of the data were repeated readings of the same fastener which included differences which were not averaged, the largest out-of-tolerance reading was recorded and subsequently used in this analysis. The statistical analysis included as Attachment 2 provides numerical distributions of the pitch diameter data from this sample inspection.

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[2] Sampling Techniques, Third Edition, W.G. Cochran, Wiley, New York, 1977

- Independent dimensional analysis of a sample of allthread fasteners was performed for comparison with data obtained during the sample inspection. The methodology and results of this testing are documented in Safety Engineering Failure Analysis Report FAR-94-005. The independent dimensional data, obtained using the Three Wire method and supermicrometer, was compared to the readings obtained using the Johnson Gages at the SONGS CGI laboratory. The majority of the readings showed differences between the Johnson Gage and Three Wire Method to vary between .1 and .5 mil (mil = 1/1,000") with the two greatest deviations being 1 and 1.5 mil. The differences in the readings could be attributed to the nature of the inspection techniques, which measure essentially one portion of one thread, and physical differences between individual threads.

The results of the Warehouse Sample Inspection revealed that over 99% of the inspected fasteners passed the System 21 requirements. This level of acceptance is the best that can be expected given the statistical nature of receipt inspection techniques. The fact that only one item failed inspection, of all inspected, validates previous receipt inspection processes as effective in meeting the previous inspection requirements.

It should be noted that a subsequent evaluation by an independent statistical analysis firm has shown the systematic sample taken in this evaluation to be essentially equivalent to a simple random sample. The data analyzed in this White Paper was shown to be conservative when compared with the 95% / 95% confidence/probability bounds of a much larger population of thread dimensional values compiled during receipt inspection over the period of 7-93 to 7-94[3].

[3] Statistical Evaluation of SCE Fastener Strength Analysis White Paper Data Base, Tetra Engineering Group, Dr. Frank Berte', Dr. Peter S. Jackson, David S. Moelling PE, August 5, 1994 (Attachment 1)

FASTENER STRENGTH ANALYSIS

The variations in Pitch Diameter will be analyzed for their impact on thread strength, and this will be compared to the ASME Code margins inherent in threaded fastener design.

- The internally threaded fasteners (nuts) and externally threaded fasteners bolts/studs/allthread) from the above Warehouse sample with the greatest deviations from the pitch diameter tolerances specified in ANSI B-1.1, were designated as Worst Case examples for the purposes of this evaluation. This does not guarantee that these items have greater deviations than any fastener which may exist in the Warehouse, due to the statistical nature of sample inspections. These Worst Case fasteners have been shown to contain dimensionally out-of-tolerance conditions which are rejectable by System 22 inspection, and will be analyzed in this report for the purpose of responding to the specific Nuclear Safety Concern.

- These Worst Case conditions were analyzed for the amount of reduction in thread strength which would result from the measured out-of-tolerance conditions. The readings for fastener Pitch Diameter were significant because Pitch Diameter relates directly to thread shear area. Fastener strength is affected by the thread shear area and the shear strength of the fastener material. Therefore, a reduction in fastener Pitch Diameter could affect the strength capacity of the threaded joint.

- While it is recognized that variation of other thread attributes, such as thread angle, taper, lead and helical deviation may also have an impact on thread strength, these attributes are not subject to current inspection methods and their effect is not evaluated in this report.

The following formulas were utilized in the development of the Fastener Strength Analysis[4]:

$$\textit{Shear Strength Of External Threads} = 0.5S_T*(A_{SS})$$

$$\textit{Shear Strength Of Internal Threads} = 0.5S_T*(A_{SN})$$

Where:

S_T = Ultimate Tensile Strength of Fastener Material

A_{SN} = Minimum Thread Shear Area for Internal Threads

A_{SS} = Minimum Thread Shear Area for External Threads

$$A_{SS} = \pi(1/P)*L_E*D1_{MAX}[\frac{1}{2(1/P)} + 0.57735\,(D2_{MIN} - D1_{MAX})]$$

$$A_{SN} = \pi(1/P)*L_E*D_{MIN}[\frac{1}{2(1/P)} + 0.57735\,(D_{MIN} - D2_{MAX})]$$

Where:

π	=	3.14159
1/P	=	Number of Threads per Inch
L_E	=	Length of Engagement
D_{MIN}	=	Minimum Major Diameter of External Thread
D_{1MAX}	=	Maximum Minor Diameter of Internal Thread
D_{2MAX}	=	Maximum Pitch Diameter of Internal Thread
D_{2MIN}	=	Minimum Pitch Diameter of External Thread

[4] - ANSI B-1.1, Unified Inch Screw Threads, 1989 Edition
- Introduction to the Design and Behavior of Bolted Joints, John H. Bickford, Marcel Dekker, Inc. 1990.
- Analysis and Design of Threaded Assemblies, E.M. Alexander, SAE, 1977

Worst Case External Threads:

Item Description: All Thread Stud, 5/8"dia-11 (UNC) by 36" length

Material Code: 305-05606 RSO#: 2063-93 Supplier: NOVA Inc.

Material Specification: ASME SA193 Grade B7 Heat Code: 8099572

Thread Functional Size Inspection was performed Satisfactorily

Thread Pitch Diameter Inspection was shown to be Out-of-Tolerance.

Pitch Diameter Reading: 0.5554 inches

Pitch Diameter Range: 0.5644 (max) to 0.5589 (min) inches.[5]

Amount Out-of-Tolerance: 0.0035 inches

Ultimate Material Strength (S_T) = 125,000 psi (min)[6]

Length of Thread Engagement (L_E) = 1 Diameter = 0.625 inches

Minimum Minor Diameter of Internal Thread = 0.5270 inches[5]

Maximum Minor Diameter of Internal Thread = ($D1_{MAX}$) = 0.5460 inches[5]

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[5] ANSI B-1.1, 1989 Edition, Table 3A, Class 2A Fit

[6] ASME Section III, 1977 Edition, Appendix I

THREAD STRENGTH CALCULATIONS FOR WORST CASE EXTERNAL THREAD CONDITION, UTILIZING THE ABOVE EQUATIONS AND DATA:

$$A_{SS} = \pi(1/P) * L_E * D1_{MAX}[\frac{1}{2(1/P)} + 0.57735\,(D2_{MIN} - D1_{MAX})]$$

CASE A: Maximum Material Conditions for both Internal and External Threads

Pitch Diameter = 0.5644" (max) $D1_{MAX}$ = 0.5270" (min)

A_{SS} = 3.14159 * 11 * .625 * .527 * [1/(2*11) + .57735(.5644 - .5270)]
= 0.7632 square inches

Thread Ultimate Shear Strength =

0.5 * 0.7632 * 125,000 = 47,700 pounds (min)

CASE B: Minimum Material Conditions for both Internal and External Threads (ANSI B-1.1)

Pitch Diameter = 0.5589" (min) $D1_{MAX}$ = 0.5460" (max)

A_{SS} = 3.14159 * 11 * .625 * .546 * [1/(2*11) + .57735(.5589 - .5460)]
= 0.6239 square inches

Thread Ultimate Shear Strength =

0.5 * 0.6239 * 125,000 = 38,992 pounds (min)

CASE C: Out-of-Tolerance Conditions from Worst Case Data

Pitch Diameter = 0.5554" (actual) $D1_{MAX}$ = 0.5460" (max)

A_{SS} = 3.14159 * 11 * .625 * .546 * [1/(2*11) + .57735(.5554 - .5460)]
= 0.6000 square inches

Thread Ultimate Shear Strength =

0.5 * 0.6000 * 125,000 = 37,507 pounds (min)

It should be noted that the Worst Case External Thread condition represents only a 3.12% reduction in thread strength from the "B" case above, which represents the industry standard calculation of thread strength based on the requirements of ANSI B-1.1. An ideal thread condition calculation is presented in case "A" above which assumes that both the internal and external threads are at the maximum material limit. Comparing "A" and "B" above provide the strength reduction over the band from the maximum tolerance to minimum tolerance numbers for both the internal and external threads, indicating a 18.26% reduction in strength across this band. The Worst Case External Thread conditions are shown to have thread strength 21.38% (i.e. 18.26% across the tolerance band plus 3.12% due to out-of-tolerance) lower than the maximum material contact thread condition from case "A".

EXTERNAL THREAD CALCULATION SUMMARY:

CONDITIONS	DIFFERENCE IN THREAD STRENGTH
Case A, Maximum Material contact both Int and Ext	
	} 18.26%
Case B, Minimum Material contact both Int and Ext	
	} 3.12%
Case C, Worst Case Out-of-Tolerance	

ASME CODE SERVICE APPLICATIONS:

The Worst Case out-of-tolerance thread conditions are shown to create reduced thread shear areas thereby reducing the shear strength of the threads. Maximum stress levels for bolting materials are specified in Section III of the ASME Code. Design stress levels are required to be lower than these maximum levels for each specified material. Shear stresses increase across the reduced shear areas when design loading is applied to a fastener with out-of-tolerance threads. The Worst Case out-of-tolerance condition must be evaluated to determine if the increased shear stresses created by the reduced thread shear area falls within the maximum allowed by the ASME Code.

For the Worst Case out-of-tolerance external thread conditions:

From ASME Section III, Appendix I, Table I-1.3 (Class 1) and Table I-7.3 (Class 2 & 3) for SA193 B7 Bolting[7]:

Design Stress Intensity Value = Sm = 35 Ksi for Class 1 applications

Allowable Stress Value = S = 25 Ksi for Class 2 and 3 applications

From ASME Section III, NB 3230 states that stresses for design conditions be limited to Sm. However, service conditions including preload in bolts may be higher than Sm but that average stresses shall not exceed two times Sm listed in Table I-1.3. These requirements may also be applied to Class 2 and 3 bolting[8].

Applied to the Worst Case 5/8"-11 Stud, the maximum allowable preload would be:

ASME Section III Class 1:

2 * 35 Ksi * 0.226 square inches tensile stress area = 15,820 pounds

ASME Section III Class 2 and 3:

2 * 25 Ksi * 0.226 square inches tensile stress area = 11,300 pounds

When compared to the ultimate Thread Strength for the Worst Case out-of-tolerance conditions calculated above (37,502 pounds), the thread strength is shown to have a margin of 2.37 to one above the maximum Code allowable preload.

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[7] Values specified are for 100°F and represent most conservative values

[8] ASME Section III; NB-3222 and NB 3234.

When the maximum Code allowable load under Design conditions is applied to the reduced thread shear area which was calculated for the Worst Case out-of-tolerance conditions:

ASME Section III Class 1:

Maximum allowable shear stress per ASME Code (NB-3227.2) is 0.6 Sm:

0.6Sm = 0.6 * 26,800 psi = 16,080 psi @ 700 degrees F

Maximum Design Load is limited to Sm times tensile stress area:

35 ksi * 0.226 square inches tensile area = 7910 pounds

Shear stress is the Maximum Design Load divided by the shear area:

7910 / 0.600 square inches shear area = 13,173 psi

Therefore, the calculated shear stress is less than the maximum allowable shear stress specified in ASME Section III.

Similarly for ASME Section III Class 2 and 3:

Maximum allowable shear stress per ASME Code (NC-3216.3(b)) is 0.6 S:

0.6 S = 0.6 * 25,000 psi = 15,000 psi @ 700 degrees F

25 ksi * .226 square inches tensile area = 5650 pounds Max Design Load

Shear Stress = 5650 pounds / 0.600 square inches = 9417 psi

Therefore, the calculated shear stress is less than the maximum allowable shear stress specified in ASME Section III.

Worst Case Internal Threads:

Item Description: Heavy Hex Nut 1/2" - 13 Threads per Inch (UNC)

Material Code: 305-04211 RSO#: 2583-92 Supplier: NOVA / Texas Bolt

Material Specification: ASME SA194 Grade 2H Heat Code: 1D3716

Thread Functional Size Inspection was shown to be Out-of-Tolerance:

Functional Size Reading: 0.4572 inches

Functional Size Range: 0.4565 (max) to 0.4500 (min) inches

Amount Out-of-Tolerance: 0.0007 inches

Thread Pitch Diameter Inspection was shown to be Out-of-Tolerance:

Pitch Diameter Reading: 0.4642 inches

Pitch Diameter Range: 0.4565 (max) to 0.4500 (min) inches[9]

Amount Out-of-Tolerance: 0.0077 inches

Material Proof Strength (S) = 175,000 psi (min) (ASME Section II)

Length of Thread Engagement (L_E) = 1 Diameter = 0.500 inches

Minimum Major Diameter of External Thread = D_{MIN} = 0.4876 inches[9]

Maximum Major Diameter of External Thread = 0.4985 inches[9]

[9] ANSI B-1.1, 1989 Edition, Table 3A, Class 2B Fit

THREAD STRENGTH CALCULATIONS FOR WORST CASE INTERNAL THREAD CONDITION, UTILIZING THE ABOVE EQUATIONS AND DATA:

$$A_{SN} = \pi(1/P) * L_E * D_{MIN}[\frac{1}{2(1/P)} + 0.57735\,(D_{MIN} - D2_{MAX})]$$

CASE A: Maximum Material Conditions for both Internal and External Threads

Pitch Diameter = 0.4500" (min) D_{MIN} = 0.4985" (max)

A_{SN} = 3.14159 * 13 * .500 * .4985 * [1/(2*13) + .57735(.4985 - .4500)]
= 0.6765 square inches

Thread Proof Strength = 0.5 * 0.6765 * 175,000 = 59,198 pounds (min)

CASE B: Minimum Material Conditions for both Internal and External Threads
(ANSI B-1.1)

Pitch Diameter = 0.4565" (max) D_{MIN} = 0.4876" (min)

A_{SN} = 3.14159 * 13 * .500 * .4876 * [1/(2*13) + .57735(.4876 - .4565)]
= 0.5618 square inches

Thread Proof Strength = 0.5 * 0.5618 * 175,000 = 49,158 pounds (min)

CASE C: Out-of-Tolerance Conditions from Worst Case Data

Pitch Diameter = 0.4642" (actual) D_{MIN} = 0.4876" (min)

A_{SN} = 3.14159 * 13 * .500 * .4876 * [1/(2*13) + .57735(.4876 - .4642)]
= 0.5174 square inches

Thread Proof Strength = 0.5 * 0.5174 * 175,000 = 45,270 pounds (min)

It should be noted that the Worst Case Internal Thread conditions represent only a 6.5% reduction in thread strength from the "B" case above, which represents the calculation of thread strength based on the requirements of ANSI B-1.1. An ideal thread calculation is presented in case "A" above which assumes that both the internal and external threads are at the maximum material limit. The Worst Case Internal Thread conditions are shown to have thread strength 23.5% (i.e. 17% due to tolerance band and 6.5% due to out-of-tolerance condition) lower than the ideal thread condition from case "A".

INTERNAL THREAD CALCULATION SUMMARY:

CONDITIONS	DIFFERENCE IN THREAD STRENGTH
Case A, Maximum Material contact both Int and Ext	
	} 17%
Case B, Minimum Material contact both Int and Ext	
	} 6.5%
Case C, Worst Case Out-of-Tolerance	

EVALUATION OF SERVICE APPLICATIONS:

Inherent in the design of threaded fasteners is a strength bias favoring the internally threaded components. The thread stripping areas for internal threads are 1.3 to 1.5 times those for external threads. A typical bolted joint will fail in tension at the root of the external threads. The reduced thread shear area calculated above for the Worst Case out-of-tolerance internal threads would still exceed the maximum material shear area for corresponding external threads.

Worst Case Internal Thread Shear Area = 0.5174 square inches

Maximum Material External Thread Shear Area = 0.4150 square inches

Therefore, even with the reduced shear area resulting from out-of-tolerance conditions, the Worst Case out-of-tolerance nut would still be stronger than the corresponding bolt or stud at the maximum material (max P.D.) conditions.

INDEPENDENT TESTING

Independent dimensional analysis and mechanical testing of a sample of allthread fasteners were performed for comparison with results obtained during the investigation of Nuclear Safety Concern (NSC) 93-11. The methodology and results of this testing are documented in Safety Engineering Failure Analysis Report FAR-94-005. The independent dimensional data was compared to the readings obtained using the Johnson Gages at the SONGS CGI laboratory. Additionally, mechanical testing of the samples was performed to verify the fastener material mechanical properties and thread stripping strength for comparison with calculated thread strength values, discussed in the White Paper associated with this NSC.

In summary, the following conclusions were demonstrated:

- The independent dimensional examinations generally correlated well with the Johnson Gage data
- The material physical test data corresponded well with that contained on the supplier Certified Material Test Reports
- The tensile test samples failed at the thread minor diameter cross section at a load very close to calculated values
- The thread stripping strength pull test results were consistent with the calculated thread strength values for the known out-of-tolerance pitch diameter data of the samples
- The thread strength, even for out-of-tolerance conditions, was shown to be inherently larger (1.3 to 1.5 Times) than the tensile load bearing capability of the externally threaded fasteners
- Even though the tested samples were shown to be out-of-tolerance with respect to pitch diameter, they developed thread strength well in excess of ASME Code requirements

The fasteners which were the subject of this test included sample pieces which represent the Worst Case out-of-tolerance example observed during sampling of Warehouse fastener stock.

GENERIC APPLICABILITY OF RESULTS

Calculations have shown that the Worst Case out-of-tolerance conditions can be generalized across all sizes of internal and external threaded fasteners. Expressing the out-of-tolerance conditions as a percentage of nominal fastener diameter will produce a relatively constant percentage reduction in thread strength, for an equivalent Worst Case condition, across all sizes of fasteners. Analysis of the most severe service conditions at SONGS was reviewed. High Temperature ASME Section III Class 1 service requirements were researched and fastener loading for these applications were found to be below the applicable ASME Code allowances. Given that the fastener strength reduction calculated from the Worst Case out-of-tolerance conditions (above) left the thread strength with a 2.37 to 1 margin over ASME Code preload stress allowables, it can be concluded that the Worst Case conditions observed during the sample of Warehouse stock, generalized across all sizes and materials, contain adequate strength margin for service in the most severe service conditions at SONGS.

MAXIMUM HYPOTHETICAL DEVIATION

In an effort to demonstrate the maximum hypothetical impact on fastener strength that could result from out-of-tolerance pitch diameter conditions, calculations were performed to determine the conditions necessary for the threads to actually strip when the maximum Code allowed preload was applied to the fastener. In other words, these are the reduced-thread conditions where the threads would strip during installation. For comparison purposes, these calculations were performed for a fastener of similar nominal size and material as that identified as the Worst Case externally threaded fastener from the Warehouse sample and for the most severe (ASME Class 1) service conditions. To restate the measured conditions for this 5/8"-11 allthread:

Maximum Pitch Diameter	=	0.5644"	(ANSI B-1.1)
Minimum Pitch Diameter	=	0.5589"	(ANSI B-1.1)
Measured Pitch Diameter	=	0.5554"	(Warehouse Sample)

		0.0035"	Out-Of-Tolerance

For Thread Strength Equal to Maximum Design Load Condition:

Calculated Pitch Diameter	=	0.5044"	(ASME Class 1)

		0.0545"	Out-Of-Tolerance

This out-of-tolerance condition represents a factor <u>15.6</u> times worse than the measured Worst Case from this Warehouse sample. A fastener this grossly out-of-tolerance condition would easily fail visual inspection, either at Receipt Inspection, or by the craft at installation; and would certainly not pass System 21 Go/No-Go inspection.

POTENTIAL FOR FASTENER LOOSENING

Vibrating environments, material relaxation and material fatigue are conditions which may result in the loosening and failure of fasteners. The potential for vibration and material relaxation and fatigue of bolted connections is routinely assessed in the design of equipment and components for operating service conditions. The following discussion assesses the affect of the measured out-of-tolerance conditions on vibration, material relaxation and fatigue.

VIBRATION

Vibration has been shown, under certain conditions, to cause the loosening of threaded fasteners. The vibration can overcome the friction forces which act between the faces of the mating interface of a bolted joint and also the friction forces at the face of the nut and/or bolt. If these friction forces, which can be typically 80 - 90% of the torque loading, are overcome or negated by the effects of vibration, the energy stored in the fastener will be released and the bolt will return to its original length with the inclined plane of the bolt threads pushing the inclined plane of the nut threads out of the way. Vibration is theorized to negate the friction forces by creating a rapid series of small relative motions between the thread mating faces in a direction perpendicular to the friction forces. Vibration loosening is agreed to occur more commonly in fasteners loaded in shear, especially those with vibration forces acting perpendicular to the axis of the fastener. Fasteners loaded in tension are therefore, less susceptible to the effects of vibration. After loss of sufficient preload in a bolted joint susceptible to vibration, the friction forces will be reduced sufficiently to allow the nut to back off.

Higher initial preload can mitigate the effects of vibration. A higher preload will increase the friction forces between the thread faces making them less susceptible to the small relative motions which negate the friction forces. In many cases, a sufficiently high preload can create a completely vibration-resistant bolted joint. In other cases, more direct physical means must be considered to prevent relative motion between the nut and bolt:

- Utilize locking devices or other form of action to prevent relative motion between the nut and bolt.
- Mechanically prevent slippage between bolted joint surfaces loaded in shear to prevent slip between bolt and joint surfaces.
- Utilize fine thread bolting to reduce the helix angle of threads and thereby reduce the back-off torque caused by the preload.

The consideration of vibration as a potential cause of fastener loosening is performed during the design of individual components for SONGS. Vendor manuals routinely specify the torque values for bolting, and these values are incorporated into maintenance procedures. These values conform to those specified by ASME Code for bolt preloads and have been evaluated by the vendor as satisfactory for the service application. The adequacy of these values is also verified by hydrotest of the component and system, which is usually conducted at 1.5 times the Design pressure. If fastener loosening, as evidenced by leaking joints, is discovered during operation or maintenance, programs are in place to require engineering evaluation of the conditions, and for the specification of corrective measures to prevent recurrence. Corrective measures may include the increase of fastener preload or any of the options listed above, as long as they are appropriately documented on design documents and procedures are updated.

In the consideration of the susceptibility of the Worst Case out-of-tolerance fastener condition identified during the Warehouse sample inspection to the effects of vibration, it should be recognized that the thread mating area would be slightly reduced from that of in-tolerance fasteners, but the total clamping forces would remain the same. The friction forces would remain constant for a given preload irrespective of a minor reduction in thread mating area. It was shown above that even with the reduced shear area from the Worst Case out-of-tolerance condition, the threads maintained a significant margin of strength above Code maximum allowable stresses and would therefore, accommodate any preload increase required for specific system or component considerations.

MATERIAL RELAXATION

Short term relaxation can create a reduction in preload. The most common cause of short term relaxation is thread embedment. The loss of preload occurs when tiny high spots on thread surfaces are overcome by pressure from clamping forces. Plastic deformation of the high spots occurs until enough of the total thread surface is loaded to prevent further deformation. Embedment is more common on new parts than on used ones due to the smoothing of thread surfaces that occurs as fasteners are torqued. Critical SONGS bolting applications require torquing of fasteners. Embedment loss of preload was identified through the SONGS Root Cause Program and a successful anti-embedment process was incorporated in applicable procedures as a corrective measure. Fasteners in the Worst Case out-of-tolerance condition would be less susceptible to embedment loss of preload due to the slightly reduced area of thread mating surfaces which would create higher contact forces during torquing. These slightly higher forces would reduce the effects of embedment, for a given torque force, by more effectively smoothing away the slight irregularities which cause the embedment.

Long term relaxation can also create a reduction in preload. This creep or stress relaxation involves the slow shedding of load by a fastener under constant deflection (strain). This process is encouraged by high temperatures. The effects of this relaxation vary for different materials and temperatures and must be considered during the original design of nuclear equipment and systems. The Worst Case out-of-tolerance condition identified in the Warehouse Sample Inspection would not create a condition to further any fasteners susceptibility to this phenomenon. The Worst Case minor reduction in thread shear area did not decrease the fasteners thread shear area below the point where thread shear strength would be less than fastener tensile strength. Therefore, the dominant effects of any relaxation would occur across the tensile area, or body, of the fastener. It should also be noted that the effects of creep and stress relaxation occur largely at temperatures above one half of the material melting temperature (Tm expressed in degrees Kelvin). The highest fastener material temperatures at SONGS are conservatively shown to be less than this value.

Thus, the effects of material relaxation would not be exacerbated by the minor thread form deviations noted in the performance of this evaluation.

