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Use NC56 Connections on 8″ Drill Collars and Cut 1″ or ¾″ Pin Stress Relief Grooves on Rotated BHA Connections NC38 and Larger

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Abstract

When designing bottom-hole assemblies, the designer must specify both the type of connections and whether or not to use stress relief features. The relative fatigue lives of NC56 and 6-5/8 REG API connections were studied to compare the two connections in identical service. In addition, three common pin stress relief groove (SRG) widths were studied to determine the width that yields the best fatigue life. The relative magnitude of improvement in fatigue life was quantified using finite element analysis (FEA) and the Morrow strain-