MATERIAL FATIGUE

The ASME Code provides requirements for the evaluation of the suitability of bolting and bolting materials for cyclic service, including stress limits and design fatigue curves. Minor reductions in thread shear area would have no impact on the fatigue failure of fasteners. Bolting stresses are concentrated at the root of the thread and any cracking propagating from fatigue, even if initiated in the thread material would be expected to propagate through the thread root and across the plane of the minimum tensile area. Samples of stock from lots containing the Worst Case internal and external threaded fasteners were dimensionally examined by independent laboratories. The tensile root stress area for the Worst Case fastener was found to be independent of measured pitch diameter and major diameter readings. A sample of the Worst Case externally threaded stock was machined and destructively tested to verify material properties stated on the supplier Certified Material Test Report. A tensile pull test of a section of this threaded stock was then performed, and tensile area was calculated from the results. The tensile area was found to be essentially unchanged from the design value and correlated well with the areas calculated from dimensional readings. This testing verified that the tensile areas of Worst Case fasteners remain essentially unchanged and therefore the stresses within the fastener would be unaffected by the measured out-of-tolerance thread conditions exhibited during sample inspection. The Worst Case out-of-tolerance condition identified in the Warehouse Sample Inspection would not create a condition to further fastener susceptibility to fatigue failure.

SYSTEM LEAKAGE MONITORING

The above discussions have stated that minor out-of-tolerance conditions, of the sort that could be expected if a fastener were to pass System 21 inspection but fail System 22, would not create conditions to increase fastener susceptibility to loosening or failure. SONGS systems and programs have, however, been designed for early detection, control and the prevention of recurrence of any leakage. SONGS systems are continuously monitored for evidence of leakage through routine operator rounds and monitoring of primary system inventory balance. Effective programs exist to correct any identified leakage and ensure corrective actions.

FASTENER EXPERIENCE EVALUATION

Reviews of both site information and industry experience were conducted to understand the causes of any reported fastener failures. Many cases of fastener failures have been analyzed both at SONGS and in the nuclear industry. A search of SONGS maintenance and nonconformance databases identified no conditions of bolting failure due to out-of-tolerance threadform. The SONGS Root Cause program has been effective in identifying causes and recommending corrective actions for bolting failures including the recommendation of alternate materials, alternate preload and the anti-embedment process. Review of SONGS Root Cause database and discussions with responsible personnel indicated that bolting failures had been analyzed, however, the causes of the failure were clearly determined to not be due to out-of-tolerance threadform. Fasteners were evaluated which had failed due to the following causes:

- Stress corrosion cracking of stud material
- Losses of preload due to improper (soft) washer material
- Corrosion of fasteners
- Overload and Overtorquing
- Cold work induced material embrittlement
- Thread embedment - special case where fastener loadings and vibration were very high. Anti-embedment torquing technique recommended by Root Cause Engineering has successfully addressed this condition.

The SONGS Root Cause program is sensitive to the potential that threadform may contribute to fastener failure, and utilizes the thread analysis capability of the SONGS CGI lab to investigate any suspect conditions.

Nuclear industry oversight organizations and interest groups routinely provide information on industry equipment failure and trend performance of specific categories of components, such as fasteners. A review of databases and reports was conducted with respect to fastener failures. Most recently, the NRC issued Generic Letter 91-17, "Bolting Degradation or Failure in Nuclear Power Plants", which closed Generic Safety Issue 29, on the same subject, based on studies conducted by EPRI, MPC and AIF which analyzed many fastener failures. In closing this issue, the NRC found no evidence to indicate that failures were directly attributable to dimensionally nonconforming fasteners. Thus, careful evaluation of cumulated nuclear operating experience has shown that no safety issues exist with current industry fastener inspection practices.

TECHNICAL CONCLUSIONS

The inspection of a statistical sample of Warehouse fastener stock with Johnson Gage equipment has identified dimensionally out-of-tolerance conditions in sample items which were shown to be 99% acceptable when inspected with the industry accepted standard of final thread dimensional inspection, Go/No-Go gages. The out-of-tolerance conditions were analyzed for impact on ultimate thread strength. It was shown that when compared to the ultimate thread strength for the Worst (observed) Case out-of-tolerance conditions a minimum margin of safety of 2.37 to one above the maximum fastener preload remains. Independent testing was performed which confirmed the calculated expectations on actual out-of-tolerance samples. It can also be concluded that the increased shear stress created by the reduced thread shear area of the Worst Case thread condition are less than the ASME Code stress allowables for the Section III Class 1, 2 and 3 Design conditions. The effects of vibration, material relaxation and fatigue were assessed to compare the susceptibility for the Worst Case out-of-tolerance conditions identified during the Warehouse Sample Inspection to facilitate fastener loosening or failure from these most common recognized failure modes. No increase in susceptibility was found during this evaluation. It was also shown that the Worst Case conditions observed during the sample of Warehouse stock, if generalized across all sizes and materials, contain adequate strength margin for service in the most severe conditions at SONGS; and that adequate thread area remains to develop the full bolt load without thread stripping and will accommodate any preload adjustments required for specific service conditions.

In an effort to demonstrate the maximum hypothetical impact on fastener strength that could result from out-of-tolerance pitch diameter conditions, calculations were performed to determine the conditions necessary for the threads to actually strip when the maximum Code allowed preload was applied to the fastener. This value was shown to be 15.6 times greater than the maximum measured deviation from ANSI B-1.1 tolerances observed for external thread in this Warehouse sample. Fasteners this grossly out-of-tolerance would easily fail visual inspection, either at Receipt Inspection or by the craft at installation, and would certainly not pass System 21 Go/No-Go inspection.

ADDENDUM TO FASTENER STRENGTH ANALYSIS

<u>COMBINED-CASE</u>
<u>THREAD STRENGTH ANALYSIS</u>

COMBINED-CASE FASTENER STRENGTH ANALYSIS

BACKGROUND

This evaluation calculates the combined effect on the strength of a threaded joint comprised of an externally threaded fastener, which has been shown to be out-of-tolerance with respect to pitch diameter, and an internally threaded fastener which is also out-of-tolerance with respect to pitch diameter. The values for pitch diameter utilized in this evaluation will correspond with the maximum observed deviations from specified values[10] reported in warehouse sample inspections associated with this White Paper. The externally threaded fastener to be considered in this analysis will be the 5/8" all-thread stock described as "Worst Case" by System 22 inspection, in the body of this report. The internally threaded nut to be considered in this analysis will be a hypothetical 5/8" nut, with out-of-tolerance conditions proportionally adjusted from the System 22 data for the "Worst Case" 1/2" nut described in the body of this report. Additional detail has been added to the technical discussion of this evaluation to provide greater clarity to the fastener strength evaluation.

Thread Shear Strength

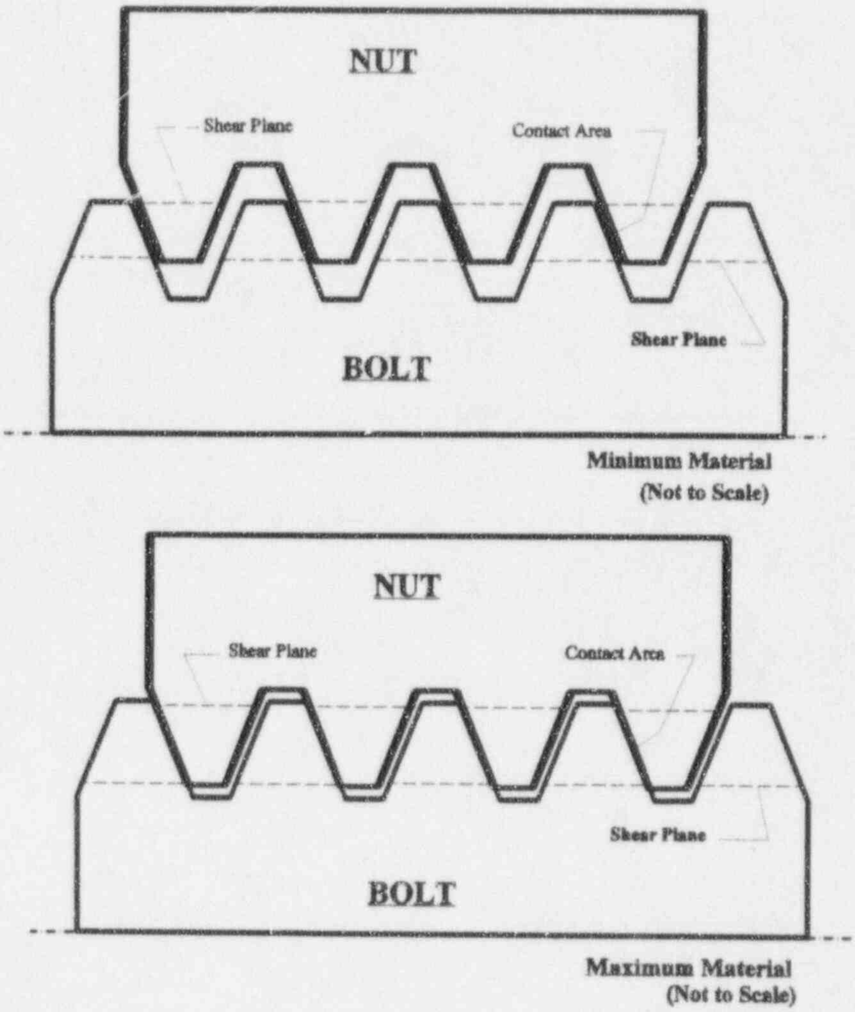
Ultimate thread strength is comprised of the amount of shear area contained in the shear plane where the internal and external threads mate, and the respective fastener material strengths. The thread shear area varies proportionally with thread pitch diameter. This is to say that:

- As the internal thread (nut) pitch diameter increases,
  and/or the bolt pitch diameter decreases,
  the nut threads engage less of the bolt threads,
  thus decreasing the thread shear area.

- As the external thread (bolt) pitch diameter increases,
  and/or the the nut pitch diameter decreases,
  the bolt threads will engage further into the nut threads,
  thus increasing the thread shear area.

The following figures demonstrate these prinicples:

[10] ASME/ANSI B-1.1, 1989 Edition



Shear Plane Width Increases With Deeper Thread Engagement

Both internal and external threads have respective maximum and minimum criteria for pitch diameter. If the minimum value of external thread (bolt) pitch diameter is combined with the maximum value of internal (nut) pitch diameter, the condition known as "minimum material" is created. This means that the threads are engaged to the minimum amount allowed by the specifications, and create the minimum thread contact and shear areas for thread strength.

If the maximum value of external thread (bolt) pitch diameter is combined with the minimum value of internal thread (nut) pitch diameter, the condition known as "maximum material" is created. This means that the threads are engaged to the maximum amount allowed by the specifications, and create the maximum thread contact and shear areas for ultimate thread strength.

The maximum and minimum material thread conditions referenced above were analyzed in the body of this report as Cases "A" and "B" of the thread strength calculations for both internal and external threads.

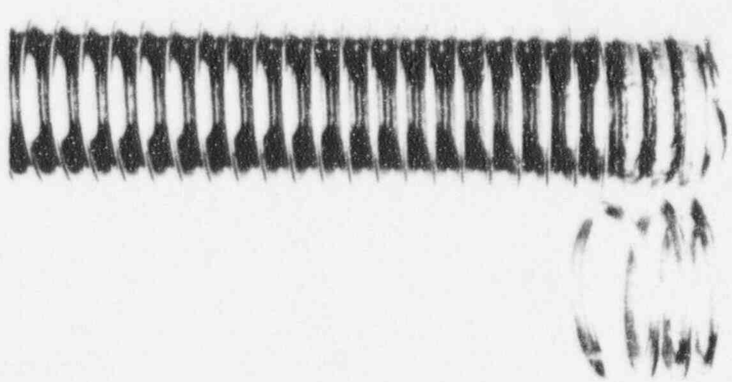
In the case to be analyzed in this example, the pitch diameter of the externally threaded item (allthread) will be below the specified amount, and the pitch diameter of the internally threaded nut will be above the specified value. This will create a condition of less thread shear area than the minimum material condition. Calculations, from formulas utilized in the body of this report, will assess the impact on this combined out-of-tolerance thread shear area and also the resulting effect on thread ultimate strength.

EVALUATION

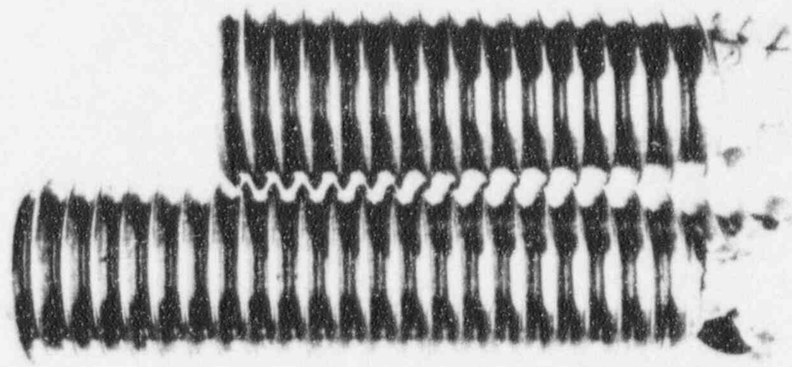
The determination of the strength of the threads of a threaded joint requires the calculation of the location of the shear plane in the thread profile, and its width.

Fastener Failure Modes

The shear plane is the location at which the threads would fail or strip when subjected to overload forces, such as could occur due to overtorquing. The following photograph from Nuclear Oversight Failure Analysis Report (FAR) 94-05 demonstrates the nature of the shear plane after threads have been stripped during tensile testing:



Another mode of failure for a bolted joint is tensile failure at the cross-sectional area of the externally threaded fastener when overloaded. This type of tensile failure is demonstrated by the following photograph from Nuclear Oversight Failure Analysis Report (FAR) 94-05:



Calculations and independent testing conducted by Nuclear Oversight and documented in Failure Analysis Report 94-05, have shown that for an externally threaded fastener, the strength of the threads, over a length of engagement equal to one nominal diameter, is greater than the tensile strength of its cross-sectional area. This strength ratio will be examined later in this combined-case example.

Additionally it can be demonstrated that an internally threaded fastener (nut) made from the same material as an externally threaded fastener (bolt/stud) will have 1.3 to 1.5 times[11] as much thread shear area, and therefore greater thread strength than that of the bolt or stud (bolt will be used here for convenience). This is due to the greater diameter of the location of the shear plane in the nut threads over that of the bolt threads, and resulting greater shear area length throughout the circumferential length of thread engagement. The fastener design materials at SONGS typically require the nut material strength to be equal to, or greater than the material strength of the bolt. Therefore the limiting factor for the design strength of a bolted connection is the tensile strength of the bolt material at its cross-sectional area, as demonstrated in the photograph above.

[11] "An Introduction To The Design And Behavior Of Bolted Joints", John H. Bickford, Second Edition

Thread Shear Plane Area

The internal and external threads come in contact in the mating surface contact area over the length of thread engagement. The graphic figures on Page 30 illustrate this description. The shear plane of the bolt exists at the inside diameter of the mating surface contact area. The shear plane of the nut exists at the outside diameter of the mating surface contact area. The greater radius of the position of the shear plane for the nut, when rotated circumferentially throughout the length of engagement of the mating surfaces, results in a longer shear plane than the bolt shear plane (which has essentially the same width).

This difference in length is between 1.3 to 1.5 times that of the bolt, depending on the width of the mating surface contact area. The mating surface contact area tends to be wider for conditions of greater thread engagement. A wider contact are will result in a greater difference in internal and external thread shear area length. It is the length of the shear plane, multiplied by its width which gives the shear area. The shear area (measured in square inches), when factored into the shear strength properties (measured in pounds per square inch) of the particular fastener material, gives the maximum force (in pounds) that the thread can withstand prior to shearing or stripping. Factors such as nut dilation and thread bending may affect the ultimate load capacity of a threaded joint, however their effect is considered neglible when heavy hex nuts are specified[12] and at loads less than the maximum preloads allowed by ASME Section III.

Combined Case Thread Failure Location

From the above, it can be concluded that the greater length of shear plane inherent in a nut, results in greater ultimate thread shear strength over the length of engagement between the nut and bolt. For an ideal case which excludes possible failure of the fastener at the tensile cross-section area, the greater strength inherent in the threads of the nut would result in failure of the bolt threads when the threaded joint is subjected to overload conditions. The failure of the bolt threads would occur at the shear plane on the inside diameter of the thread mating surface contact area, which is the bolt (external thread) shear plane. This type of failure is demonstrated in the photograph on Page 31.

The fastener design materials at SONGS typically[12] require the nut material strength to be equal to, or greater than, the material strength of the external fastener. For this case, the nut material (SA 194-2H) is stronger than the allthread material (SA 193-B7). This nut material strength is reflected in greater material strength at the nut shear plane than at the allthread shear plane. This greater material strength at the nut shear plane further empasizes that the overload thread failure of this bolted connection would occur at the shear plane of the bolt (external) threads (excluding consideration of failure at the bolt tensile cross-section area). This assumption will be verified through the calculation of nut thread strength for this combined-case example, and comparison with the bolt thread strength.

[12] 90004, Piping & Material Specification, San Onofre Units 2 and 3.

CALCULATIONS

In order to determine the force necessary to cause overload failure of the threads, the thread shear areas must first be determined. The location that would be most susceptible to overload failure must also be determined. Formulas for the determination of internal and external thread shear areas are restated here:

A_{SS} = Minimum Thread Shear Area for External Threads :

$$A_{SS} = \pi (1/P) * L_E * D1_{MAX}[\frac{1}{2(1/P)} + 0.57735 (D2_{MIN} - D1_{MAX})]$$

A_{SN} = Minimum Thread Shear Area for Internal Threads:

$$A_{SN} = \pi (1/P) * L_E * D_{MIN}[\frac{1}{2(1/P)} + 0.57735 (D_{MIN} - D2_{MAX})]$$

Where:

	π	= 3.14159
	1/P	= Number of Threads per Inch
	L_E	= Length of Engagement
	D_{MIN}	= Minimum Major Diameter of External Thread
	D_{1MAX}	= Maximum Minor Diameter of Internal Thread
	D_{2MAX}	= Maximum Pitch Diameter of Internal Thread
	D_{2MIN}	= Minimum Pitch Diameter of External Thread

These equations consider the location of the shear plane on the thread profile, and determine its width for a specific pitch diameter. This value is multiplied in the equation by the length of engagement to give the total area of the shear plane. The length of engagement for this example is given to be that of a heavy-hex nut, which equals one nominal diameter in length.

For the externally threaded item which was shown to exhibit the greatest deviation from specified tolerances during warehouse sampling associated with this paper, the following data was obtained and is restated for this example:

External Threads:

Item Description: All Thread Stud, 5/8"dia-11 (UNC) by 36" length

Material Code: 305-05606 RSO#: 2063-93 Supplier: NOVA Inc.

Material Specification: ASME SA193 Grade B7 Heat Code: 8099572

Thread Functional Size Inspection was performed Satisfactorily

Thread Pitch Diameter Inspection was shown to be Out-of-Tolerance.

Pitch Diameter Reading: 0.5554 inches = D_{2MIN}

Pitch Diameter Range: 0.5644 (max) to 0.5589 (min) inches.[13]

Amount Out-of-Tolerance: 0.0035 inches

Ultimate Material Strength (S_T) = 125,000 psi (min)[14]

Length of Thread Engagement (L_E) = 1 Diameter = 0.625 inches

[13] ANSI/ASME B-1.1, 1989 Edition, Table 3A, Class 2A Fit

[14] ASME Section III, 1977 Edition, Appendix I

For the internally threaded item which was shown to exhibit the greatest deviation from specified tolerances during warehouse sampling associated with this paper, the following data was obtained and is restated for this example:

Internal Threads:

Item Description: Heavy Hex Nut 1/2" - 13 Threads per Inch (UNC)

Material Code: 305-04211 RSO#: 2583-92 Supplier: NOVA / Texas Bolt

Material Specification: ASME SA194 Grade 2H Heat Code: 1D3716

Thread Functional Size Inspection was shown to be Out-of-Tolerance:

Functional Size Reading: 0.4572 inches

Functional Size Range: 0.4565 (max) to 0.4500 (min) inches

Amount Out-of-Tolerance: 0.0007 inches

Thread Pitch Diameter Inspection was shown to be Out-of-Tolerance:

Pitch Diameter Reading: 0.4642 inches

Pitch Diameter Range: 0.4565 (max) to 0.4500 (min) inches[15]

Amount Out-of-Tolerance: 0.0077 inches

Material Proof Strength (S) = 175,000 psi (min) (ASME Section II)

Length of Thread Engagement (L_E) = 1 Diameter = 0.500 inches

Minimum Major Diameter of External Thread = D_{MIN} = 0.4876 inches

Maximum Major Diameter of External Thread = 0.4985 inches

[15] ANSI/ASME B-1.1, 1989 Edition, Table 3-A, Class 2A Fit

Pitch Diameter of Hypothetical Nut

In order to perform a combined-case evaluation utilizing the out-of-tolerance conditions exhibited by 5/8" allthread and the 1/2" nut, the Pitch Diameter readings from the nut must be converted to a proportionally equivalent out-of-tolerance reading for a hypothetical 5/8" nut. This may be accomplished by two different methods; by ratio of pitch diameter tolerances for 1/2"-13 internal threads and those for 5/8"-11 threads, or by a direct ratio of fastener nominal diameters.

Tolerance Ratio

Tolerance in this case is the difference between the Maximum and Minimum specified pitch diameters for each of the two size fasteners. Tolerances for pitch diameter do not vary linearly with respect to fastener nominal diameter or thread pitch, that is to say that the tolerances for a 1" fastener are not twice as large as those for a 1/2" fastener. From Table 3A of ASME B-1.1:

5/8"-11 Class 2B Internal Thread Pitch Diameter Tolerance = 0.0072"

1/2"-13 Class 2B Internal Thread Pitch Diameter Tolerance = 0.0065"

Therefore, for a 1/2"-13 nut pitch diameter out-of-tolerance reading to be proportionally equivalent to one for a 5/8"-11 nut, it would have to be multiplied by the ratio of the tolerances shown above:

5/8"-11 to 1/2"-13 Tolerance Ratio = 0.0072" ÷ 0.0065" = 1.108

Multiplication of the 1/2"-13 nut pitch diameter out-of-tolerance reading by this Tolerance Ratio will result in a proportionally equivalent out-of-tolerance value for the hypothetical 5/8"-11 nut to be analyzed in this combined case:

1/2"-13 nut out-of-tolerance reading (from above data) = 0.0077"

Multiplying by the Tolerance Ratio: 0.0077" × 1.108 = 0.0085"

Nominal Diameter Ratio

Developing a proportionally out-of-tolerance pitch diameter for the hypothetical 5/8"-11 nut by a nominal diameter ratio would involve the following:

Nominal Diameter Ratio = 5/8" ÷ 1/2" = 1.25

1/2"-13 Nut Pitch Diameter Out-of-Tolerance Amount (from above data) = 0.0077"

Multiplying by the Nominal Diameter Ratio: 0.0077" × 1.25 = 0.0096"

A greater out-of-tolerance condition is created by using the Nominal Diameter Ratio, therefore this number will be conservatively utilized for the purposes of this example. From this, the following data may be stated for the hypothetical 5/8"-11 nut to be used in this analysis:

Item Description: Heavy Hex Nut 5/8" - 11 Threads per Inch (UNC)

Material Specification: ASME SA194 Grade 2H

Thread Pitch Diameter Out-of-Tolerance:

Pitch Diameter Range: 0.5732 (max) to 0.5660 (min) inches

Amount Out-of-Tolerance: 0.0096 inches

Pitch Diameter Reading: 0.5828 inches (Hypothetical)

Material Proof Strength (S) = 175,000 psi (min) (ASME Section II)

Length of Thread Engagement (L_E) = 1 Diameter = 0.625 inches

Calculation of External Thread Combined-Case Shear Strength

From the preceding discussion, it can be expected that for this combined-case, thread shear failure from overload forces would occur at the external thread shear plane (excluding possible failure at the fastener tensile cross-sectional area). The shear area and resulting ultimate thread strength for the external threads will be calculated first, and will then be compared to calculations for the internal thread shear area and thread strength.

The combined case thread strength calculation for the combination of the 5/8"-11 allthread and hypothetical 5/8"-11 nut described above can be performed using the following equation to determine the thread shear area in the shear plane of the all thread:

A_{SS} = Minimum Thread Shear Area for External Threads :

$$A_{SS} = \pi(1/P) * L_E * D1_{MAX}\left[\frac{1}{2(1/P)} + 0.57735\,(D2_{MIN} - D1_{MAX})\right]$$

Where:

	π	= 3.14159
	$1/P$	= Number of Threads per Inch
	L_E	= Length of Engagement
	D_{1MAX}	= Calculated Minor Diameter of Internal Thread
	D_{2MIN}	= Actual Pitch Diameter of External Thread

The preceding equation calculates the shear area for the specific allthread pitch diameter at the place in the thread profile where the minor diameter of the nut contacts the allthread.

Adjusted Minor Diameter

The routine calculation of minimum external thread shear area utilizes a Maximum Minor Diameter of the Internal Thread, or D_{1MAX}, which is specified in ASME B-1.1. In this combined-case analysis, the value of D_{1MAX} is conservatively modified to reflect the out-of-tolerance pitch diameter condition considered for the hypothetical 5/8"-11 nut. In the basic form of the screw thread profile described in ASME B-1.1, the pitch diameter is defined as falling at the halfway point in the Height (H) of the sharp V-thread (fundamental triangle). The height of thread engagement is that of H, minus the losses associated with the crest and root of the thread, and is equal to 0.625 H. Truncations associated with the crest and root of the thread are essentially equal therefore, the pitch diameter lies at the halfway point of the height of thread engagement, for both internal and external threads. For internal threads, the minor diameter of the nut is equal to the major diameter at the root of the thread plus twice the height of thread engagement (accounting for thread engagement on opposing sides of the nut). Therefore, in accordance with the basic form of the screw thread, a variation in pitch diameter would have a corresponding effect on the height of thread engagement equal to 1/2 the variation in pitch diameter (assuming a constant major diameter). This resulting variation in thread height, when considered for opposing positions on the nut diameter, would result in the nut minor diameter varying an amount twice that of the thread height variation, which would equal the amount of pitch diameter variation. Therefore we can conclude, for the purposes of this evaluation, that a given variation outside of specified tolerances for nut pitch diameter will produce an equivalent variation outside specified tolerances for nut minor diameter.[16] This would result in the following:

5/8"-11 Nut Pitch Diameter Calculated Reading	=	0.5828"
		−
5/8"-11 Nut Pitch Diameter Spec Maximum	=	0.5732"
Amount Out-of-Tolerance	=	0.0096"
5/8"-11 Nut Maximum Minor Diameter [16]	=	0.5460"
		+
Add Equivalent Pitch Diameter Out-of-Tolerance	=	0.0096
Resultant Adjusted Nut Minor Diameter (D_{1MAX})	=	0.5556"

[16] Independent thread dimensional inspection performed in accordance with Nuclear Oversight Division Failure Analysis Report (FAR) 94-05 did not demonstrate a linear correllation between pitch diameter variations and major (minor) diameter readings. Therefore, the maximum Minor Diameter is being conservatively utilized in this calculation to result in an out-of-tolerance Adjusted Minor Diameter for this example.

Applying this value to the above equation for external thread shear area:

$$A_{SS} = \pi (1/P) * L_E * D1_{MAX} [\frac{1}{2(1/P)} + 0.57735 (D2_{MIN} - D1_{MAX})]$$

$D2_{MIN}$ = Allthread Pitch Diameter = 0.5554" (Page 35)

$D1_{MAX}$ = Nut Adjusted Minor Diameter = 0.5556" (Page 39)

A_{SS} = 3.14159 * 11 * 0.625 * 0.5556 [1/(2*11) + .57735(0.5554 - 0.5556)]

A_{SS} = <u>0.5441 in^2</u>

Factoring this combined-case external thread shear area into the equation to determine thread shear strength:

$$\textit{Shear Strength Of External Threads} = 0.5S_T * (A_{SS})$$

Where:

S_T = Ultimate Tensile Strength of Allthread Material
= 125,000 psi (ASME Section II)

A_{SS} = Calculated Thread Shear Area for External Threads

Shear Strength of Allthread Threads = 0.5 * 125,000 psi * 0.5441 in^2

= <u>34,006</u> pounds force

Thus, the external threads of the 5/8"-11 allthread could be expected to fail when subjected to a load in excess of approximately 34,000 pounds.

Calculation of Internal Thread Combined-Case Shear Strength

In order to verify the location of expected thread failure, the shear area and resulting ultimate thread strength for the internal nut threads will be calculated and compared to the allthread external thread shear area and thread strength. The combined-case thread strength calculation for the combination of the 5/8"-11 allthread and hypothetical 5/8"-11 nut can be performed using the following equation to determine the thread shear area in the shear plane of the nut threads:

A_{SN} = Thread Shear Area for Internal Threads:

$$A_{SN} = \pi (1/P) * L_E * D_{MIN}[\frac{1}{2(1/P)} + 0.57735 (D_{MIN} - D2_{MAX})]$$

Where:

- π = 3.14159
- $1/P$ = Number of Threads per Inch
- L_E = Length of Engagement
- D_{MIN} = Calculated Major Diameter of External Thread
- D_{2MAX} = Calculated Pitch Diameter of Internal Thread

This equation calculates the shear area for the pitch diameter of the hypothetical nut, at the place in the thread profile where the major diameter of the allthread contacts the nut.

The routine calculation of minimum internal thread shear area utilizes a Minimum Major Diameter of the External Thread, or D_{MIN}, which is specified in ASME B-1.1. In this combined-case analysis utilizing the above equation, the value of D_{MIN} is conservatively modified to reflect the out-of-tolerance pitch diameter condition of the 5/8"-11 allthread. In the basic form of the screw thread profile described in ASME B-1.1, the pitch diameter is defined as falling at the halfway point in the Height (H) of the sharp V-thread (fundamental triangle). The height of thread engagement is that of H, minus the losses associated with the crest and root of the thread, and is equal to 0.625H. Truncations associated with the crest and root of the thread are essentially equal and therefore the pitch diameter also lies at the halfway point of the height of thread engagement, for both internal and external threads. For external threads, the major diameter of the fastener is equal to the minor diameter at the root of the thread plus twice the height of thread engagement (accounting for thread engagement on opposing sides of the bolt). Therefore, in accordance with the basic form of the screw thread, a variation in pitch diameter would have a corresponding effect on the height of thread engagement equal to 1/ 2 the variation in pitch diameter (assuming a constant minor diameter). This resulting variation in thread height, when considered for opposing positions on the allthread diameter, would result in the allthread major diameter varying an amount equal to twice that of the thread engagement height variation, which would equal the amount of pitch diameter variation.

Therefore we can conclude, for the purposes of this evaluation, that a given variation below specified tolerances for allthread pitch diameter will produce an equal variation in major diameter below specified values.[17] This would result in the following:

5/8"-11 Allthread Pitch Diameter Minimum	=	0.5589"
		-
5/8"-11 Allthread Pitch Diameter Reading	=	0.5554"

Amount Out-of-Tolerance	=	0.0035"

Applying this amount of variation in pitch diameter, outside of specified tolerances to adjust the Major Diameter of the allthread:

5/8"-11 Allthread Minimum Major Diameter	=	0.6113"
		-
Subtract Equivalent Pitch Diameter Out-of-Tolerance	=	0.0035

Resultant Adjusted Allthread Major Diameter (D_{MIN})	=	0.6078"

Applying this value to the equation for internal thread shear area:

$$A_{SN} = \pi (1/P) * L_E * D_{MIN} [\frac{1}{2(1/P)} + 0.57735 (D_{MIN} - D2_{MAX})]$$

$D2_{MAX}$ = Nut Pitch Diameter = 0.5828" (Page 38)

D_{MIN} = Allthread Major Diameter = 0.6078" (Above)

A_{SN} = 3.14159 * 11 * 0.625 * 0.6078 [1/(2*11) + .57735(0.6078 - 0.5828)

A_{SN} = 0.7862 in^2

---

[17] Independent thread dimensional inspection performed in accordance with Nuclear Oversight Division Failure Analysis Report (FAR) 94-05 did not demonstrate a linear correlation between pitch diameter variations and major diameter readings. Therefore, the minimum Major Diameter is being conservatively utilized in this calculation to result in an out-of-tolerance Adjusted Major Diameter for this example.

It should be noted that the combined-case nut shear area of 0.7862 in^2 is 1.45 times the value of the combined-case allthread shear area of 0.5441 in^2, and is 1.26 times the minimum material external thread shear area of 0.6239 in^2 (Case B).

Factoring this combined-case internal thread shear area into the equation to determine thread shear strength:

$$\textbf{\textit{Shear Strength of Internal Threads}} = 0.5S_T*(A_{SN})$$

Where:

A_{SN} = Minimum Thread Shear Area for Internal Threads
S_T = Proof Load Strength of Nut Material

Shear Strength of Internal Threads = 0.5 * 175,000psi * 0.7862 in^2

= <u>68,793</u> pounds

Thus, the proof load that the combined-case nut threads could withstand without yielding would be 68,793 pounds, or approximately twice the ultimate strength of the combined-case allthread external threads. From this information, we can conclude that if fastener tensile failure is not considered, the combined-case threaded joint would fail when subjected to an overload condition of approximately 34,400 pounds and would fail at the shear plane of the allthread material.

<u>Externally Threaded Fastener Tensile Strength</u>

The ultimate tensile strength of the externally threaded fastener (in pounds) equals the ultimate strength of the fastener material (in pounds per square inch) multiplied by the fastener tensile cross-sectional area (in square inches). For the SA 193-B7 allthread, the ultimate material strength is specified in ASME Section II as being a minimum of 125,000 psi. The ASME Code tensile area is based on the root area, which for the 5/8"-11 thread is 0.202 square inches. The calculation of tensile strength:

125,000 psi × 0.202 in^2 = <u>25,250</u> pounds (min).

The value of 25,250 pounds tensile strength is less than the combined-case external thread strength and therefore overload of the threaded joint would result in failure of the allthread in tension, not in stripped threads. It should be noted that the combined-case out-of-tolerance thread strength (34,006) is 1.35 times (135%) stronger than the fastener ultimate tensile strength.

Comparison of Results
For purposes of comparison, it would be useful to restate the external thread strength calculation results which were developed in the body of this report.

CASE A: Maximum Material Conditions for both Internal and External Threads

Pitch Diameter = 0.5644" (max) $D1_{MAX}$ = 0.5270" (min)

A_{SS} = 3.14159 * 11 * .625 * .527 * [1/(2*11) + .57735(.5644 - .5270)]
= 0.7632 square inches

Thread Ultimate Shear Strength =

0.5 * 0.7632 * 125,000 = 47,700 pounds (min)

CASE B: Minimum Material Conditions for both Internal and External Threads

Pitch Diameter = 0.5589" (min) $D1_{MAX}$ = 0.5460" (max)

A_{SS} = 3.14159 * 11 * .625 * .546 * [1/(2*11) + .57735(.5589 - .5460)]
= 0.6239 square inches

Thread Ultimate Shear Strength =

0.5 * 0.6239 * 125,000 = 38,992 pounds (min)

CASE C: Out-of-Tolerance Conditions from "Worst Case" Data

Pitch Diameter = 0.5554" (actual) $D1_{MAX}$ = 0.5460" (max)

A_{SS} = 3.14159 * 11 * .625 * .546 * [1/(2*11) + .57735(.5554 - .5460)]
= 0.6000 square inches

Thread Ultimate Shear Strength =

0.5 * 0.6000 * 125,000 = 37,502 pounds (min)

Addition of the Combined-Case evaluation to these results would provide a Case D for comparison:

CASE D: Combined-Case Internal and External Thread Out-of-Tolerance (Hypothetical)

Pitch Diameter = <u>0.5554"</u> (Actual) D_{IMAX} = <u>0.5556</u> (Adjusted)

A_{SS} = 3.14159 * 11 * 0.625 * 0.5556 [1/(2*11) + .57735(0.5554 - 0.5556)]

A_{SS} = 0.5441 in^2

Thread Ultimate Shear Strength =

0.5 * 0.5441 * 125,000 = <u>34,006 pounds</u> (min)

Addition of the fastener Tensile Strength evaluation to these results provides a Case E for comparison:

CASE E: Fastener Tensile Strength[18]

Root Area = 0.202 in^2 :	125,000 psi × 0.202 in^2	=	<u>25,250</u> pounds (min)
Tensile Area = 0.226 in^2 :	125,000 psi × 0.226 in^2	=	<u>28,250</u> pounds (min)

Addition of the maximum Section III allowable preloads for Class 1, 2 and 3 applications provides Cases F and G for comparison:

CASE F: Maximum Preload for Section III,[18] Class 1 Applications

ASME Section III Class 1:

Root Area	2 * 35,000 psi * 0.202 in^2	= <u>14,140</u> pounds
Tensile Area	2 * 35,000 psi * 0.226 in^2	= <u>15,820</u> pounds

CASE G: Maximum Preload for Section III,[18] Class 2 and 3 Applications

ASME Section III Class 2 and 3:

Root Area	2 * 25,000 psi * 0.202 in^2	= <u>10,100</u> pounds
Tensile Area	2 * 25,000 psi * 0.226 in^2	= <u>11,300</u> pounds

[18] Strength based on Root Area and Tensile Area are shown here for comparison. Root areas are used for ASME analysis at SONGS.

FASTENER STRENGTH CALCULATION SUMMARY

CONDITIONS	STRENGTH
Thread Strength:	
Case A, Maximum Thread Material Contact Area (Int and Ext)	47,700 pounds
Case B, Minimum Thread Material Contact Area (Int and Ext)	38,992 pounds
Case C, Worst Case Thread Out-of-Tolerance (Ext)	37,502 pounds
Case D, Combined-Case Thread Out-of-Tolerance (Int and Ext)	34,006 pounds
Tensile Loads:	
Case E, Ultimate Allthread Tensile Strength (Root Area)	25,250 pounds
Case F, Maximum Section III Preload Class 1 (Root Area)	14,140 pounds
Case G, Maximum Section III Preload Class 2 and 3 (Root Area)	10,100 pounds

STRENGTH ANALYSIS RESULTS

The Combined-Case thread strength analysis for both internal and external thread out-of-tolerance conditions reveals that ultimate thread strength of the bolted connection is reduced 7.33% more than the case where only the external fastener was considered to be out-of-tolerance (Case C). This results in the Combined-Case evaluation indicated that the ultimate thread strength of the bolted connection is reduced a total of 10.45% from the Minimum Material condition (Case B).

When the thread strength of the Combined-Case bolted connection is compared to the maximum allowable preload for the joint specified per ASME Section III Class 1 service (14,140 pounds),[19] the ultimate thread strength of the Combined-Case connection is found to be 2.40 times (240%) stronger. When compared to the maximum allowable preload for ASME Section III, Class 2 and 3 service (10,100 pounds) the ultimate thread strength of the Combined-Case connection is found to be 3.37 times (337%) stronger. This clearly demonstrates the significant strength margin inherent in ASME Code fastener design.

[19] It should be noted that for purposes of comparison in this report, that the fastener tensile area (0.226 in^2) was previously utilized in obtaining this value to produce a conservative comparison of thread tensile load to thread strength.

SUMMARY

This evaluation has conservatively calculated the effect on fastener strength when an externally threaded fastener, which has been shown to be out-of-tolerance with respect to pitch diameter, is combined with an internally threaded fastener which is also out-of-tolerance with respect to pitch diameter. The values for pitch diameter utilized in this evaluation corresponded with the maximum observed deviations from specified values reported in warehouse sample inspections associated with this White Paper.

Comparison of the reductions in thread strength to ASME Code design requirements were made and it was demonstrated that the combined-case thread shear strength reductions were on the order of 10.45%, with the Code strength margin being on the order of 200 to 300%. The shear stresses across the combined-case external thread shear area which would result from the maximum allowable preloads were compared with ASME Section III allowables and were verified to be less than the Code maximums for Class 1, 2 and 3 service. This evaluation clearly demonstrates the significant strength margin inherent in ASME Code fasteners. In addition to the strength margin of the individual fastener, installation configurations would require multiple fastener failures prior to joint failure. The nature of the design of fastener preload requires the maximum fastener load to occur at installation, thereby identifying fasteners which may fail at the time of installation instead of in-service. If a fastener should loosen due to loss of preload, leakage would occur before joint failure. This paper discusses the various leakage monitoring and corrective action systems in place at SONGS, including an effective Root Cause evaluation program.

It is therefore concluded that an unlikely combination of internal and external fasteners demonstrating out-of-tolerance conditions proportionally equal to the maximum observed deviations noted in this paper would have no impact on the bolted connections ability to withstand design loads and fulfill a safety function. The results of this evaluation does not change conclusions stated in this White Paper that plant nuclear safety is not compromised by current industry fastener inspection practices.

The following discussion is provided in response to specific comments:

OFFSET THREADS

A condition where the externally threaded fastener may not concentrically penetrate the internal fastener may result from circumstances where the contact faces of the nut or bolt head are not perpendicular, or where the fasteners may be bent. In this type of case the self-centering nature of the inclined thread faces across the thread contact surface area respond to loading of the fastener by inducing bending moments across the tensile cross-sections of the fastener. Stresses associated with these bending moments have been shown to affect fastener fatigue life, relaxation - loss of preload, and preload control.

Issues associated with this type of condition do not routinely include thread profile or pitch diameter compliance, but rather center on flange face perpendicularity, nut and bolt head perpendicularity, and fastener straightness. For Safety-Related applications, these conditions are considered during the Procurement Engineering and Receipt Inspection processes at SONGS.

ATTACHMENT 1

LICENSING POSITION
FASTENER FINAL DIMENSIONAL ACCEPTANCE INSPECTION

REGULATORY ANALYSIS

A. Introduction

The purpose of this presentation is to provide a regulatory analysis regarding the use of fasteners at the San Onofre Nuclear Generating Station (SONGS).

When threaded fasteners are manufactured, there are numerous properties (size, diameter, thread pitch, helix angle, etc.) of thread form which, depending on the application, may or may not be important. A listing of all possible properties is contained in ANSI B1.1, "Unified Inch Screw Threads".

Once it has been determined which properties are important from a design engineering standpoint, selection of a gaging system is made by the design engineer to ascertain thread form acceptability. ANSI B1.3, "Screw Thread Gaging Systems for Dimensional Acceptability", lists four gaging methods [System 21, System 21A, System 22, and System 23]. Each of the methods evaluates certain screw thread characteristics.

ANSI B1.3, Screw Thread Gaging Systems for Dimensional Acceptability - Inch and Metric Screw Threads, states in part:

> *"4(b) The difference between gaging systems is the level of inspection deemed necessary to satisfy that dimensional conformance has been achieved. The following gaging systems describe four accountable levels of dimensional inspection..."*
>
> *"4(b)(1) System 21. Provides of interchangeable assembly with functional size control at the maximum material limits within the length of standard gaging elements, and also control of the characteristics identified a NOT GO functional diameters."*
>
> *"6(d) Relationship of Gaging Systems to Product Screw Thread Acceptability. (1) Product screw threads acceptable to System 23 are acceptable where System 22 and 21 are specified. The reverse is not necessarily true.*
> *(2) Product screws acceptable to System 22 are acceptable where System 21 is specified. The reverse is not necessarily true."*

B. Current Licensing Basis for Fasteners

This section discusses the legal, binding requirements imposed upon SONGS by the U.S. Nuclear Regulatory Commission (NRC). There are two aspects to the current licensing basis for fasteners: (1) NRC regulations do reference/link ASME and ANSI standards which provide, by fastener material type and application, what receipt acceptance method is acceptable; and (2) NRC regulations do reference/link ASME and ANSI standards which state that it is up to the design engineer to determine what, of the many individual thread form parameters which can be evaluated, should be evaluated for receipt acceptance.

1. Fastener Requirements - CODE CASES

SONGS was issued an Operating License by the NRC in 1982. License Condition 2.C states:

> *"This license shall be deemed to contain and is subject to the conditions specified in the Commission's regulations set forth in 10 CFR Chapter I..."*

10 CFR 50.55a, Codes and standards, states in part:

> *"Each operating license for a boiling or pressurized water-cooled nuclear power facility is subject to the conditions in paragraphs (f) and (g) of this section and each construction permit for a utilization facility is subject to the following conditions in addition to those specified in Sec. 50.55..."*
>
> *"... (a)(2) Systems and components of boiling and pressurized water-cooled nuclear power reactors must meet the requirements of the ASME Boiler and Pressure Vessel Code specified in paragraphs (b), (c), (d), (e), (f), and (g) of this section..."*
>
> *"...(c) Reactor coolant pressure boundary. (1) Components which are part of the reactor coolant pressure boundary must meet the requirements for Class 1 components in Section III of the ASME Boiler and Pressure Vessel Code, except as provided in paragraphs (c)(2), (c)(3), and (c)(4) of this section..."*
>
> *"...(d) Quality Group B components. (1) For a nuclear power plant whose application for a construction permit is docketed after May 14, 1984 components classified Quality Group B must meet the requirements for Class 2 Components in Section III of the ASME Boiler and Pressure Vessel Code..."*

> *"...(e) Quality Group C components. (1) For a nuclear power plant whose application for a construction permit is docketed after May 14, 1984 components classified Quality Group C must meet the requirements for Class 3 components in Section III of the ASME Boiler and Pressure Vessel Code.*

Section III of the ASME Boiler and Pressure Vessel Code, Subpart NA-1220, Materials, states in part:

> *"Materials are manufactured to an SA, SB, or SFA specification or any other material specification permitted by this section. Such material shall be manufactured and certified in accordance with the requirements of this Section..."*

There are two categories of fasteners: fasteners manufactured to the A, B, or F specification; and fasteners manufactured to unique specifications which are produced on a case-by-case application. Materials manufactured to an A, B, or F specification have specific characteristics depending on the type and application of the fastener.

There are specific individual ASME A, B, F standards for each. The attached Table 1 delineates the fastener type and controlling ASME standard number.

Within each ASME standard, is a specific reference to a controlling ANSI standard which specifies thread form. Table 1 also contains a column for each ASME standard, which has its corresponding ANSI standard.

Table 1- Thread Acceptance Criteria
Associated With ASTM Fastener Material Specifications

ASTM Material Type	ASTM Number	ANSI Standard	ANSI Thread Gaging Acceptability Requirements
Alloy Steel and Stainless Steel Bolting Materials for High-Temperature Service	A 193	18.2.1	System 21
Carbon and Alloy Steel Nuts for Bolts for High-Pressure and High-Temperature Service	A 194	18.2.2	System 21
Carbon Steel Bolts and Studs, 60,000 psi Tensile Strength	A 307	18.2.1	System 21
Alloy Steel Bolting Materials for Low-Temperature Service	A 320	18.2.1	System 21
High-Strength Bolts for Structural Steel Joints	A 325	18.2.1	System 21
Quenched and Tempered Alloy Steel Bolts, Studs, and Other Externally Threaded Fasteners	A 354	18.2.1	System 21
Alloy Steel Turbine-Type Bolting Material Specially Heat Treated for High-Temperature Service	A 437	18.2.1	System 21
Quenched and Tempered Steel Bolts and Studs	A 449	18.2.1	System 21
Bolting Materials, High Temperature, 50 to 120 Ksi [345 to 827 MPa] Yield Strength, with Expansion Coefficients Comparable to Austenitic Steels	A 453	18.2.1	System 21
Heat-Treated Steel Structural Bolts, 150 Ksi Minimum Tensile Strength	A 490	18.2.1	System 21
Carbon and Alloy Steel Nuts	A 563	18.2.2	System 21

ASTM Material Type - Continued	ASTM Number	ANSI Standard	ANSI Thread Gaging Acceptability Requirements
Alloy Steel Socket-Head Cap Screws	A 574	18.3	System 22
High-Strength Nonheaded Steel Bolts and Studs	A 687	N/A	Not Addressed
Nonferrous Nuts for General Use	F 467	18.2.2	System 21
Nonferrous Bolts, Hex Cap Screws, and Studs for General Use	F 468	18.2.1	System 21
Carbon and Alloy Steel Externally Threaded Metric Fasteners	F 568	18.2.3	Not Addressed
Stainless Steel Bolts, Hex Cap Screws, and Studs	F 593	18.2.1	System 21
Stainless Steel Nuts	F 594	18.2.2	System 21
Alloy Steel Socket Button and Flat Countersunk Head Cap Screws	F 835	18.3	**System 22**
Stainless Steel Socket Head Cap Screws	F 837	18.3	**System 22**
Stainless Steel Socket Button and Flat Countersunk Head Cap Screws	F 879	18.3	**System 22**
Stainless Steel Socket-Set Screws	F 880	18.3	**System 22**
Alloy Steel Socket Set Screws	F 912	18.3	**System 22**

Within each ANSI standard is a specific reference to the gaging system which is acceptable for use in determining thread acceptance. Table 1 also contains a column for each ANSI standard, which has its corresponding gaging system specified.

Of the 23 categories, 15 categories specify System 21, 6 specify System 22, and 2 are not addressed.

An example of this would Specification SA-193, Standard Specification for Alloy Steel and Stainless Steel Bolting Materials for High Temperature Service [note: SA-193 is for RCS bolting applications, similar cross references to ANSI B18.2.1 and ANSI B1.1 exist for other bolting materials and applications], which states in part:

> *"11.1 All bolts, studs, stud bolts, and accompanying nuts, unless otherwise specified in the purchase order shall be threaded in accordance with the American National Standard for Screw Threads (ANSI B1.1), Class 2A fit, ..."*
>
> *"13.3...Unless otherwise specified in the purchase order, the Heavy Hex Screws Series should be used... for sizes not covered in the Heavy Hex Screws Series in ANSI B18.2.1..."*

ANSI B18.2.1, "Square and Hex Bolts and Screws - Inch Series", states in the notes to Tables on Threads:

> ***"Acceptability of screw threads shall be determined based on System 21, ANSI B1.3 Screw Thread Gaging Systems for Dimensional Acceptability."***

2. Fastener Requirements - ENGINEERING

ANSI B1.3 states in part:

"5.a Screw threads of threaded products are defined by the applicable thread document... 5(b) the gaging system used to inspect the screw thread of a threaded product shall be as specified in the product standard, procurement drawing, or purchase inquire."

The ANSI B1.1, Sections 5(a and b), authorize product standards and purchase documents, which are created by the design engineer, to define which thread characteristics are important and select the appropriate gaging system to ascertain thread conformance. If the thread characteristics are undefined, then the program defaults to System 21.

The American Society of Mechanical Engineers, in a letter from Mr. Kurt Wessely, Director, ASME Codes and Standards, to Senator Joseph Lieberman, dated June 10, 1994, states in part:

> *"As the result of a hearing before the Senate Subcommittee on Antitrust and Monopolies, ASME agreed to publish an array of gaging systems and describe the attributes of each system. From this array of gaging systems, it is intended that the engineer or related scientist who is designing, fabricating or inspecting equipment will select the system that addresses the need. The selection of the gaging system is intended to be made by the user of the threaded product based on the intended application of the threaded fastener. The ASME Standard does not recommend one gaging system over the others."*

Accordingly, it is recognized that design engineers do <u>not</u> have to specify thread conformance and thread acceptance via gaging system <u>to every dimensional limit</u> listed in ASME B1.1. It may be appropriate for some gaging to be made by System 22 (as indicated in Table 1), however, for the majority of threaded fasteners, System 21 is the acceptable method.

NEDO has endorsed the practice of using system 21 as the method of choice.

C. NRC POSITION

References:

1. U. S. Nuclear Regulatory Commission, NUREG-1349, "Compilation of Fastener Testing Data Received in Response to NRC Compliance Bulletin 87-02, June 1989

On November 6, 1987, the NRC issued Bulletin 87-02, "Fastener Testing to Determine Conformance with Applicable Material Specifications." The bulletin was issued so that the NRC staff could gather data to determine whether fasteners obtained from suppliers and/or manufacturers meet the mechanical and chemical specifications stipulated in the procurement documents. Based on the results, the NRC concluded that nonconforming fasteners do not seem to represent a significant safety hazard to the nuclear industry (Reference 1).

References:

2. Letter from Stanley P. Johnson to Ivan Selin (NRC), dated March 8, 1994.
3. Letter from William T. Russell (NRC) to Stanley P. Johnson, dated March 25, 1994.

In response to the Johnson Gage Company's initial alleged concerns outlined in Reference 2, the NRC indicated (Reference 3) that "the NRC staff does not consider System 21 or the use of go-no-go gauges to be inappropriate ("flawed") for accepting certain fastener threads...." The NRC further stated "that, although System 22 may be an improvement over System 21, there is not sufficient basis to make its use a requirement for NRC licensees." In summary, the NRC noted that "the NRC staff has not found evidence that failures due to dimensionally nonconforming fasteners are occurring and therefore, does not consider it to be a safety concern."

References:

4. NRC Memorandum from Brian W. Sharon to Ashok C.Thadani, "Meeting with NIST Regarding Gauging of Threaded Fasteners," dated May 5, 1994.

5. Letter from Richard Jackson (NIST) to
James A. Davis (NRC), dated March 10, 1994.

A meeting (Reference 4) between members of the NRC staff and the National Institute of Standards and Technology (NIST) was held on May 4, 1994, to obtain clarification of letters NIST has written over the years (e.g., Reference 5) regarding the unacceptability of System 21 for ensuring threaded fasteners meet the tolerance specifications in ANSI B1.1. NIST had gone on record to state:

> "System 21 (plug and ring) acceptance methods do not assure dimensional conformance with material limits specified in ASME B1.1...."

The senior official from NIST at the meeting (Richard Jackson) was asked why their letters only stated that use of System 21 would not ensure compliance with the dimensional tolerances of ANSI B1.1, but were silent on the fact that neither would System 22 ensure compliance with all of the dimensional tolerances in the ANSI standard. NIST was also asked what the purpose of was for making such a statement, since NIST did not imply or state that this meant the System 21 was considered unacceptable or would result in fastener failures. In response, Mr. Jackson agreed that NIST would write a letter (Reference 6) to the NRC clarifying their position that failure of threaded fasteners to meet the dimensional tolerance of ANSI B1.1 does not necessarily imply that an unsafe condition will result from their use. They also agreed to state that the acceptability of the gauging system used to accept threaded fasteners is the responsibility of the user of the fasteners.

Mr. Sharon (NRR) concluded that the Division of Engineering was preparing a technical report that will document their assessment of this issue. The analysis will demonstrate why fasteners that meet System 21 but do not meet System 22 gauging tolerances are considered acceptable by the staff from a structural standpoint, that operational data does not support Mr. Johnson's implication that fastener failures pose a threat to Nuclear Power safety, that risk assessments show that the risk of core melt from threaded fastener failures is extremely low; and finally, that redundancy and ample structural safety margins in the design of commercial nuclear plants do not result in situations in which single fastener performance is critical.

D. NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY (NIST) POSITION

References:

6. Letter from Richard Jackson (NIST) to
   Brian W. Sharon (NRC), dated May 19, 1994.

7. Draft Handbook, "Fasteners and Metals,"
   Stiefel, S. W., U. S. Department of Commerce, NIST, dated
   February 1993

As indicated above, NIST had gone on record to state:

> "System 21 (plug and ring) acceptance methods do not assure dimensional conformance with material limits specified in ASME B1.1...."

In response to the NRC's request for clarification, as indicated above, NIST responded (Reference 6) by stating that "we [NIST] are not similarly able to make any definitive statements [assurance of dimensional conformance] about Systems 22 and 23. Unfortunately, there is not enough data to support such conclusion. ... Thus, in matters of product performance, we defer to industry standards committees and government regulatory agencies, and view the responsibility for any safety issues associated with choice of fastener or gauging method to rest solely with the user."

To assist in complying with Public Law 101-592, "Fastener Quality Act," the Department of Commerce has drafted a handbook for establishing an laboratory accreditation program (Reference 7). This publication was drafted by the NIST, but appears to contradict their statements above in that this handbook endorses System 21. Specifically, under the Section titled "Gaging requirements," NIST states that "The System 21 gaging system and its derivatives allow the test laboratory to choose among specified types of screw thread gages and measurement equipment for use in determining the required thread characteristics."

E. AMERICAN SOCIETY OF MECHANICAL ENGINEERS POSITION

References:

8. Letter from William T. Russell (NRC) to
   Walter R. Mikesell (ASME), dated June 1, 1994.
9. Letter from Stanley P. Johnson to Ivan Selin
   (NRC), dated April 12, 1994.

In a letter (Reference 8) to the ASME Chairman in regards to additional concerns submitted to the NRC Chairman by Mr. Johnson (Reference 9), the NRC staff indicated that they "concluded an initial review of this issue and has determined that there is no immediate safety concern. However, the NRC staff believes that there is an inconsistency in the existing standards in that the required gaging systems cannot assure that all the tolerances will be satisfied." In order for the NRC to address Mr. Johnson's concerns, the NRC requested ASME to address 1) what is the significance of the tolerances given in the ANSI B1.1 tables when gaging systems specified do not guarantee conformance, and 2) the safety significance with fasteners failure to meet ANSI B1.1 tolerances but are found to be acceptable by the various gaging systems specified in ASME B1.2.

References:

10. Letter from M. R. Green (ASME) to
    William T. Russell (NRC), dated June 10, 1994.

11. Letter from Kurt Wessely (ASME) to
    Senator Joseph I. Lieberman (U. S. Senate), dated June 10, 1994.

12. Letter from Senator Joseph I. Lieberman to
    Kurt Wessely (ASME), dated May 20, 1994.

ASME responded (Reference 10) to the NRC's request above by forwarding a copy of their response (Reference 11) to similar concerns raised by Senator Lieberman (Reference 12). ASME concluded that "The ASME Standard does not recommend one gaging system over the others. Qualified engineers can be relied upon to make the proper selection of gaging systems based upon application. ... ASME has no information that compliance with any ASME B1 Screw Thread Standard has caused any unsafe condition".

F. INDUSTRIAL FASTENERS INSTITUTE POSITION

Reference:

13. Recommendations for Fastener Thread Acceptability, Industrial Fasteners Institute (IFI).

IFI recommendations for fastener thread inspections (Reference 13) conclude that the most practical thread measuring systems currently in existence which should be used are ANSI/ASME B1.3M-1986 System 21 (Method A) for all internal and external threads, except Class 3A external threads, where System 22 (Method B) is applicable. This position is interpreted to be in basis agreement with FED-STD-H28/20A September, 1987, "Inspection Methods for Acceptability of UN, UNR, UNJ, M and MJ Screw Threads."

ADDENDUM TO ATTACHMENT 1

ASME CODE INTERPRETATION OF FASTENER INSPECTION REQUIREMENTS

Question (1): Tables NB/NC/ND/NE-3132-1 identify standards B18.2.1, B18.2.2, and B18.3 for bolting and B1.1 for threads. ANSI standards B18.2.1, B18.2.2, and B18.3 also identify dimensions and tolerances for threads. B18.2.1 and B18.2.2 provide for the use of System 21 (ASME B1.3M) gaging and B18.3 provides for the use of System 22 (ASME B1.3M). Does Section III require that the dimensions and tolerances of B18.2.1, B18.2.2, and B18.3, and System 22 gaging, be used for threads on bolting produced to the requirements of ASME material specifications?

Reply (1): No. Section III requires that the provisions of the ASME bolting material specifications be met, for thread dimensions, tolerances, and gaging systems. Section III provides for the use of these material specifications with System 21 gaging or other systems designated by the purchaser.

Question (2): Does Section III require the use of B18.3 and the associated tolerances for bolting other than socket cap, shoulder, and set screws?

Reply (2): No.

Question (3): Does Section III permit use of System 21 Go/Not-Go gaging for acceptance of threads used in Class 1, 2, 3, and MC components?

Reply (3): Yes.

Question (4): Is it a requirement of Section III that thread dimensions be measured to verify they are within the tolerances specified in ASME B1.1?

Reply (4): No.

Question (5): Does Section III accept the use of thread gages for bolting, such as Go/Not-Go gages, that do not verify compliance of all dimensions and tolerances given in ASME B1.1?

Reply (5): Yes.

Question (6): Does use of System 21 Go/Not-Go gages for Section III bolting ensure the dimensional conformance to the ASME bolting standards identified in Tables NB/NC/ND/NE-3132-1 to the degree necessary to be consistent with the design philosophy and criteria of Section III?

Reply (6): Yes.

Codes and Standards

The American Society of Mechanical Engineers

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October 4, 1994

Mr. Keith R. Wichman
07-D4
U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

Subject: Section III, Division 1, Table NB/NC/ND/NE-3132-1; Design - Thread Gaging Requirements

File: NI94-10

Dear Mr. Wichman:

Our understanding of the question in your letter of September 8, 1994 and our reply are as follows:

Question (1): Tables NB/NC/ND/NE-3132-1 ider standards B18.2.1, B18.2.2 and B18.3 for bolting and B1.1 for threads. ANSI standards .1, B18.2.2, and B18.3 also identify dimensions and tolerances for threads. B18.2.1 and . 2.2 provide for the use of System 21 (ASME B1.3M) gaging and B18.3 provides for the use of System 22 (ASME B1.3M). Does Section III require that the dimensions and tolerances of B18.2.1, B18.2.2, and B18.3, and System 22 gaging, be used for threads on bolting produced to the requirements of ASME material specifications?

Reply (1): No. Section III requires that the provisions of the ASME bolting material specifications be met, for thread dimensions, tolerances, and gaging systems. Section III provides for the use of these material specifications with System 21 gaging or other systems designated by the purchaser.

Question (2): Does Section III require the use of B18.3 and the associated tolerances for bolting other than socket cap, shoulder, and set screws?

Reply (2): No.

Question (3): Does Section III permit use of System 21 Go/Not-Go gaging for acceptance of threads used in Class 1, 2, 3, and MC components?

Reply (3): Yes.

Question (4): Is it a requirement of Section III that thread dimensions be measured to verify they are within the tolerances specified in ASME B1.1?

Reply (4): No.

Question (5): Does Section III accept the use of thread gages for bolting such as Go/Not-Go gages that do not verify compliance of all dimensions and tolerances given in ASME B1.1?

Reply (5): Yes.

Question (6): Does the use of System 21 Go/Not-Go gages for Section III bolting assure the dimensional conformance to the ASME bolting standards identified in Tables NB/NC/ND/NE-3132-1 to the degree necessary to be consistent with the design philosophy and criteria of Section III?

Reply (6): Yes.

Regards,

DSPoulis

D. S. Poulis
Secretary, Committee on
Section III Interpretations

ATTACHMENT 2

STATISTICAL VALIDATION OF SCE FASTENER STRENGTH ANALYSIS WHITE PAPER DATABASE

BY
TETRA ENGINEERING GROUP
August 5, 1994

Statistical Validation of SCE Fastener Strength Analysis White Paper Data Base

Tetra Engineering Group, Inc.
Report 94-SCE-003
August 5, 1994

Authors

Dr. Frank J. Berte'
Dr. Peter S. Jackson, PE
Mr. David S. Moelling, PE

EXECUTIVE SUMMARY

The White Paper ("A White Paper, Fastener Strength Analysis, Nuclear Safety Concern 93-11" Reference 1) was reviewed by Tetra Engineering Group to with regard to statistical and sampling concerns. The White Paper was written to demonstrate that there is a high degree of confidence that fasteners now inservice in the San Onofre Nuclear Generating Station (SONGS) and in the SONGS Warehouse possess adequate margin with respect to ASME code and design requirements. This was addressed in the White Paper by selecting a sample of warehouse stock fasteners, inspecting them with Johnson Indicating gages to provide accurate measurements of thread pitch diameter, and assessing the impact of the worst observed deviations on fastener strength.

Several concerns were raised with regard to the sampling and analysis of warehouse stock. We have restated those concerns as follows;

1. Was the sampling procedure used to select items from warehouse stock structured to allow valid statistical conclusions regarding the probability/confidence level of "worst case" deviations in pitch diameter? (i.e. was it a "random" sample)

2. Was the stock in the warehouse procured and inspected in such a way that the current warehouse stock is representative of fasteners taken from stock? (i.e. does it appear that accepted fasteners are produced and accepted to a consistent quality level?)

3. Based upon the samples drawn, can it be shown that the fastener stock has adequate strength margin at a high probability/confidence level?

We examined the White Paper sampling procedure, the data obtained as well as the past year's incoming receipt inspection data to address these questions.

With regard to Question 1, the sampling procedure used was essentially a systematic sampling plan in nature. Provided the ordering of samples by material code was random with respect to dimensional characteristics, systematic sampling is essentially equivalent to a random sampling. Since there is some order to the assignment of material codes by alloy, size, thread pitch etc., it is not possible to positively determine randomness in sample order. The sampling procedure for individual lots was properly employed to draw random samples. The statement in the White Paper that the use of an attributes based lot sampling plan with a 95% confidence limit implies the same confidence level in the limiting thread measurement is not correct. That said, a comparison of the White Paper data with the recent receipt inspection data leads us to believe that conclusions drawn from the White Paper Data would be consistent with data drawn using a truly random sample. We recommend that a limited re-sample be made to confirm this conclusion.

With regard to Question 2, the data indicates that the fasteners procured to SYSTEM 21 have been manufactured so that the average thread dimensions (in this case pitch diameter) are maintained within or close to ANSI B1.1 tolerances. While individual items may exceed ANSI B1.1 tolerances, over a number of items the average tends to lie within or close to the tolerance bands for external thread items. For internal thread items (nuts) it appears that the suppliers have allowed the production lots to trend slightly in direction of greater assembability, and a greater number are somewhat outside of the ANSI B1.1 tolerances. In all cases production tolerances appear to be in statistical control with no observable trends with regard to item type or size. From this we conclude that the dimensional quality of fasteners has been consistent at least over the period covered by the supplied data (1992-1994). Thus fasteners drawn from warehouse stock and installed in the plant should have the same dimensional characteristics as the warehouse stock.

With regard to Question 3, we conclude that the White Paper data would be similar to data obtained from a truly random sample. Based on this conclusion we determined the limiting out-of-tolerance condition for internal and for external thread items. These conditions were determined such that for an item drawn at random from any Material Code bin in the warehouse there would be a 95% probability that the item's Pitch Dimension would be more conservative that the limiting condition. The distributions of dimensional deviations obtained from the data were used to confirm that the strength margin calculations in the White Paper were consistent with the data. The average fastener has negligible difference from the nominal fastener strength, and the "limiting worst case" fasteners have only minor loss of strength margin.

INTRODUCTION

The White Paper ("A White Paper, Fastener Strength Analysis, Nuclear Safety Concern 93-11" Reference 1) was reviewed by Tetra Engineering Group to with regard to statistical and sampling concerns. The White Paper was written to demonstrate that there is a high degree of confidence that fasteners now inservice in the San Onofre Nuclear Generating Station (SONGS) and in the SONGS Warehouse possess adequate margin with respect to ASME code and design requirements. This was addressed in the White Paper by selecting a sample of warehouse stock fasteners, inspecting them with Johnson Indicating gages to provide accurate measurements of thread pitch diameter, and assessing the impact of the worst observed deviations on fastener strength.

Several concerns were raised with regard to the sampling and analysis of warehouse stock. We have restated those concerns as follows;

1. Was the sampling procedure used to select items from warehouse stock structured to allow valid statistical conclusions regarding the probability/confidence level of "worst case" deviations in pitch diameter? (i.e. was it a "random" sample)

2.Was the stock in the warehouse procured and inspected in such a way that the current warehouse stock is representative of fasteners taken from stock.? (i.e. does it appear that accepted fasteners are produced and accepted to a consistent quality level?)

3. Based upon the samples drawn, can it be shown that the fastener stock has adequate strength margin at a high probability/confidence level?

We examined the White Paper sampling procedure, the data obtained as well as the past year's incoming receipt inspection data to address these questions.

REVIEW OF WHITE PAPER SAMPLING PLAN

The White Paper sampling plan can be summarized as follows:

1. All safety-related fastener Material Codes (for sampling purposes these constitute lots) were identified.

2. Material Codes not inspected to SYSTEM 21 or SYSTEM 22 (Ref. 2) were deleted as were all items with diameter less than 0.5".

3. Material Codes previously inspected with indicating gages were also deleted from the population.

4. The remaining Material Codes (286) were ordered by general item type (bolt, nut, all-thread, etc.)

5. Every ninth Material Code was selected from the list for sample selection. This resulted in 32 Material Codes (lots) with at least one of each general item type.

6. From each of these 32 lots, samples were drawn using the sampling procedure of Reference 3. This is an attributes based lot acceptance sampling plan that imposes random selection of samples. The number of samples ranges from 4 to 32 and depends on the lot size, in this case the number of items in the warehouse bin. This resulted in 356 items drawn.

7. These item were inspected by indicating gages to determine the thread pitch diameter (PD) as defined by ANSI/ASME B1.1 (Ref. 7). Some items (long all thread) were measured in more than one location so the total number of measurements was 425.

The first observation we made with regard to this sampling procedure was that it was developed in an ad-hoc manner using attribute based acceptance sampling plans as guidance. For this type of problem where obtaining an accurate estimate of population characteristics is the goal, a survey sampling plan should have been used. The reason for using a formal, theoretically defined plan is that the precision (confidence) of the results can be computed. We then looked to see if the White Paper sampling procedure could be matched to any standard survey sampling plan.

The WP plan is of the same type as what are called "systematic sampling plans". In a standard systematic sampling plan (Ref. 4), the N units of a population are numbered from 1 to N in some order. To select a sample of n units, a unit is drawn at random from the first k units and then every k units thereafter until n units are drawn. This type is called an *every kth systematic sample*. Systematic sampling is easy to implement, but suffers from the drawback that the precision can vary depending on any underlying structure in the dataset. If the ordering of the data is essentially random, so that the measured statistic is not correlated with the ordering variable, "there will then be no trend or stratification in [the measured statistic] as we proceed along the file and no correlation between neighboring values. In this situation we would expect systematic sampling to be essentially equivalent to simple random sampling and to have the same variance." (Reference 4).

This situation would be true if there were no correlation between material code and the dimensional characteristics of the fasteners. A review of some of the material code listings shows some degree of trend in MC number with item diameter, length, thread pitch, etc. Although an examination of the data (discussed later) does not indicate any correlation of these with pitch diameter deviations, it would require a detailed analysis to prove this definitively. We would recommend a limited random resampling to confirm randomness in the data order.

The second stage of the sampling process uses the procedure of Reference 3. It produces an acceptably random selection of items from the lot. The White Paper (Page 10) makes the following statement with regard to the precision of the sampling plan:

"Following the statistical logic that provides a 95% confidence level to lots/batches found acceptable using this sampling program, it can be shown that the same confidence level can be applied to the band of readings found during this specific warehouse sample. In short, the sampling plan is designed to provide a 95% confidence level that there are no greater out-of-tolerance conditions in the warehouse stock than those identified in this sample."

This statement is not correct. Attributes acceptance plans such as that of Reference 3 are developed using hypergeometric probability computations or the equivalent. Relating the confidence levels to tolerance bounds on specific diameters may or may not be true depending on the details of the plan. As described later, the observed "worst case" out-of-tolerance items in fact are essentially 95%/95% probability/confidence bounds, but this is not due to the design of the sampling procedure as taken from Reference 3.

For the purposes of further statistical analysis, we refined the goal with regard to warehouse stock. A 95% confidence level on the largest out-of-tolerance condition with respect to the entire warehouse stock could be affected by the relative stock numbers of individual items (for example lots of nuts and few large bolts). Since items from any safety-related material code could be installed in the plant, we formulated the following goal:

For any item, drawn from the stock in any Material Code, there will be a 95% probability at a 95% confidence that the deviation of the items Pitch Diameter from nominal will be less than the computed limiting values.

REVIEW OF AVAILABLE DATA

White Paper

Inspection data from the White Paper (Reference 5) was supplied in EXCEL spreadsheet format. Information was supplied for 425 individual items in 32 separate material codes. Five of these material codes are used for both internal thread (nuts) and external threads (all thread) items. These were split into separate groups for the purpose of this analysis. Each item had the following information provided.

- Sequence number from White Paper Inspection
- Material Code (SCE)
- Supplier
- Material Grade
- Internal/External Thread Identification
- Material Specification
- Nominal Pitch Diameter (inches)
- Measured Pitch Diameter (inches)
- Maximum Pitch Diameter from ANSI B1.1 (Reference 2)
- Minimum Pitch Diameter from ANSI B1.1. (Reference 2)
- RSO Number (SCE)
- Description of Item

Recent Inspection Samples

A set of inspection data from new material receipt inspections (Reference 6) was supplied in EXCEL spreadsheet format. This data covered inspections from 7/15/93 to 7/15/94. The data included all inspections for those fastener types covered by the White Paper, but excluding re-inspection or re-stocking inspections. It includes all lots inspected including those rejected by receipt inspections. 2089 Individual Items were inspected from 65 separate material codes. Only one material code was common between this data set and the White Paper sample data set. Each item had the following information provided.

- Sequence number
- Material Code (SCE)
- Supplier Number
- Material Grade
- Internal/External Thread Identification
- Material Specification

- Nominal Pitch Diameter (inches)
- Measured Pitch Diameter (inches)
- Maximum Pitch Diameter from ANSI B1.1 (Reference 2)
- Minimum Pitch Diameter from ANSI B1.1. (Reference 2)
- RSO Number (SCE)
- RSO Revision (SCE)
- Requester
- Test Type
- Test Lab Report Number
- Description of Item
- Purchase Order Number
- Supplier Name

Statistics Used

Strength is one of the primary features important to the end use of the fasteners. Fastener pitch diameter is the dimension which has the primary impact on fastener thread strength. Thus the thread characteristic of interest is the Pitch Diameter (PD). The deviation of actual PD (PD_a) with respect to the nominal PD (PD_n) and the acceptance band about the Nominal PD is the population characteristic of interest. To assess this the following statistic was computed for each item in the two data sets:

$$R\Delta PD = (PD_n - PD_a) / (Max\ PD - Min\ PD)$$

where: Max PD = Maximum Allowed PD from ANSI B1.1

Min PD = Minimum Allowed PD from ANSI B1.1

This represents the deviation of the items PD from nominal PD as a fraction of the allowed tolerance band. Values of R∆PD within -0.5 to +0.5 thus lie within the allowed band. Note that in all plots of this deviation there are three reference lines given:

Nominal = 0 (No deviation from Nominal PD)

Min = 0.5 (Measured PD is at the Minimum Allowed value by ANSI B1.1)

Max = -0.5 (Measured PD is at the Maximum Allowed Value by ANSI B1.1)

CHARACTERISTICS OF DATA

Homogeneity of PD data

To assess any set of data in a statistical method, certain key assumptions of randomness and homogeneity of the data set must be examined. Since both the White Paper data and the Recent Inspection Data represent mixtures of item types and sources, the assumption of homogeneity with respect to RΔPD was examined. This is a prerequisite to determining if the effect of ordering samples by material code in the White Paper Sample resulted in non-random samples.

Internal/External thread

Internal Thread (Nuts) and External Thread (bolts, studs, threaded rods) items are manufactured in different manners. To check for homogeneity of RΔPD between internal and external thread items, histograms were prepared for all internal thread and all external thread items in both data sets and compared.

White Paper Data

Figure 1 shows the Internal Threads from the WP Data Set:

Figure 1

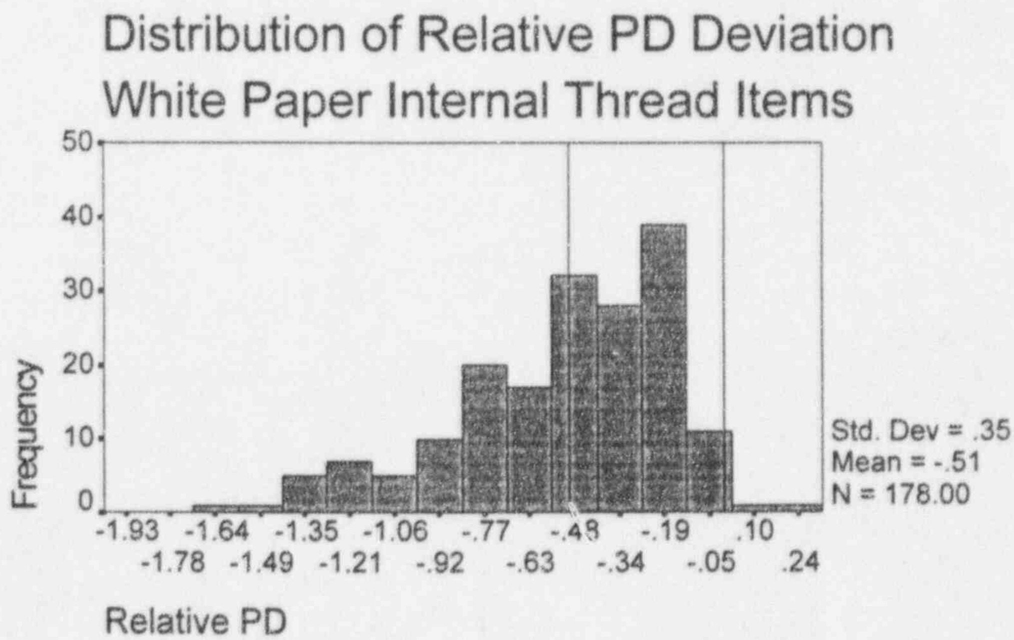
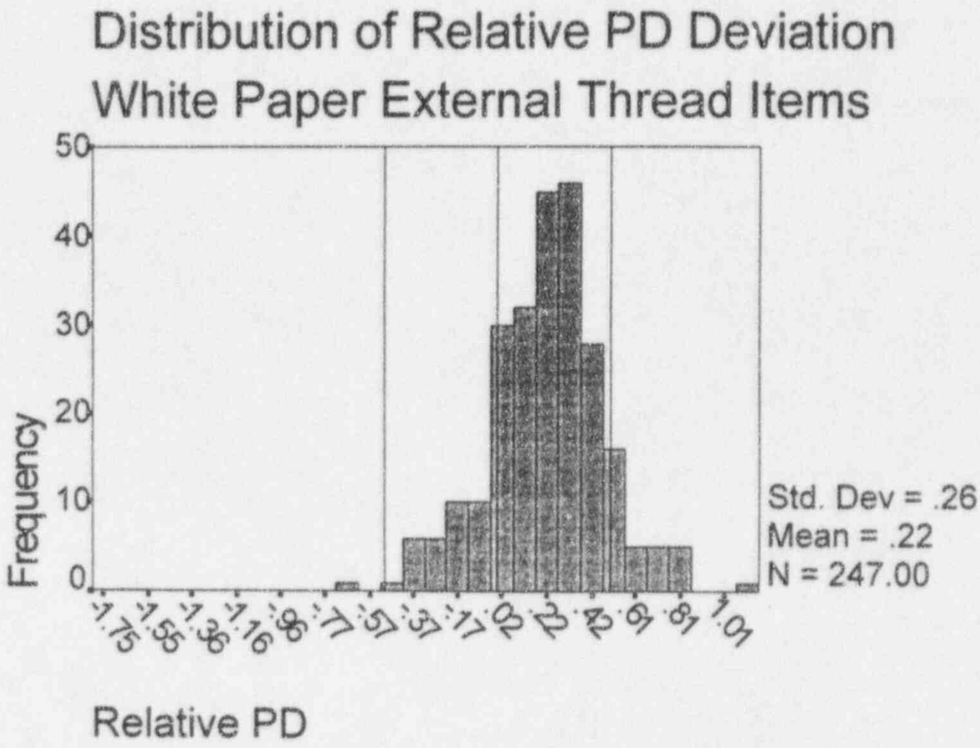


Figure 2 shows the External Threads from the WP Data Set.

Figure 2



These represent a large number of different item types and suppliers but it is clear that the two groups are statistically different. The Internal Thread items tend to have deviations in the direction of larger than nominal pitch diameters and the External thread items tend to have deviations in the direction of smaller than nominal pitch diameters.

Recent Inspection Data

Figure 3 shows the Internal Threads from the Recent Inspection Data.

Figure 3

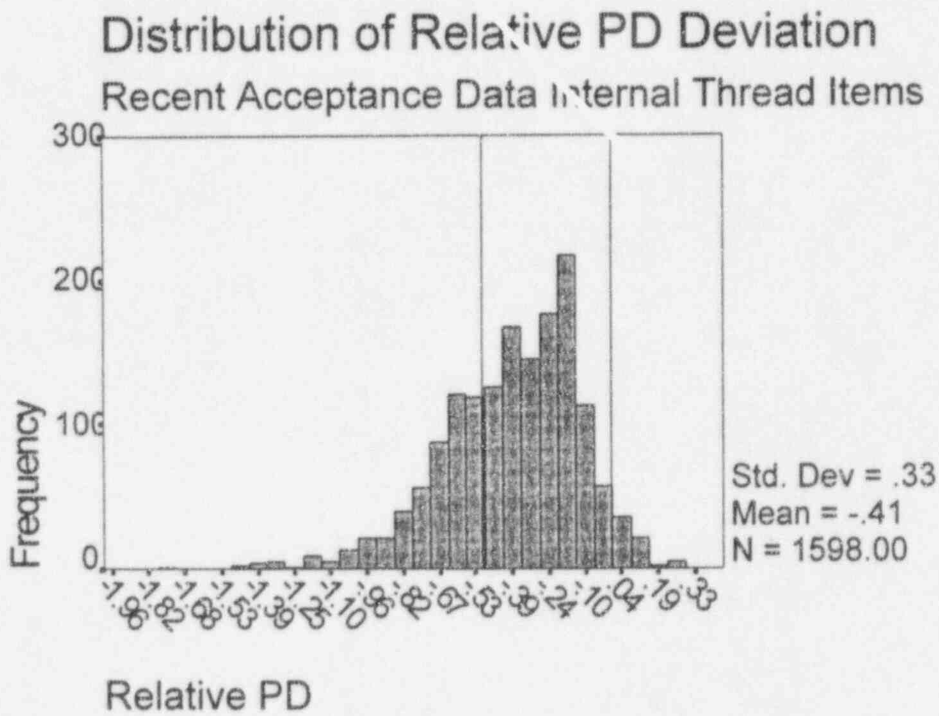
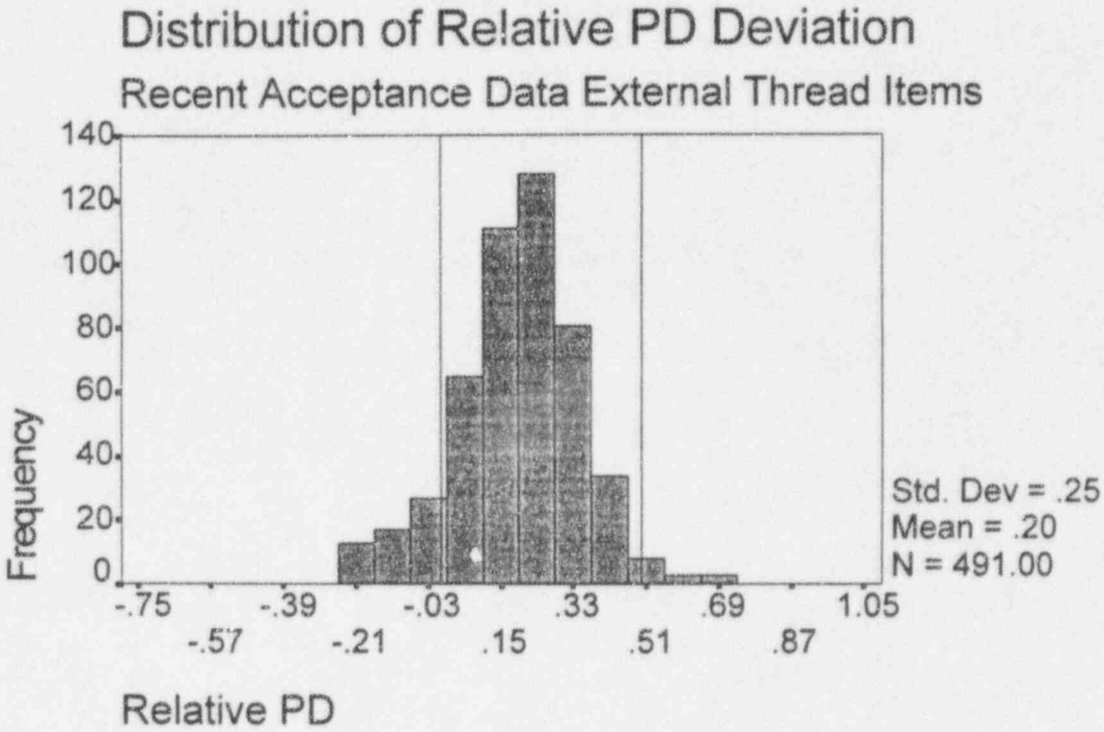


Figure 4 shows the External Threads from the Recent Inspection Data.

Figure 4



The distributions of the Recent Inspection data show similar populations for the Internal and External Thread items as the White Paper data. The presence of larger extreme values in the Recent Inspection data is related to the inclusion of all lots, not just those that passed acceptance tests. Comparison of the overall means and standard deviations for the White Paper and Recent Inspection Results show reasonable consistency. The Table 1 illustrates this:

Table 1

Comparison of White Paper and Recent Receipt Inspection Data

External Thread Summary Statistics - Relative PD (Dimensionless)

Statistic	White Paper	Recent Receipt
Mean PD Deviation	0.22	0.20
Std. Dev. of PD Deviation	0.26	0.25

Internal Thread Summary Statistics - Relative PD (Dimensionless)

Statistic	White Paper	Recent Receipt
Mean PD Deviation	-0.51	-0.41
Std. Dev. of PD Deviation	0.35	0.33

Material Code

The key factor to be examined for homogeneity of PD deviation is the individual material codes. It is important as like items are installed in many plant applications (for example several bolts are installed in a single flange or coupling would be of the same material code). Boxplots were used to rapidly compare the statistics of individual material codes. These are provided in the Appendix. There is significant difference between many material codes in both the external and internal data. The internal thread MC to MC variation appears to be larger than the external thread MC to MC variation.

Supplier

Differences between Suppliers in a systematic way would be a source of significant non-randomness in the samples. Again boxplots were used to provide a rapid comparison. These are supplied in the Appendix. By looking at various material code data from a single supplier we found considerable deviation between material codes. The variability in deviations between material codes is a good indication of a lack of systematic effect of a supplier. The data for a single supplier was also examined for any trend with item size as indicated by nominal PD. There was no obvious systematic trend giving further evidence for randomess in the Material Codes.

The data shows that is no indication of a significant systematic supplier effect and that variations are very likely to be production lot to production lot variations rather than supplier to supplier in nature.

Control of Pitch Diameter

The observed deviations of Pitch Diameter in both the White Paper and the Recent Inspection data indicate that the fastener manufacturers are in general achieving a good state of statistical control. For external thread items is appears that most suppliers are attempting to control the **mean** PD to the ANSI B1.1 tolerances. A similar situation is found with the internal thread items except that a wider range of deviation is observed in the direction of greater assembility. This allows a reasonable expectation that future shipment from these suppliers will show a similar behavior as some measure of statistical control is in place. The variability in any particular item seems to be due to production lot to production lot variations. Figures 9, 10, and 11 show this clearly. Production from a single supplier varies from MC to MC but does not depend on item size for example.

COMPUTATION OF LIMITING OUT-OF-TOLERANCE VALUES

Assuming that the WP data is essentially random in order, limiting out-of-tolerance values can be computed. As is clear from the data, internal and external thread items must be treated separately. It does not appear that item type or dimensions have systematic effect, nor that the item supplier has a strong non-random effect. Clearly there is both a between-lots and a within-lot (material code) effect. This is possibly due to the particular sequence of production lots going into a material code bin at any particular time. Because of the varying sample sizes resulting from the use of the Reference 3 procedure, sample size weighted estimates of these effects must be used.

For the estimation of the lot-to-lot variation the following procedure is used (both for internal and external thread groups);

1. Compute the mean and variance of RΔPD for each material code lot.
2. Compute the weighted mean and variance of the lot averages (means) as.

$$\bar{x} = \frac{\sum \frac{n_i \bar{x}_i}{\sigma_i^2}}{\sum \frac{ni}{\sigma_o^2}}$$

$$\sigma^2_{lot-to-lot} = \frac{\sum \left(\bar{x}_i - \bar{x}\right)^2}{k-1}$$

where:

x_i = mean value of lot i

σ_i = standard deviation of lot i

k = number of lots

(Reference 9).

3. Compute the weighted variance of the within lot variation as:

$$\sigma^2_{lot} = \frac{\sum (n_i - 1)\sigma_i^2}{\sum n_i - k}$$

(Reference 8)

4. Combine the variances as;

$$\sigma^2_{total} = \sigma^2_{lot-to-lot} + \sigma^2_{lot}$$

5. Estimate the upper and lower limits as:

$$U = \bar{x} + K\sigma_{total}$$

$$L = \bar{x} - K\sigma_{total}$$

where K is the tolerance factor for a population proportion P of a normal distribution (Reference 8) with n=k lots and P=0.90 (two sided limits) and a significance level of 0.05 (95% confidence).

Limiting External Threads

Applying the calculation to the White Paper External thread data results in the following values:

K	=	2.244	(K Factor)
X	=	0.208	(Weighted Lot Mean)
$\sigma_{Lot-to-Lot}$	=	0.2439	(Lot-to-Lot Variation)
σ_{lot}	=	0.1652	(Within Lot Variation)
σ_{total}	=	0.2945	(Total Variation)
UL_{95}	=	0.8689	(Upper 95% Limit)
LL_{95}	=	-0.4528	(Lower 95% Limit)

The minimum material condition is defined by the UL_{95} value. Thus for any external thread item, there is a 95% probability its Pitch Diameter will be greater than the nominal PD minus 87% of the ANSI B1.1 Tolerance value.

Limiting Internal Threads

Applying the calculation to the White Paper internal thread data results in the following values:

K	=	2.529	(K Factor)
X	=	-0.274	(Weighted Lot Mean)
$\sigma_{Lot\text{-}to\text{-}Lot}$	=	0.2737	(Lot-to-Lot Variation)
σ_{lot}	=	0.2396	(Within Lot Variation)
σ_{total}	=	0.3638	(Total Variation)
UL_{95}	=	0.646	(Upper 95% Limit)
LL_{95}	=	-1.194	(Lower 95% Limit)

The minimum material condition is defined by the LL_{95} value. Thus for any internal thread item, there is a 95% probability its Pitch Diameter will be less than the nominal PD plus 119.4% of the ANSI B1.1 Tolerance value.

ASSESSMENT OF IMPACT ON FASTENER STRENGTH

To examine the impact of the observed dimensional deviations on the strength of a typical fastener installation the example in the White Paper was used. This computation (page 4 of Reference 1) is an all thread stud 5/8" diameter - 11 threads per inch with a matching nut. A Monte Carlo simulation was prepared using the same formulations as in the White Paper. The exception is that the distributions of Internal and External Thread deviations were used instead of the worst case values. By simulating the analysis, the upper and lower limits on thread strength corresponding to 95% probability limits can be estimated.

The formulations used were:

$$Ass = \pi(1/P)(LE)D1\max\left[0.5(1/P) + 0.57735(D2\min - D1\max)\right)$$

Where:	Ass	=	Minimum Thread Shear Area for External Threads
	P	=	Thread Pitch (inches)
	LE	=	Length Engaged (inches)
	D1max	=	Maximum Minor Diameter of Internal Thread
	D2min	=	Minimum Pitch Diameter of External Thread

The Shear Strength of the Threads is then:

Thread Strength = 0.5 * St * Ass

where:

St	=	Ultimate Tensile Strength of the Bolt Material.

For the simulation the values used were:

P	=	0.09091 inches
LE	=	0.625 inches
D1max	=	0.5460 inches (Minimum Material Condition)
D2min	=	Normal Distribution from WP Data
St	=	125 Ksi

The normal distribution had the parameters:

Mean = 0.5648 inches

S.D. = 0.0050 inches

The basic thread shear area computation is an input to the design load computation. The applied shear stress at Maximum Design Load is given by:

Applied Shear Stress = Maximum Design Load/Shear Area

From the White Paper Example (page 7) the Maximum Design Load is 7.91 Ksi and the Maximum allowable shear stress (0.6Sm) is 16 Ksi. The Monte Carlo Simulation then computes the distribution of the ratio;

Rcode =Maximum Allowable Shear Stress/Applied Shear Stress at Design Load.

The formulations were implemented in an Excel spreadsheet and the simulations run using the @Risk Monte Carlo Simulation Package.

One thousand simulation trials were run in the simulation which were sufficient to produce good convergence. The results are shown in the following tables:

Table 2- Basic Thread Strength:

Percentile	Simulation - Thread Strength	White Paper Worst Case Thread Strength
5%	38,667 pounds	37,502 pounds
50%	40,820 pounds	Not Computed
95%	42,971 pounds	47,700 pounds

The 5% percentile is the limit of interest and it is seen that the White Paper 'Worst Case" is conservative. Since the limiting values computed from the sample statistics are normalized and do not depend on item type or dimensions, it can be expected that similar results would be obtained for other item types.

The range of the predicted thread strength is also consistent with the White Paper results.

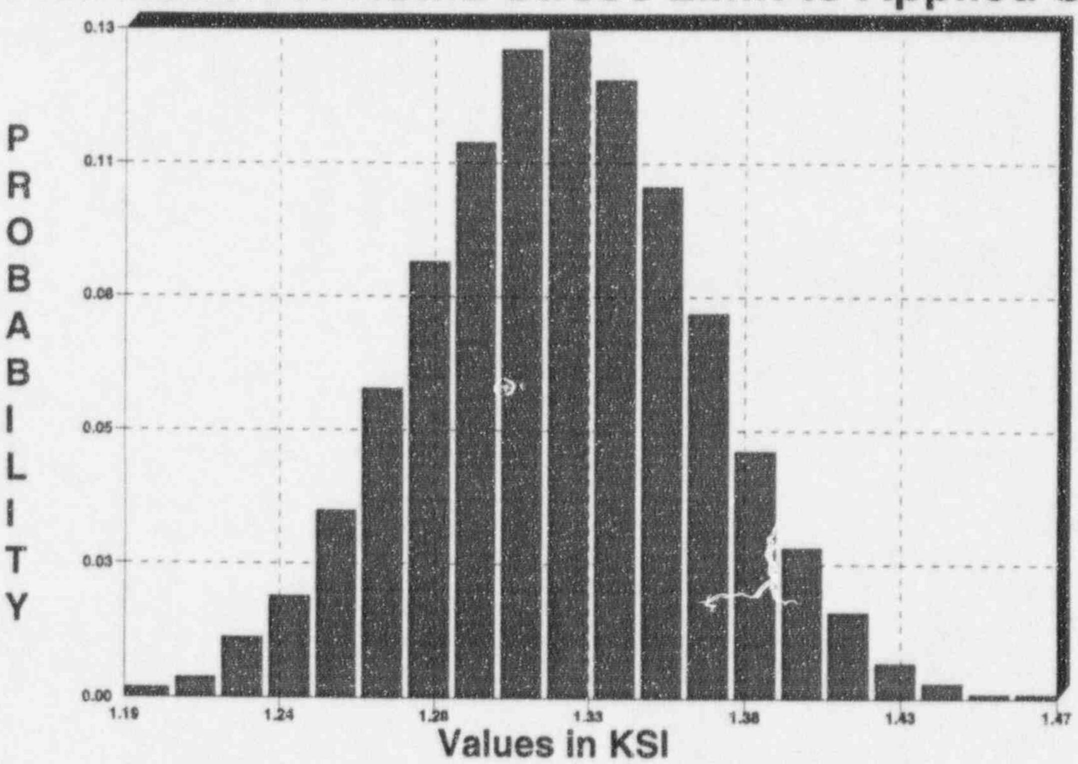
Table 3- Applied Shear Stress at Maximum Load for ASME Section III Class 1

Percentile	Simulation - Shear Stress	White Paper Worst Case Shear Stress
5%	11,500 pounds	Not Computed
50%	12,120 pounds	Not Computed
95%	12,780 pounds	13,173 pounds

The margin to the code limits is very good. Figure 5, shows the distribution of the ratio of Maximum Allowable Shear Stress/Applied Shear Stress at Design Load.

Figure 5

Distribution of ASME Stress Limit to Applied Stres



The key percentiles are shown in the following table.

Table 6 - Load Capability

Percentile	Simulation -	White Paper Worst Case
5%	1.25 (125%)	1.19 (119%)
50%	1.32 (132%)	Not Computed
95%	1.47 (147%)	Not Computed

Thus for items with adverse thread dimensions at the 95% probability/95% confidence level (approximately) there is still a 25% margin to the code allowable limits. This confirms the conclusions of the White Paper bounding computations.

It would be expected that computations for other items would show similar results.

RECOMMENDATIONS

Development of Revised Acceptance Plan for PD

Based upon the behavior of the data for both data sets, it should be possible to implement a revised acceptance plan based upon indicating gage measurement of Pitch Diameter (as well as other dimensional features). A strength based tolerance on PD can be established which should be only a modest increase over the ANSI B1.1 tolerances. The current fastener suppliers should be able to provide acceptable product with little increase in lot rejection over the current SYSTEM 21 plan.

REFERENCES

1. "A White Paper, Fastener Strength Analysis, Nuclear Safety Concern 93-11", Michael B. Ramsey, May 6, 1994.

2. ASME/ANSI B1.3M-1993 "Screw Thread Gaging Systems for Dimensional Acceptability - Inch and Metric Screw Threads"

3. "Sampling Program for Assessing, Estimating and Reporting Commercial Grade Item Quality", Procedure SO123-XXXII-2.5, Revision 1.

4. Sampling Techniques, Third Edition, W. G. Cochran, Wiley, New York, 1977.

5. Transmittal of White Paper Sample Data, Disk and Letter, Michael Ramsey to Frank Berte', July 24, 1994.

6. Transmittal of Recent Inspection Sample Data, Disk and Letter, David Ovitz to Frank Berte', July 22, 1994.

7. ASME/ANSI B1.1-1989, Unified Inch Screw Threads.

8. Statistics Manual, E.L. Crow et. al, Dover, New York, 1960.

9. Statistical Theory with Engineering Applications, A. Hald, Wiley, New York, 1952.

APPENDIX - MATERIAL CODE AND SUPPLIER VARIATION

Boxplots

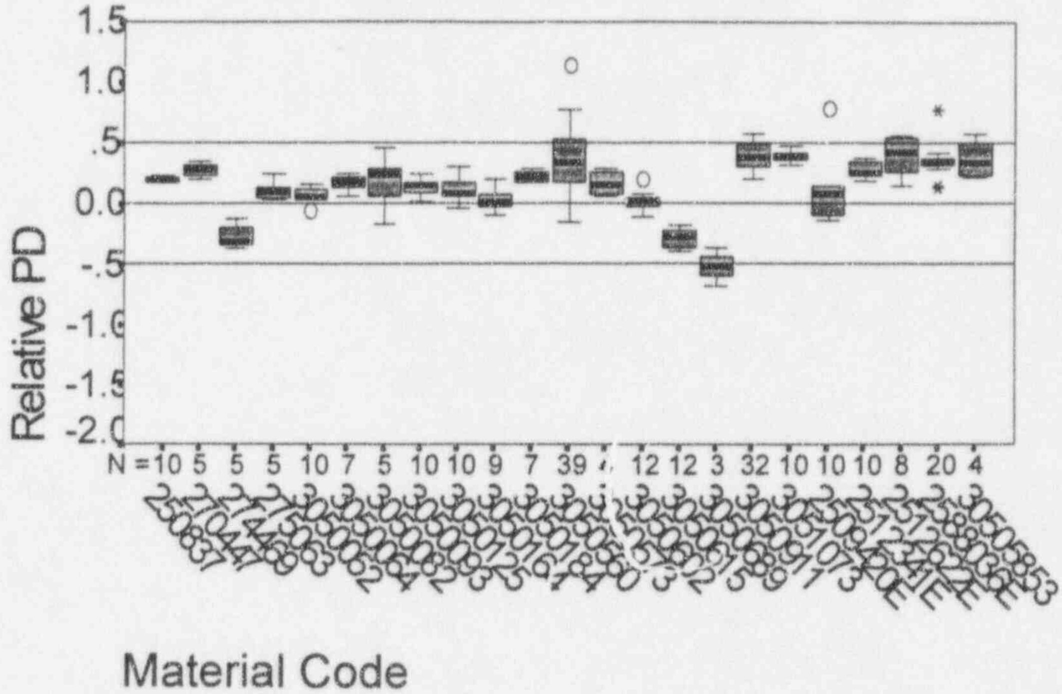
Boxplots show a number of summary statistics on a single plot. The horizontal line in the box is the median. The lower boundary of the box is the 25th percentile and the upper limit of the box is the 75th percentile. The largest and smallest observed data lying within 1.5 box lengths of the median are shown by the lines from the box ("whiskers"). Outliers and extreme values (greater than 1.5 box lengths away) are shown by asterisks and circles.

From a boxplot much information can be drawn quickly. The median shows the center of the data. The length of the box shows the spread of the data. If the median is not in the center of the box the data is skewed. The length of the distribution tails are visible in the "whiskers" and outliers.

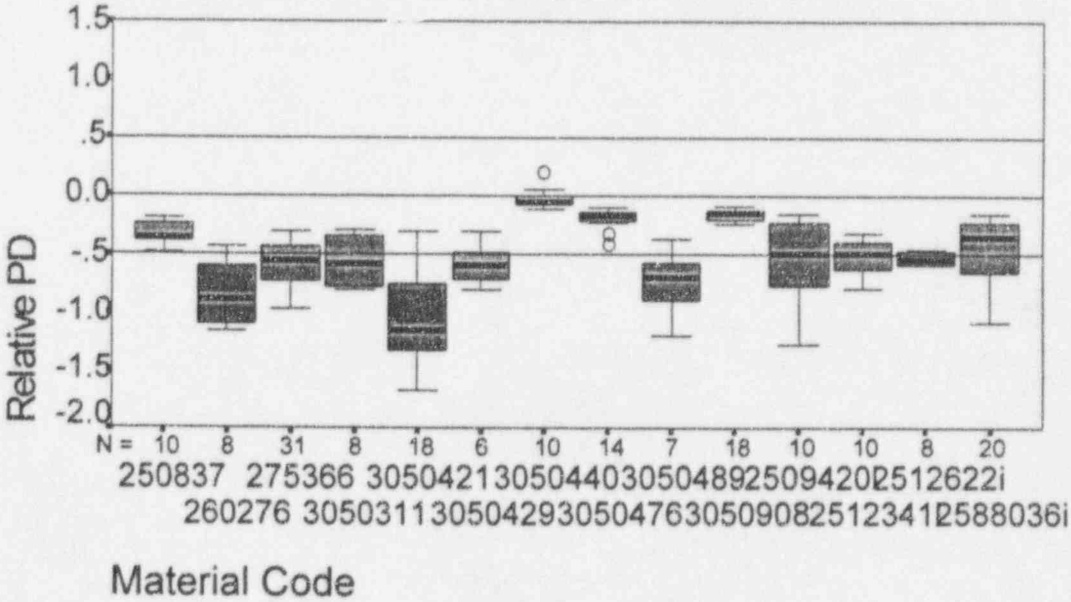
Relative PD Deviation by Material Code

The variation of pitch diameter deviation between material codes is seen in these boxplots.

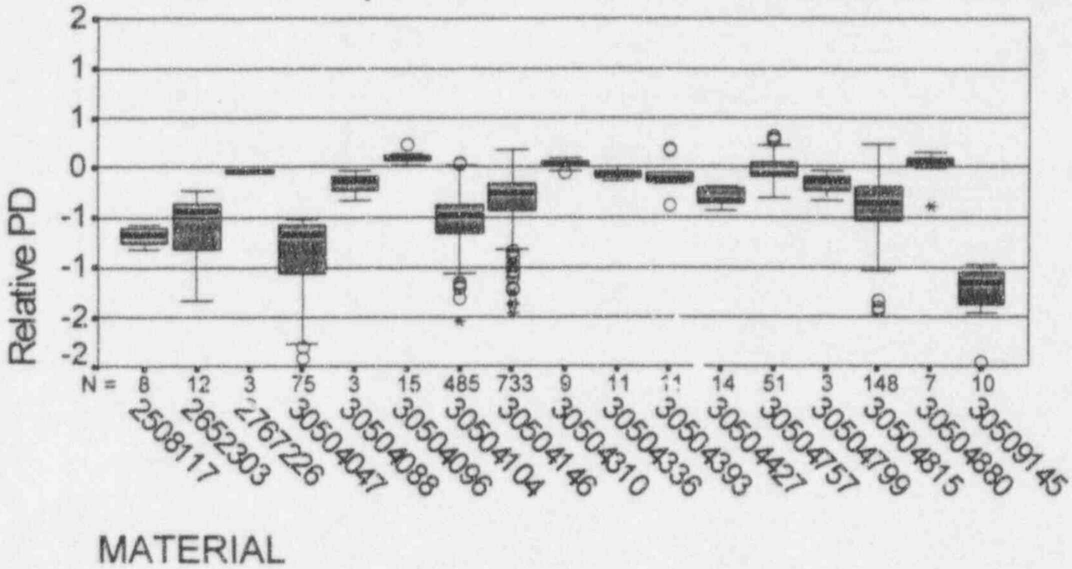
Boxplot of Relative PD Deviation vs MC WP External Thread Items



Boxplot of Relative PD Deviation by MC WP Internal Thread Data



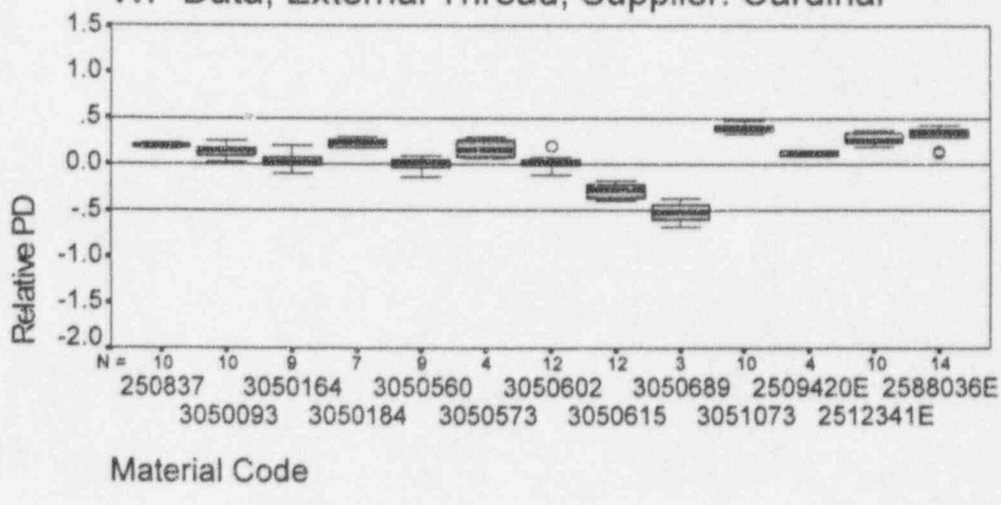
Boxplot of Relative PD Deviation by MC
Recent Acceptance Data Internal Thread Items

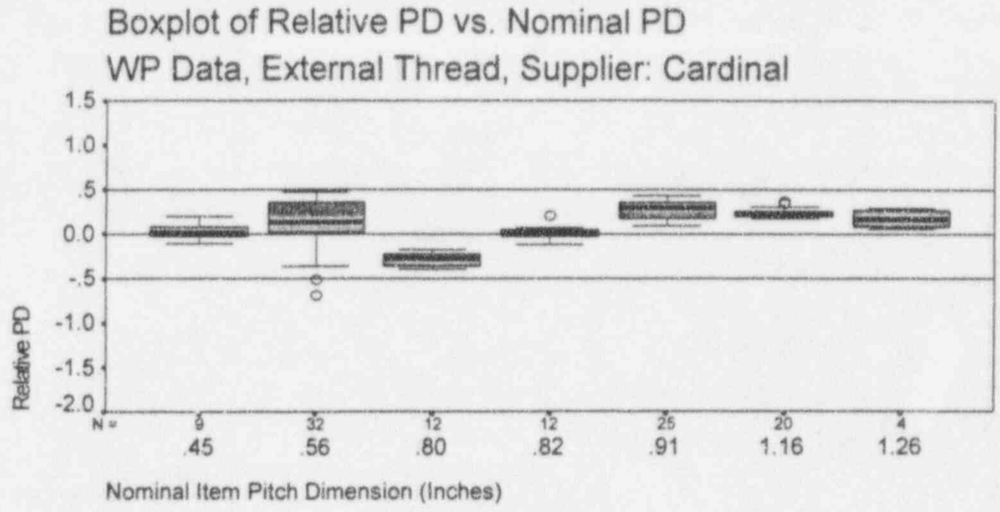


Relative PD Deviation by Supplier

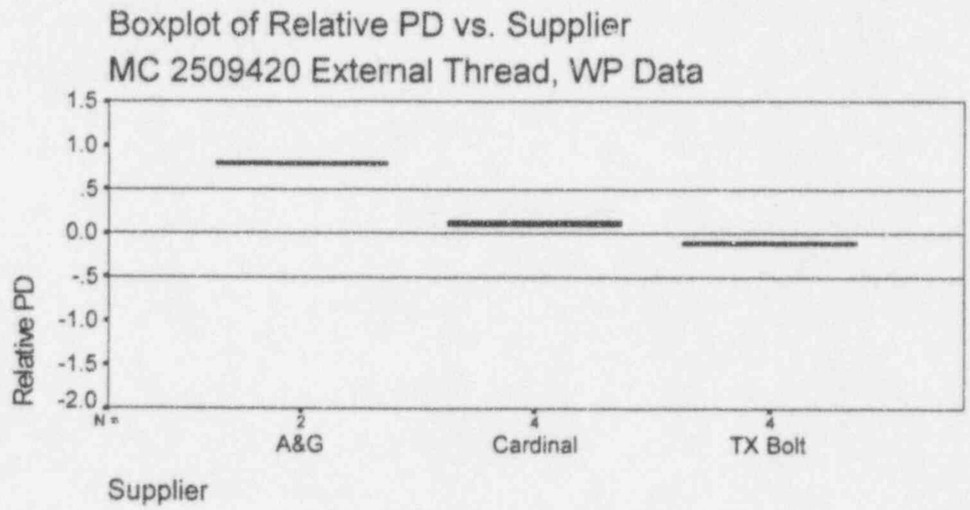
The variation in pitch diameter deviation between material codes for a single supplier is shown in these plots.

Boxplot of Relative PD vs. MC
WP Data, External Thread, Supplier: Cardinal



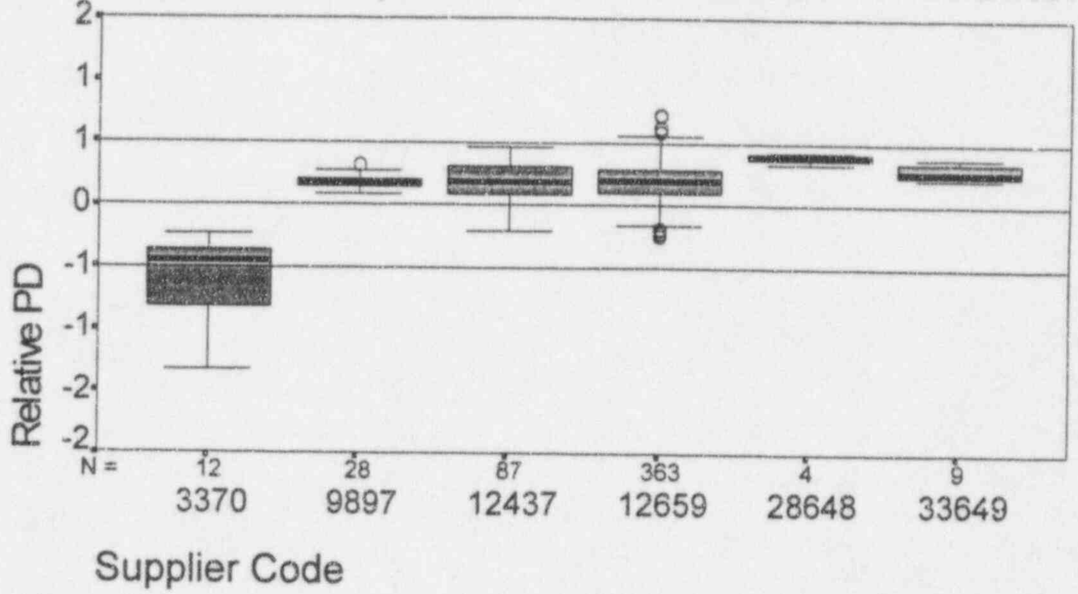


This plot shows a single material code data by supplier. These are 36" lengths of all thread and several measurements were taken along a single item. Thus the data for each supplier is a single item.

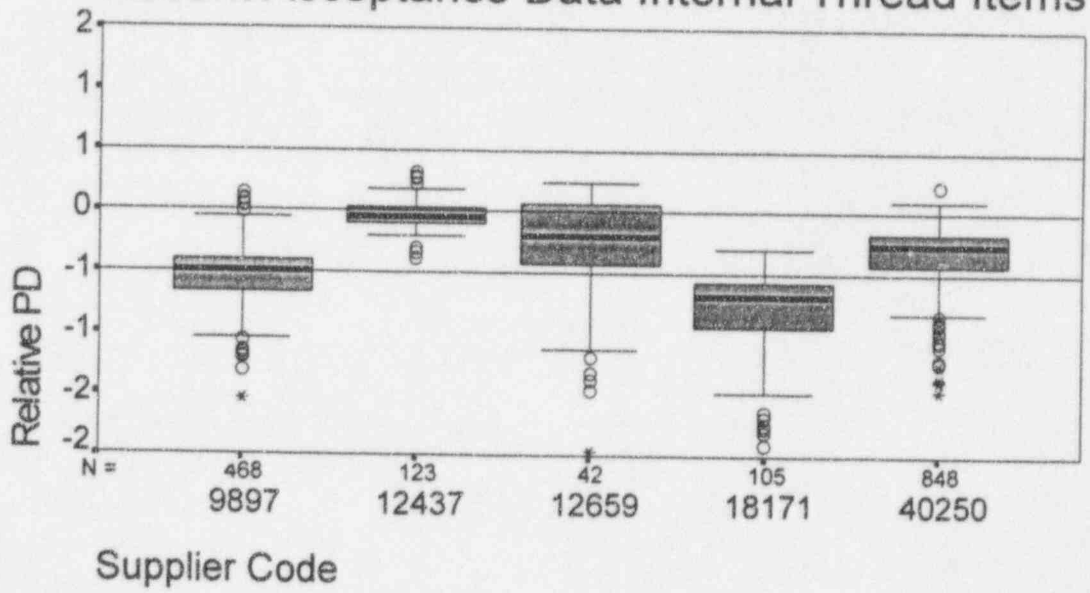


These plots show similar data for Recent inspections data.

Boxplot of Relative PD Deviation by Supplier
Recent Acceptance Data External Thread Item



Boxplot of Relative PD Deviation by Supplier
Recent Acceptance Data Internal Thread Items



ATTACHMENT 3

COMMENTS ON SAN ONOFRE NUCLEAR GENERATING STATION BOLTING

BY

REEDY ASSOCIATES, INC.

May 2, 1994

REEDY ASSOCIATES, INC.
ENGINEERING MANAGEMENT CONSULTANTS

May 2, 1994
SCE-94-008

Mr. Michael B. Ramsey
Senior Engineer
Souther California Edison
San Onofre Nuclear Generating Station
Post Office Box 128
San Clemente, CA 92674-0128

Dear Mike,

Enclosed are my comments regarding the San Onofre Nuclear Generating Station bolting.

If you have any questions, please feel free to give me a call.

Very truly yours,

Roger F. Reedy

Roger F. Reedy

RFR\n
Enc.

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REEDY ASSOCIATES, INC.
ENGINEERING MANAGEMENT CONSULTANTS

COMMENTS ON SAN ONOFRE NUCLEAR GENERATING STATION BOLTING

Nuclear power plants are designed by Registered Professional Engineers trained and qualified to design pressure retaining equipment and the associated bolting. The pressure retaining components in these plants are designed and constructed to strict requirements of the ASME (American Society of Mechanical Engineers) Boiler and Pressure Vessel Code, Section III for Nuclear Components. The requirement to meet the provisions of the ASME Section III Code is mandated by the U.S. Federal Regulations 10CFR50.55a. Bolting for pressure components (including all nuclear components in the piping systems) must comply with Section III of the ASME Code.

I have reviewed and evaluated the system used by SCE (Southern California Edison) at their San Onofre Plant to inspect, evaluate, and accept or reject bolting and I am firmly convinced that the SCE inspection program for bolting fully complies with the Federal Regulations and the ASME Code. In fact, the bolting inspection program exceeds both Federal and ASME Code requirements.

Although the act of bolting items together appears to be a simple operation (almost everyone has assembled nuts and bolts at some time in their life), there are significant engineering principles involved in the design of bolted connections. It is the responsibility of the design engineer to consider these principles in the design of the bolted equipment and to establish or approve appropriate tolerances and acceptance criteria. The nuclear plant must be constructed and maintained to the tolerances and acceptance criteria selected by the responsible engineer.

For pressure retaining components (piping, pumps, valves, and pressure vessels), bolting is pre-stressed. That is, during assembly, the bolts are tightened in such a manner that system pressure and seismic loads will not increase the stress in the bolts. This operation is known as prestressing the bolts. That means that the worst condition of loading occurs when the workmen first tighten the bolts. If the bolts don't fail during this tightening as prestressing, they will not fail in service due to dimensional variations. The prestressing operation is a good test for any bolting because any significant problems associated with the bolting will show up during the initial bolting of the items rather than during plant operation.

In the design of bolting, the ASME Code requires a design factor (safety factor) of 4 or 5. This means that bolts shouldn't fail until they are loaded to a level 4 or 5 times the design load. There has been more than 50 years of experience using these design factors,

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and bolting failures caused by dimensional variations has not been a problem in the piping industry. It should also be noted that bolting in pressure piping is redundant. That is, failure of one bolt will never cause failure of the connection because there are many other bolts to take up any additional load caused by the failure of a single bolt.

The ASME bolting standards allow the design engineer to select from a series of acceptable tolerances. The most common way of assuring tolerances for pressure retaining applications is to allow the "Go/No Go" gage system. This is known as System 21. This system of tolerances is fully acceptable for ASME Code applications and is used extensively in industry, including the U.S. Navy nuclear reactor submarine program. The ANSI/ASME B1.2-1983 Standard "Gages and Gaging for Unified Inch Screw Threads" states:

> "Product threads accepted by a gage of one type may be verified by other types. It is possible, however, that parts which are near a limit may be accepted by one type and rejected by another. Also, it is possible for two individual limit gages of the same type to be at opposite extremes of the gage tolerances permitted, and borderline product threads accepted by one gage could be rejected by another. For these reasons, a product screw thread is considered acceptable when it passes a test by <u>any</u> of the permissible gages in ANSI B1.3 for the gaging system specified, provided the gages being used are within the tolerances specified in this Standard." [Emphasis added.]

The manufacturer of the bolts has the responsibility for meeting the bolting tolerances specified by the purchaser. The user of the bolts can re-check the bolt dimensions if it is felt necessary. Normal nuclear industry practice is that sometimes this tolerance verification is performed, sometimes it is not. The tolerance verification is not a requirement of any regulations or Code. Tolerance Systems 21, 22, and 23 are not used to determine the structural adequacy of the bolts. As stated in the ASME B1.3M-1992 Standard (4b), which defines the Systems, "Screw Thread Gaging Systems for Dimensional Acceptability - Inch and Metric Screw Threads," paragraph 4(b), "The difference between gaging systems is the level of inspection deemed necessary to satisfy that dimensional conformance has been achieved . . . ."

Note that the Standard implies there is no strength criteria associated with any of these inspection systems. This is the correct implication. The bolting gaging systems are for inspection purposes only. The engineer designing the bolting determines which system is appropriate to the design, and inspectors must assure that the engineer's tolerances are met.

The issue at San Onofre centers around whether fasteners which pass System 21 requirements, but exhibit minor dimensional out-of-tolerance conditions when examined to System 22 requirements, will fail in service. The increase in measurement accuracy identified by System 22 or 23 inspections has no significant effect on the strength of the bolting. Therefore, "Go/No Go" gages are appropriate for inspecting ASME Code bolting.

SCE has decided to screen bolting suppliers by using closer tolerances by using special gages which are more accurate, measure major characteristics and can check bolts much more quickly. These special gages are good, but are many times more accurate than necessary. If the readings from the special gage show deviations from either System 22 or 23 tolerances, the bolting is still acceptable if within the tolerances specified by, or acceptable to, the responsible engineer.

It is my professional opinion that the SCE quality program meets the ASME Code and the Federal Regulations. Neither the ASME Code nor the Federal Regulations, nor any heavy industry bolting standards require the use of the new "high tech" gages and System 22 or 23 inspections. Their use is far beyond requirements and industry practice. The increase in dimensional accuracy afforded by measuring thread attributes by System 22 or 23 has no significant effect on the strength of the bolting, is much less than the design safety factors inherent in Code applications and therefore has no bearing on nuclear safety. The analysis provided in the SONGS White Paper adequately demonstrates these conclusions by the specific sample inspection results. The issues regarding "high rejection rates", "don't meet standards" and "out-of-tolerance" regarding the SCE bolting inspection program, are absolutely untrue.

Roger H Reedy

Roger F. Reedy, P.E.
California #27562

ATTACHMENT 4

STATISTICAL ANALYSIS OF NRC AUDIT DATA FOR SONGS FASTENER STOCK

TETRA ENGINEERING GROUP
November 17, 1994

Statistical Analysis of NRC Audit Data for SONGS Fastener Stock

Tetra Engineering Group, Inc.
Report 94-SCE-005
November 17, 1994

Authors

Dr. Frank J. Berte'
Mr. David S. Moelling, PE
Mr. Frederick C. Anderson, PE

Executive Summary

A random sample of SCE warehouse fastener stock was obtained and inspected as requested by a NRC team. This random sample included 44 bins, with a total of 519 items. The intent was to provide an independent sample of the warehouse stock. This sample included small fasteners (nominal diameter less than 0.5") which were excluded from previous samples used to prepare the SCE White Paper on Fastener Strength. The intent of this report is to compare the results of dimensional measurements of the NRC data set with the data used for the White Paper.

The following conclusions can be drawn from the NRC sample data.

1. The NRC data is consistent with the White Paper Data. This confirms that the sampling procedures performed for the White Paper data set were sufficiently random to allow valid statistical results to be drawn. The limits computed from the White Paper Data bound the pitch diameter deviations in the NRC Data set.

2. The NRC data shows similar behavior in bin-to-bin and supplier-to-supplier variation as the White Paper Data set. This confirms the conclusion that there is no systematic supplier related trends in thread deviations.

3. NRC data for smaller items not considered by the White Paper show a slightly wider range of deviation than the larger items. Otherwise they are consistent with the observations for the White Paper Data.

4. Thread Strength Margins for the smaller items are adequate even with the slightly larger deviations from the ANSI B1.1 thread limits.

5. Thread Strength Margins for the larger items are consistent with those computed for the White Paper Data. These demonstrate large margins to ASME code service requirements.

Introduction

A random sample of SCE warehouse fastener stock was obtained and inspected as requested by an NRC team. This random sample included 44 bins, with a total of 519 items. The intent was to provide an independent sample of the warehouse stock. This sample included small fasteners (nominal diameter less than 0.5") which were excluded from previous samples used to prepare the SCE White Paper on Fastener Strength. The intent of this report is to compare the results of dimensional measurements of the NRC data set with the data used for the White Paper.

Review of NRC Sampled Data

The NRC requested an audit sample of 44 randomly selected bins on September 13, 1994. Random samples were drawn from these bins giving a total of 519 items (Reference 1) . Of these 188 were of nominal size (diameter) of 0.5" or greater and 331 were of nominal size less than 0.5". Of the set of 0.5" and greater, 7 had no measured data due to the fact that no Johnson Gage Segments were available in this size for System 22 Pitch Diameter measurements, this left 181 fasteners with measured data. The set of 0.5" and greater diameter is directly comparable with the set of data used for the White Paper (Reference 2) which excluded items smaller than this. The analysis of the smaller diameter data set was performed in the same manner as was done for the White Paper data set (Reference 3) and is reported in Appendix 1. The data shows the same large bin-to-bin (Material Code) variation seen in the White Paper data, and the same lack of systematic trend with supplier as seen in the White Paper data. The boxplots illustrating these effects are provided in Appendix 2.

Comparison of NRC Sampled Data with White Paper Data

External Threads

There were 43 external thread items of nominal size greater than or equal to 0.5" in the NRC sampled data. These were drawn from nine material codes. The basic statistics for these items are given in Table 1.

Table 1

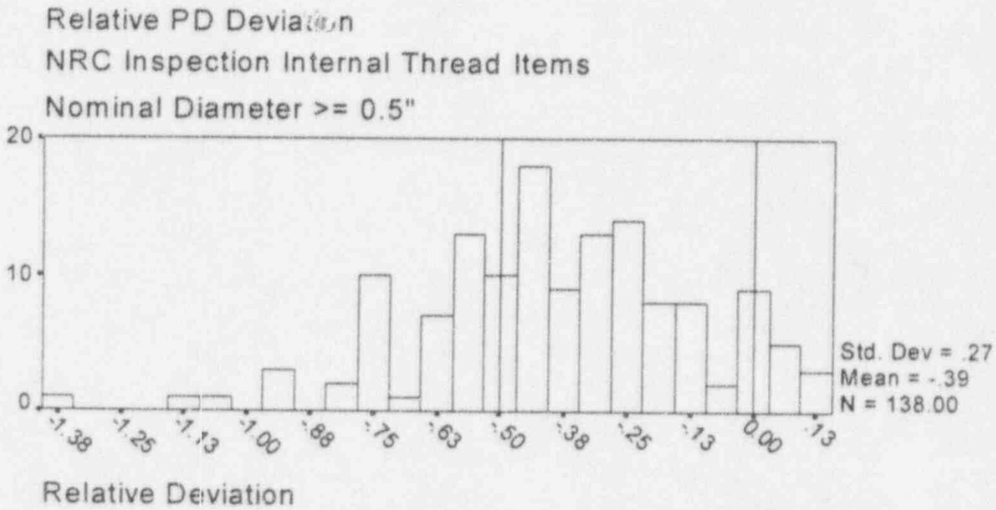
Relative Pitch Diameter Deviation Statistics

Statistic	NRC Sample	White Paper Sample
Raw Mean	0.1658	0.22
Weighted Mean	0.2153	0.208
Raw Standard Deviation	0.2606	0.260
Weighted Standard Deviation	0.2945	0.2870

Since the NRC Data Set covered fewer material codes than the White Paper data set, the effects of bin-to-bin variation have a larger impact on the computed raw means and standard deviations. Weighted Means and Standard Deviations account for this effect and a comparison of these shows the two samples are in good agreement.

The distribution of this data is shown in Figure 1.

Figure 1



Another comparison is the fraction of samples from the NRC data lying outside of the upper and lower limits computed from the White Paper Data (Reference 3). Table 2 shows this comparison.

Table 2

Comparison of NRC Data Quantiles with White Paper Limits

Limit	White Paper Value	% of NRC Data Exceeding
Upper 95% limit	0.8689	0%
Lower 95% limit	-0.4528	0%

The computed limits clearly bound the NRC data set. This clearly supports the applicability of the conclusions of the White Paper to the Warehouse stock.

Internal Threads

There were 138 internal thread items of nominal size greater than or equal to 0.5" in the NRC sampled data. These were drawn from 12 material codes. The basic statistics for these items are given in Table 3.

Table 3

Relative Pitch Diameter Deviation Statistics

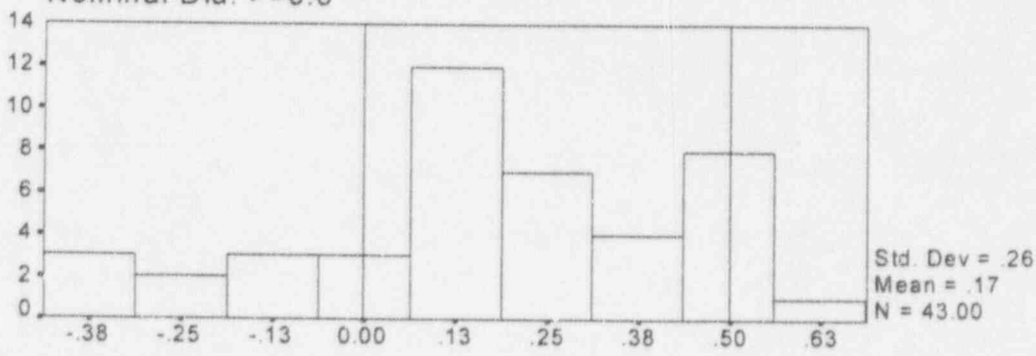
Statistic	NRC Sample	White Paper Sample
Raw Mean	-0.3873	-0.51
Weighted Mean	-0.553	-0.27
Raw Standard Deviation	0.2714	0.35
Weighted Standard Deviation	0.2765	0.364

The White Paper data set had 178 items from 12 material codes. The White Paper data showed a greater spread and larger average deviation than the NRC data.

The distribution of this data is shown in Figure 2.

Figure 2

Histogram of Relative PD Deviation
NRC Data External Thread Items
Nominal Dia. >=0.5



Relative Deviation

Table 4 shows the fraction of samples from the NRC data lying outside of the upper and lower limits computed from the White Paper Data (Reference 3).

Table 4

Comparison of NRC Data Quantiles with White Paper Limits

Limit	White Paper Value	% of NRC Data Exceeding
Upper 95% limit	0.646	0%
Lower 95% limit	-1.194	1%

The NRC data set is consistent with the limits computed from the White Paper Data. This clearly supports the applicability of the conclusions of the White Paper to the Warehouse stock.

Assessment of Fastener Strength

The impact of the observed dimensional deviations on the strength of fasteners greater than (or equal to) 0.5 inches in diameter was examined by performing the White paper strength calculations for one typical fastener. The 5/8 all thread stud - material code 30505606, Class 2a, was selected a representative fastener. A Monte Carlo simulation was prepared using the distributions of internal and external thread deviations for fasteners greater than or equal to, 0.5 inches in diameter. One thousand simulation trials were performed to generate the distributions. From this simulation, upper and lower limits on thread strength corresponding to 95% probability limits were estimated. Since the NRC data set is bounded by the White Paper Data, the White Paper data was used for the strength assessment.

The formulations used were:

$$\mathbf{Ass = \pi(1/P)(LE)D1max[0.5(1/P) + 0.57735(D2min - D1max)]}$$

Where:

Ass	=	Minimum Thread Shear Area for External Threads
P	=	Thread Pitch (inches)
LE	=	Length Engaged (inches)
D1max	=	Maximum Minor Diameter of Internal Thread
D2min	=	Minimum Pitch Diameter of External Thread

The shear strength of the external thread (TSS) is then given by:

TSS = 0.5(St)(Ass)

Where:
St = Minimum Ultimate Tensile Strength of the Bolt Material

For the simulation the values used were:

P	=	0.0909inches
LE	=	0.625 inches
D1max	=	0.6113
D2min	=	Normal Distribution from White Paper Data Set
St	=	125000 psi

The Normal Distribution had the parameters:

Mean	=	0.5628
S.D.	=	0.0022

The thread shear area (Ass) and the thread shear strength were calculated. In addition, the ratio of the Thread Shear Strength to the Maximum Preload was determined using the following formula:

$$TSS/MP = TSS/[0.7854(D-\{0.9743/(1/P)\})^2(2Sm)]$$

Where:

D	=	0.625 inches	(Nominal Diameter)
Sm	=	25000 psi	(Code Allowable Stress - Class 2 and 3)

Figure 3

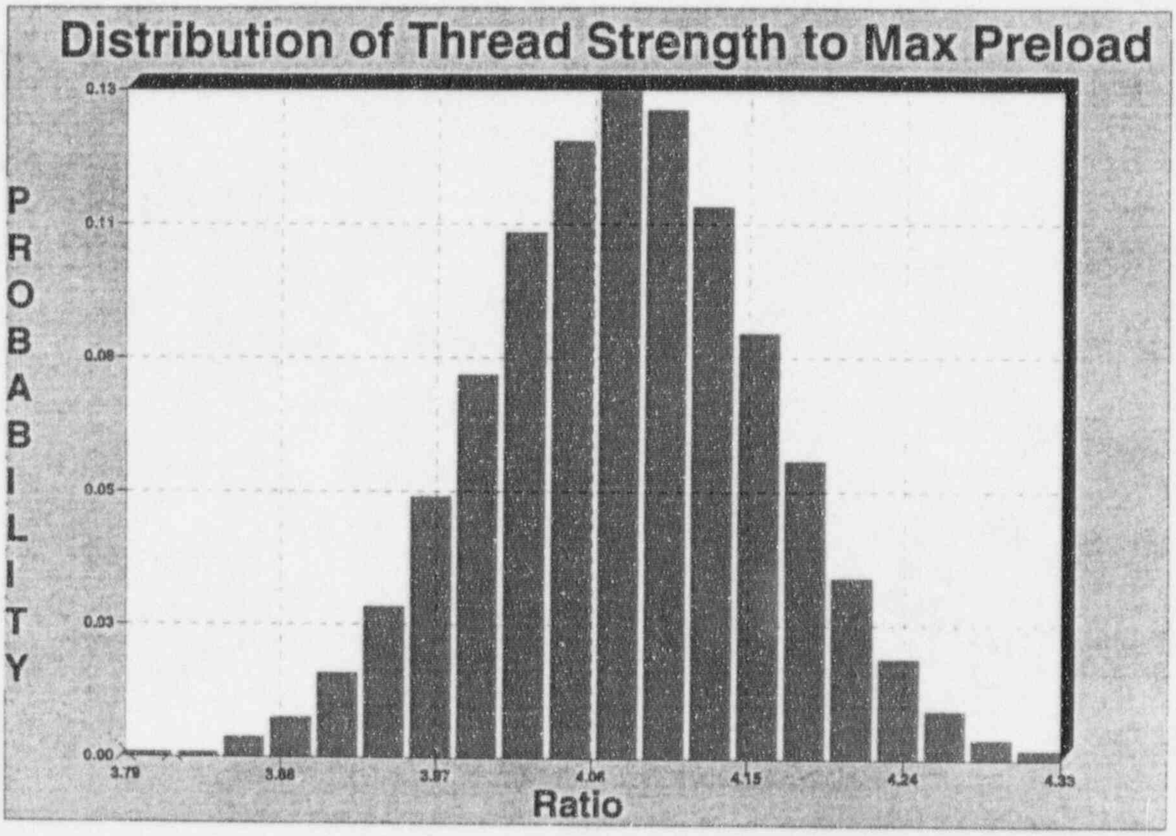


Table 5 - White Paper Sample Data Set - 5/8 All thread

Percentile	Ratio Thread Shear to Max Preload
5%	3.947
50%	4.079
95%	4.211

The results show that even at the 5% percentile level the thread shear strength is considerably higher than the maximum bolt preload.

Conclusions

The following conclusions can be drawn from the NRC sample data.

1. The NRC data is consistent with the White Paper Data. This confirms that the sampling procedures performed for the White Paper data set were sufficiently random to allow valid statistical results to be drawn. The limits computed from the White Paper Data bound the pitch diameter deviations in the NRC Data set.

2. The NRC data shows similar behavior in bin-to-bin and supplier-to-supplier variation as the White Paper Data set. This confirms the conclusion that there is no systematic supplier related trends in thread deviations.

3. NRC data for smaller items not considered by the White Paper show a wider range of deviation than the larger items. Otherwise they are consistent with the observations for the White Paper Data.

4. Thread Strength Margins for the smaller items are adequate even with the slightly larger deviations from the ANSI B1.1 thread limits.

5. Thread Strength Margins for the larger items are consistent with those computed for the White Paper Data.

References

1. Electronic Transmittal, M. Ramsey (SCE) to F. Berte' (Tetra), 10/3/94.

2. "A White Paper; Fastener Strength Analysis Nuclear Safety Concern 93-11", M.B. Ramsey, (SCE), 5/6/94.

3. "Statistical Validation of SCE Fastener Strength Analysis White Paper Data Base", Tetra Engineering Group Report, 94-SCE-003, 8/5/94.

APPENDIX 1

NRC Sample Data Set
Fasteners Less than 0.5 Inches in Diameter

Characteristics of Data

Homogeneity of PD Data

The NRC Sample data set of items less than 0.5 inches in diameter contains a mixture of item types and sources. The assumption of homogeneity with respect to the relative deviation in Pitch Diameter (R△PD) was examined for both internal and external threads. The R△PD is defined as the nominal value minus the measured value. To check for homogeneity of R△PD between internal and external thread items, histograms were prepared for all internal and external items less than 0.5 inches in diameter and compared.

Figure 1 shows the distribution of internal threads from the NRC Sample data set items less than 0.5 inches in diameter.

Figure A-1

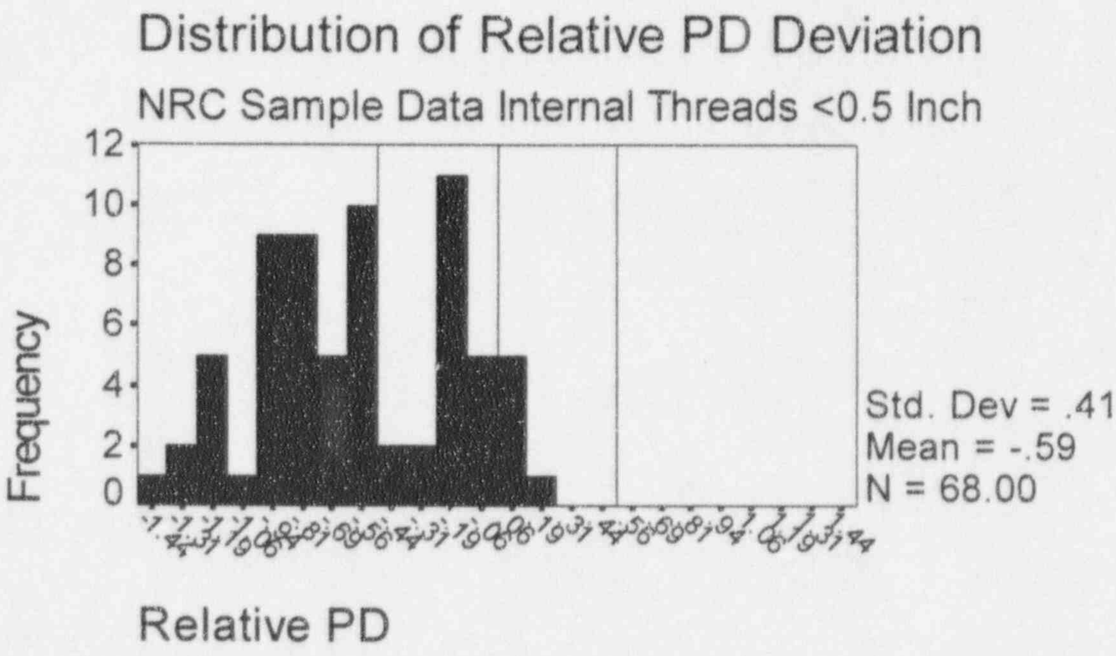
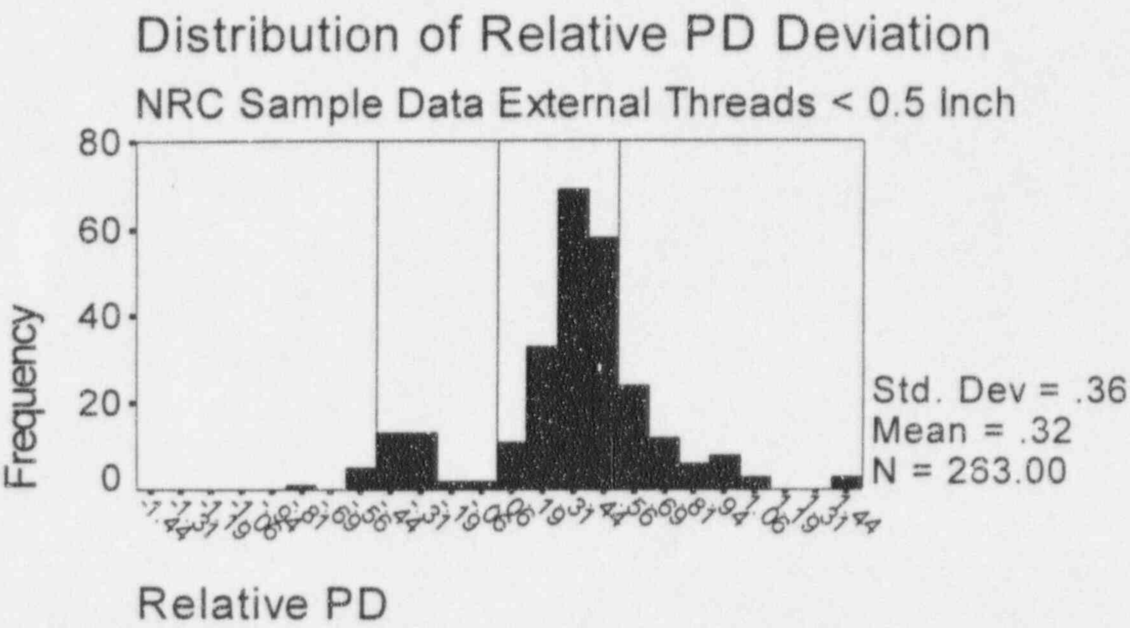


Figure A-2 shows the distribution of external threads from the NRC Sample data set items less than 0.5 inches in diameter

Figure A-2



A large number of different item types and suppliers are represented in the sample population for both internal and external threads. Examination of the two figures clearly shows that the internal and external thread types are statistically different. The internal thread items tend to have deviations in the direction of larger than nominal pitch diameters and the external thread items tend to have deviations in the direction of smaller than nominal pitch diameters.
Table A-1 shows a comparison between the internal and external thread item statistics.

Table A-1

Statistic	External Thread	Internal Thread
Mean PD Deviation	.32	-.59
Std. Dev. of PD Deviation	.36	.41

Material Code

A key factor to be examined for homogeneity of PD deviation is the individual material codes. Homogeneity within a material code is important since many plant applications would require multiple items from the same material code. For example, several of the same type bolt installed in a single flange. Boxplots were used to rapidly compare statistics of the individual material codes. Figures A-3 and A-4 show the relative PD deviation as a function of material code for internal and external thread items respectively. There are significant differences between many material codes for both the external and internal thread types. The external thread type has a wider range of variation than the internal thread type, although this may be due to the larger external thread population.

Figure A-3

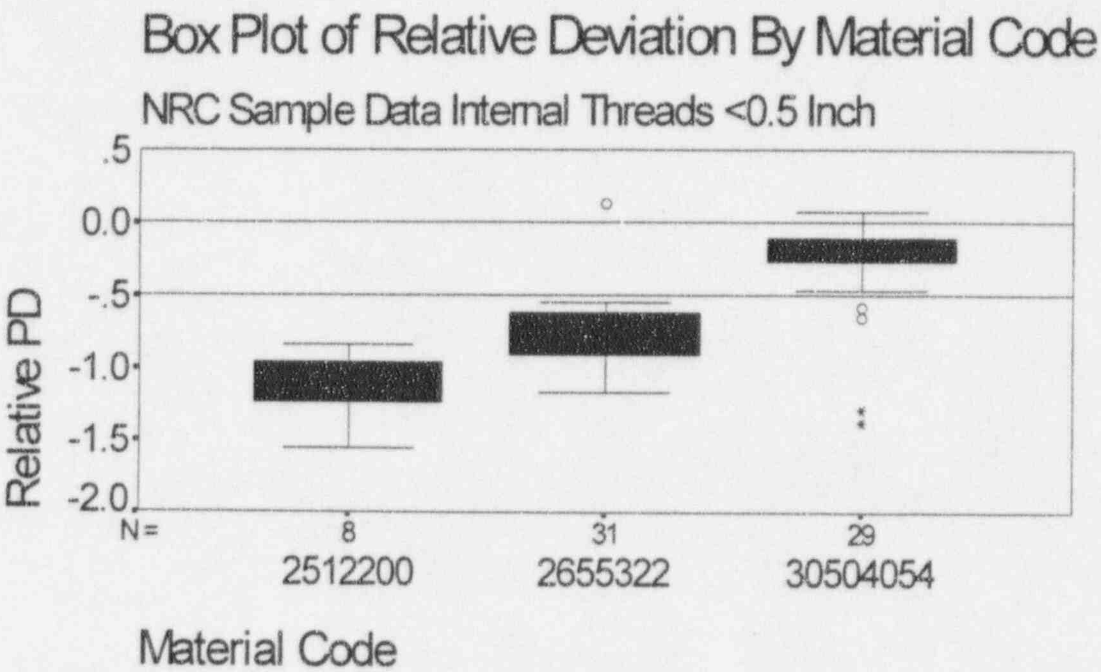
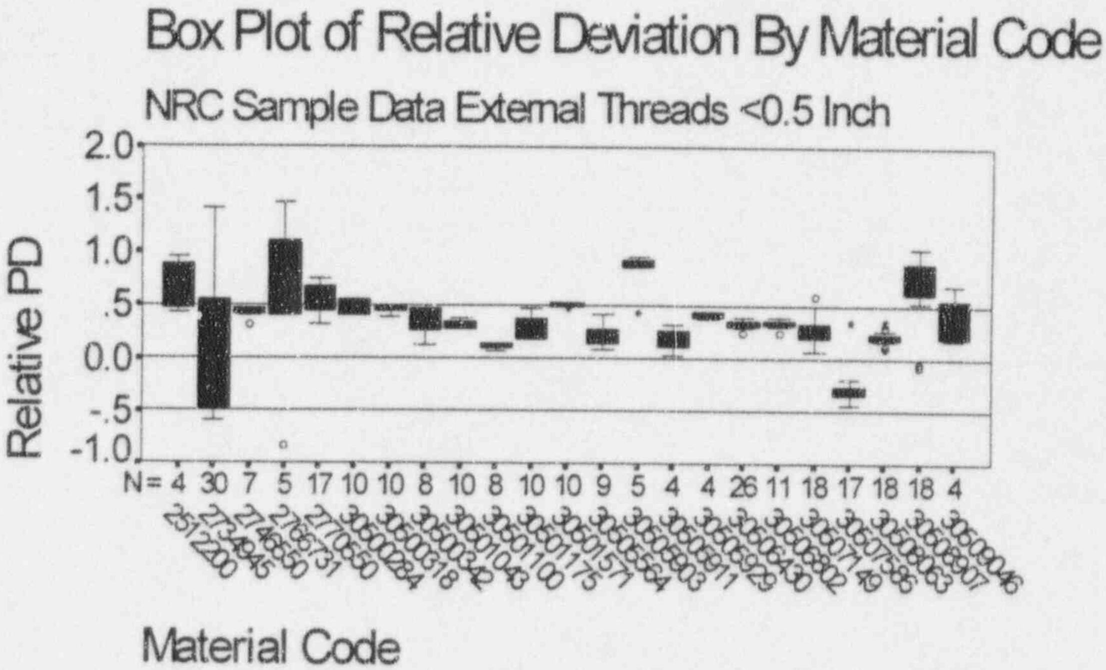


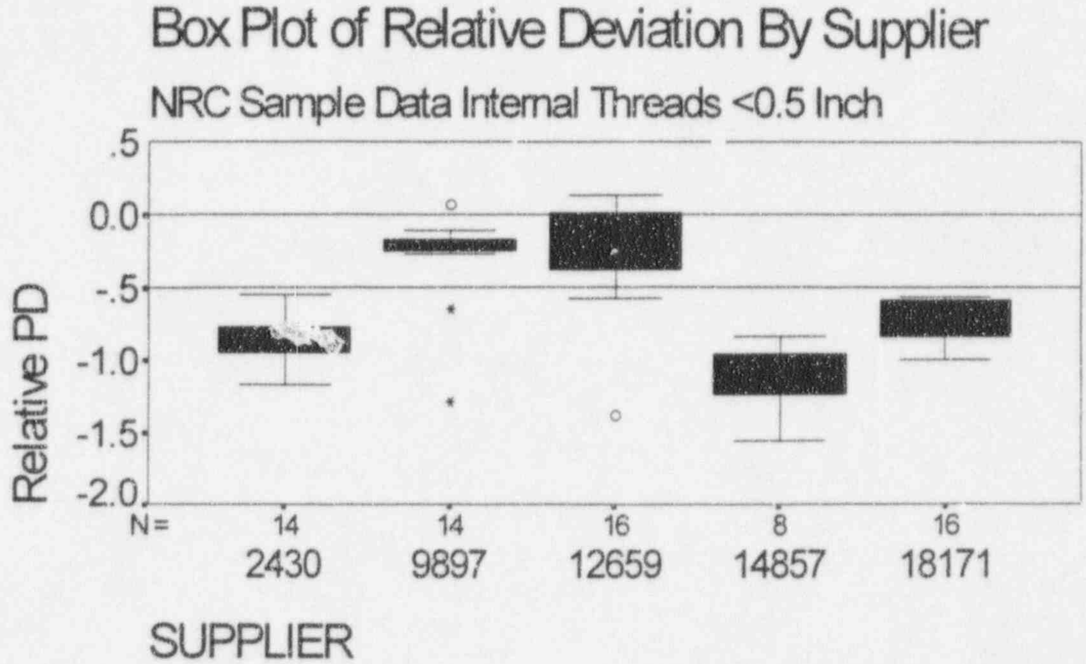
Figure A-4



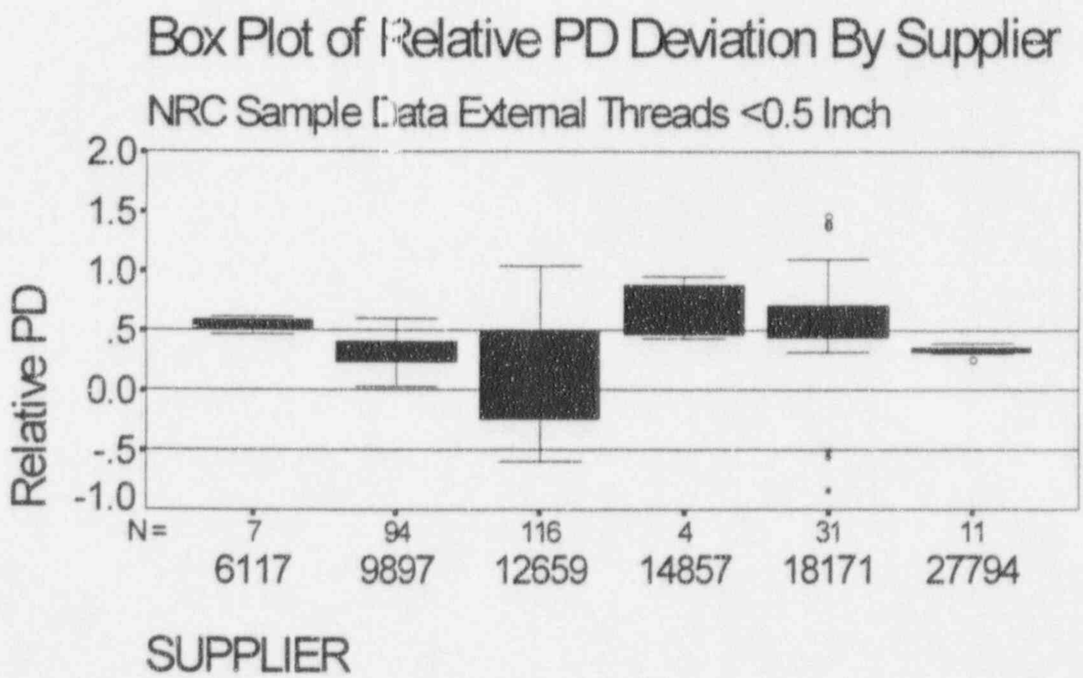
Supplier

Variations in thread items from a given supplier and variations between suppliers were examined using boxplots. Figure A-5 is the boxplot for internal thread items by supplier. Figure A-6 is the boxplot for external thread items by supplier. A significant variation between suppliers is observed for both thread types. However, the variation in material code for a given supplier would argue against any systematic effect of supplier. The conclusion is that there is no significant systematic supplier effect and that variations are very likely to be on a production lot basis.

Figure A- 5



FigureA-6



Computation of Limiting Out of Tolerance Values

The limiting out of tolerance values were computed for both external and internal thread types using the methodology provided in Reference 3. Table A-2 provides the results of those computations.

Table A-2

Statistic	External Threads	Internal Threads
K (Statistical Factor)	2.244	8.38
X (Weighted Lot Mean)	0.445	-0.686
$\sigma_{\text{Lot to Lot}}$ (Lot to Lot Variation)	0.0614	0.1874
σ_{Lot} (Within Lot Variation)	0.0776	0.0818
σ_{Total} (Total Variation)	0.1390	0.2692
UL_{95}(Upper 95% Limit)	0.757	1.570
LL_{95} (Lower 95% Limit)	0.1335	-2.942

The minimum material condition is defined by the LL_{95} value. Thus for any internal thread item, there is a 95% probability that its pitch diameter will be less than the nominal PD plus 294.2% of the ANSI B1.1 Tolerance value. For external threads the limiting condition is defined by the UL_{95} value. Similarly, there is a 95% probability that any external thread item will have a pitch diameter which is greater than the nominal PD minus 75.7% of the ANSI B1.1 Tolerance value.

Assessment of Impact on Fastener Strength

The impact of the observed dimensional deviations on the strength of fasteners less than 0.5 inches in diameter was examined by performing the White paper strength calculations for one typical fastener. The 3/8 Cap Screw - material number 2734945, Class 3a, was selected a representative f. tener. A Monte Carlo simulation was prepared using the distributions of internal and external thread deviations for fasteners less than 0.5 inches in diameter. One thousand simulation trials were performed to generate the distributions. From this simulation, upper and lower limits on thread strength corresponding to 95% probability limits were estimated.

The formulations used were:

Ass = π(1/P)(LE)D1max[0.5(1/P) + 0.57735(D2min - D1max)]

Where:

Ass	=	Minimum Thread Shear Area for External Threads
P	=	Thread Pitch (inches)
LE	=	Length Engaged (inches)
D1max	=	Maximum Minor Diameter of Internal Thread
D2min	=	Minimum Pitch Diameter of External Thread

The shear strength of the external thread (TSS) is then given by:

TSS = 0.5(St)(Ass)

Where:

St	=	Minimum Ultimate Tensile Strength of the Bolt Material

For the simulation the values used were:

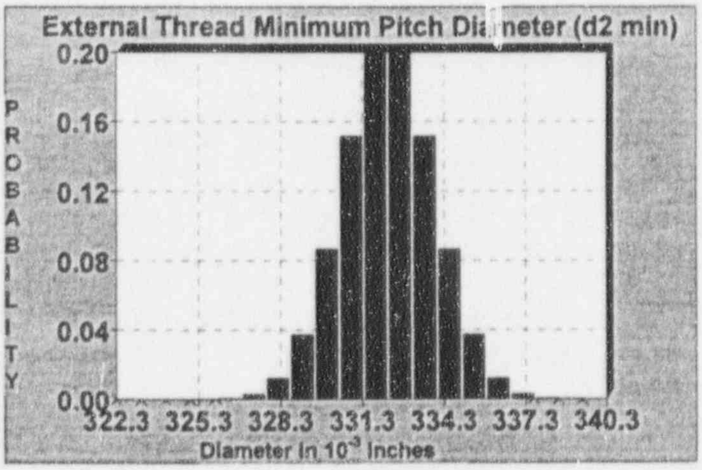
P	=	0.0625 inches
LE	=	0.375 inches
D1max	=	0.3387
D2min	=	Normal Distribution from NRC Sample Data Set
St	=	125000 psi

The Normal Distribution had the parameters:

Mean	=	0.445327
S.D.	=	0.508586

This distribution is shown graphically in figure A-7.

Figure A-7- NRC Sample Data Set - 3/8 Cap Screw Data



The thread shear area (Ass) and the thread shear strength were calculated for the 3/8 Cap Screw data from the NRC Sample data set. In addition, the ratio of the Thread Shear Strength to the Maximum Preload was determined using the following formula:

$$TSS/MP = TSS/[0.7854(D-\{0.9743/(1/P)\})^2(2Sm)]$$

Where:

D	=	0.375 inches	(Nominal Diameter)
Sm	=	25000 psi	(Code Allowable Stress - Class 2 & 3)

Figure A-8 and A-9 show the distributions of thread shear strength and thread shear area respectively.

Figure A-8

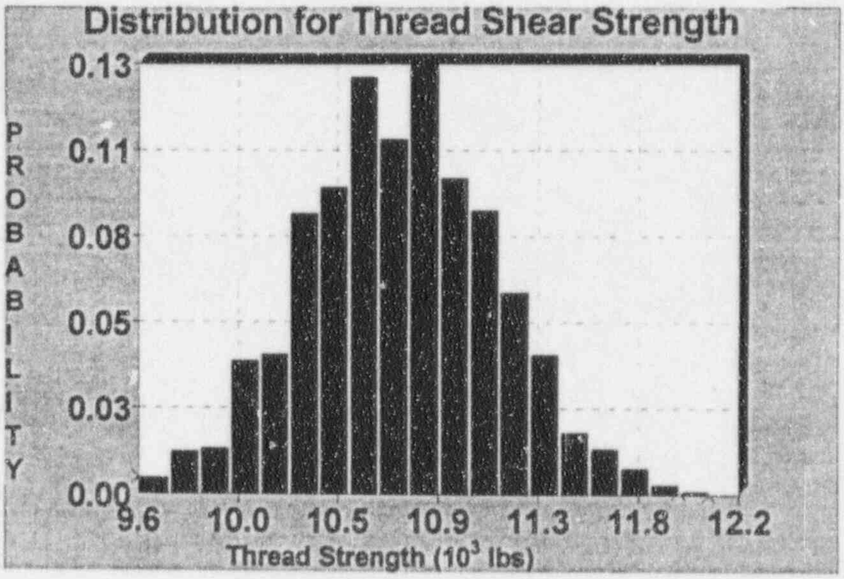


Figure A-9

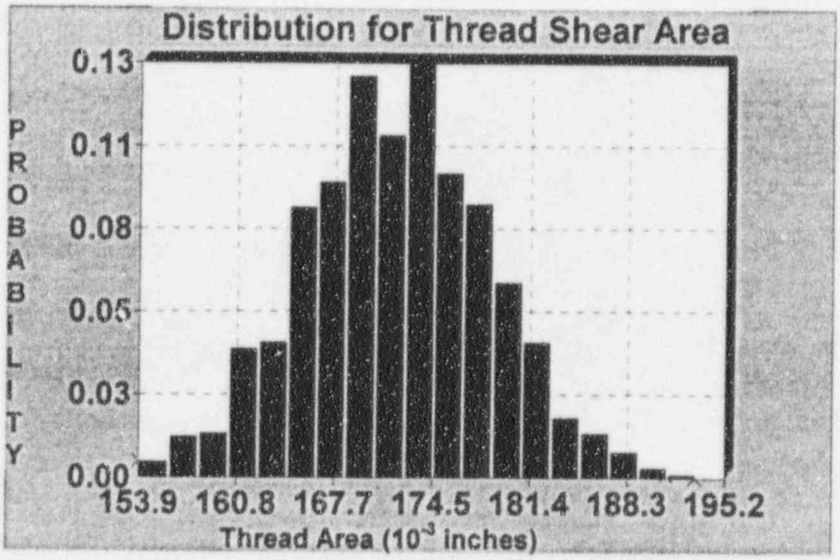


Figure A-10 shows the ratio of thread shear strength to the maximum preload. Table A-3 provides a summary of the Thread shear strength and ratio of thread shear strength to maximum preload for specific percentile levels of interest.

Figure A-10

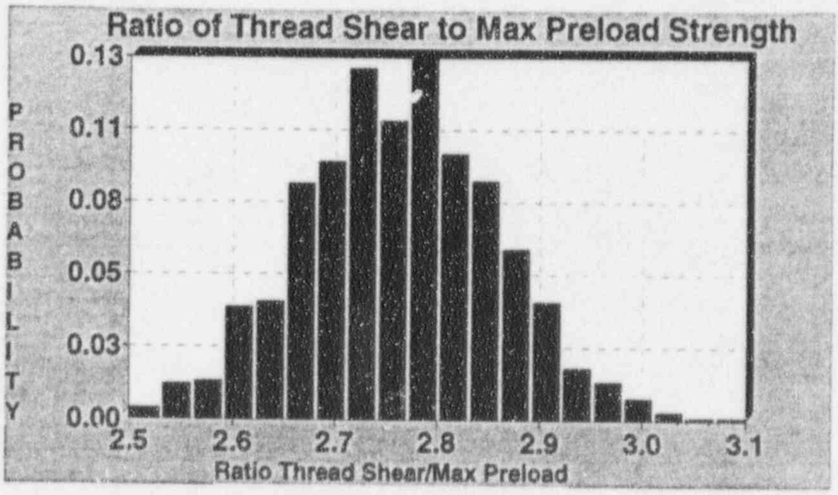


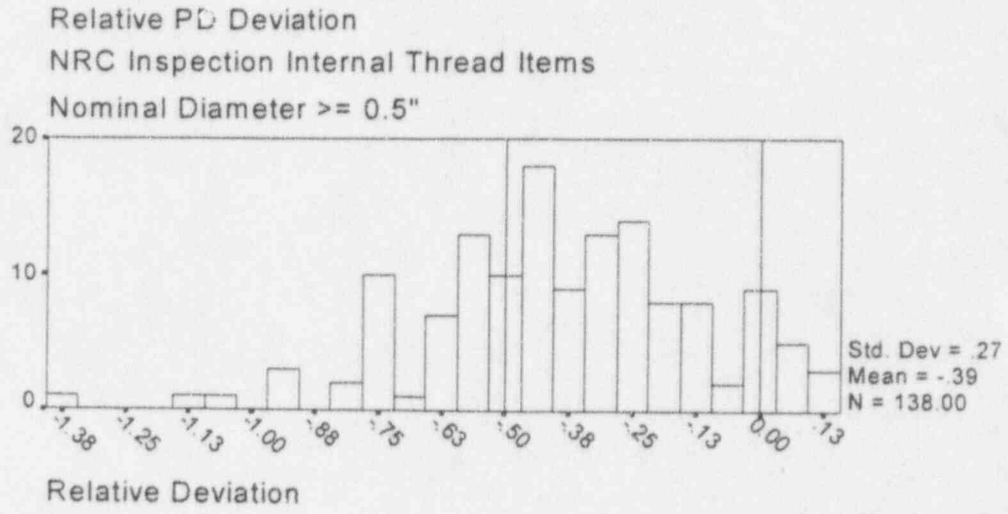
Table 3 - NRC Sample Data Set - 3/8 Cap Screw Data

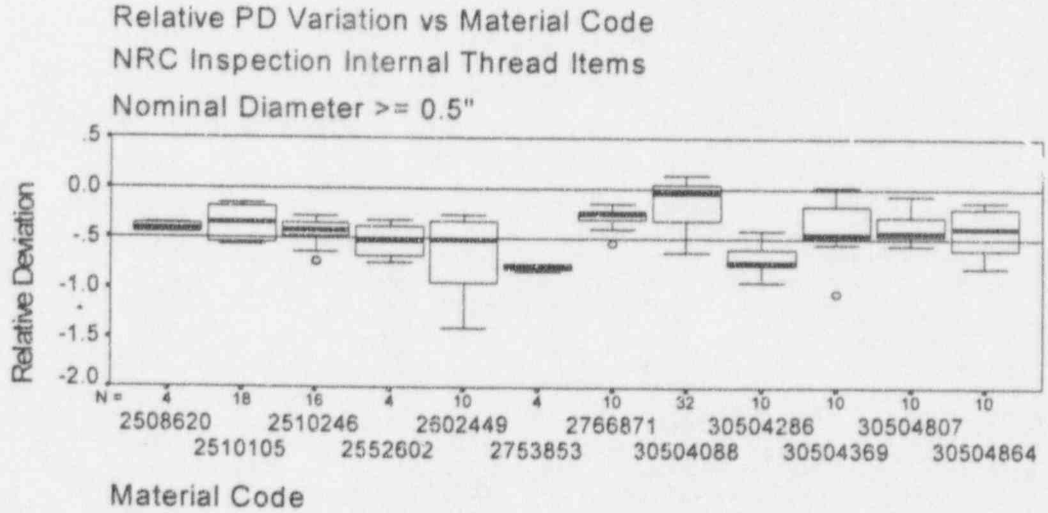
Percentile	Thread Shear Strength (lbs)	Ratio Thread Shear to Max Preload
5%	10067.33	2.598
50%	10749.95	2.775
95%	11402.42	2.943

The results show that even at the 5% percentile level the thread shear strength is considerable higher than the maximum bolt preload.

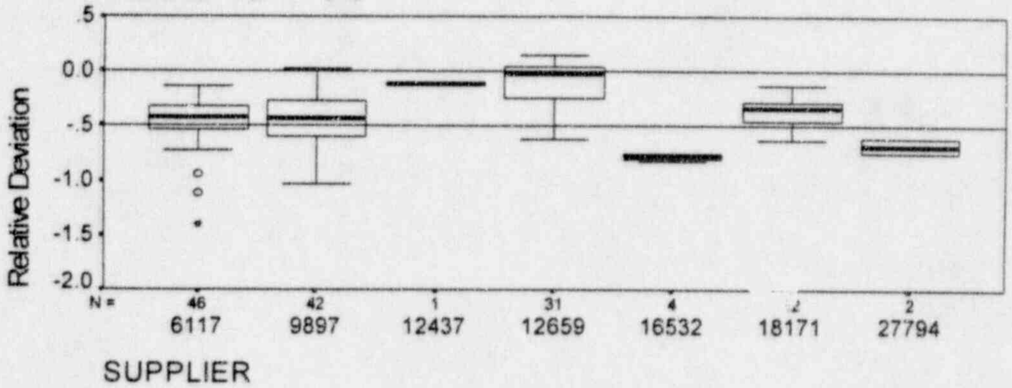
Appendix 2

NRC Data Set - Nominal Diameter > 0.5"

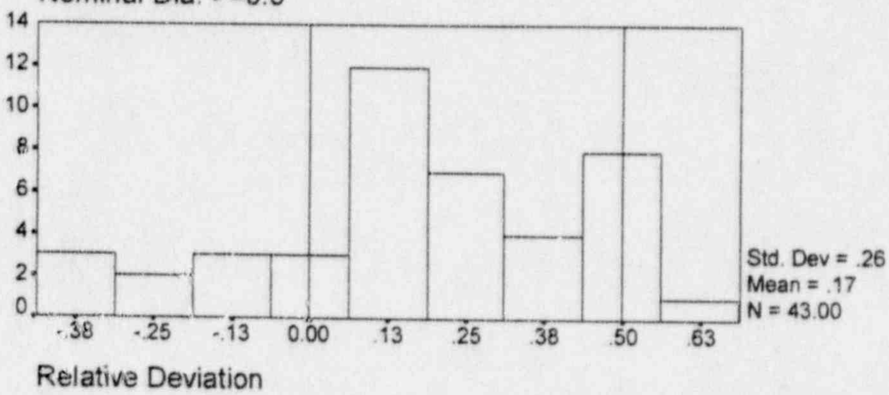




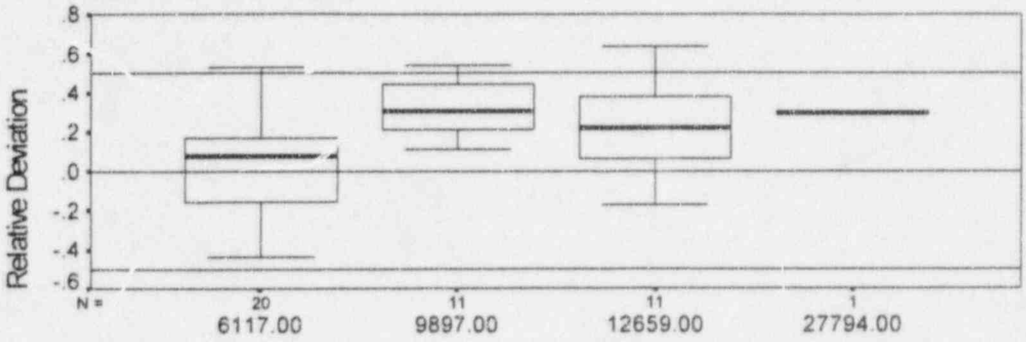
PD Variation vs. Supplier
Internal Thread Items NRC Data
Nominal PD >= 0.5"



Histogram of Relative PD Deviation
NRC Data External Thread Items
Nominal Dia. >=0.5

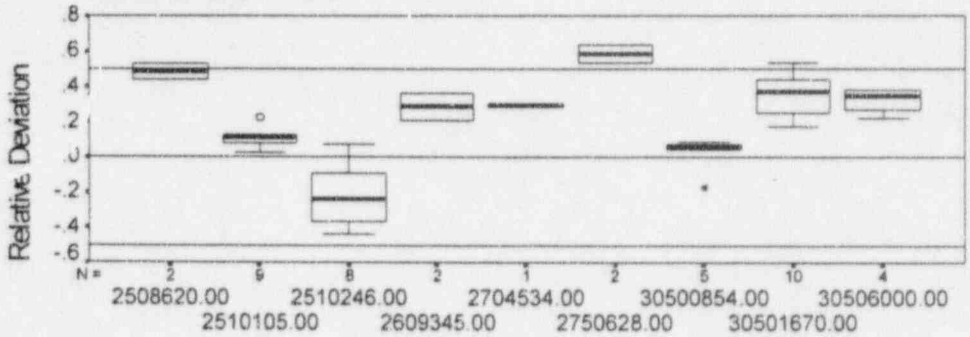


Variation of PD vs. Supplier
NRC Data External Thread Items
Nominal Dia. >=0.5"



SUPPLIER

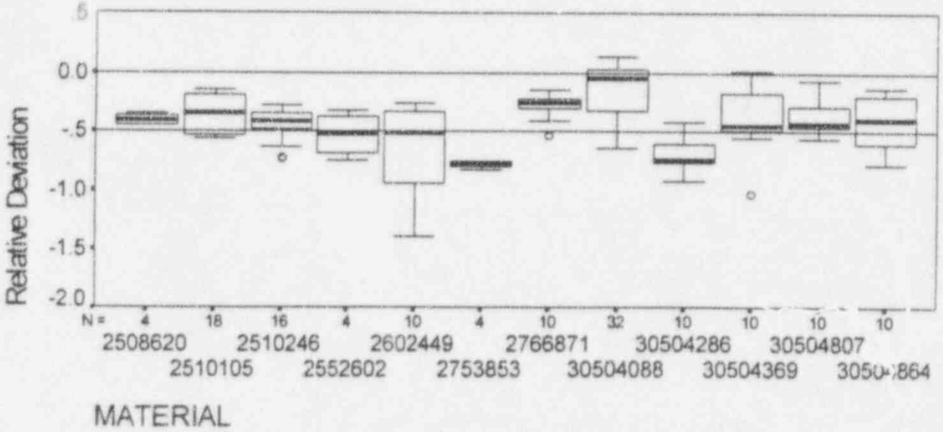
Variation of PD Vs. Material Code
NRC Data External Thread Items
Nominal Dia. >=0.5"



MATERIAL

NRC Sample Data Internal Thread Items

Nominal Diameter > 0.5"



NRC Sample Data External Thread Items

Nominal Diameter > 0.5"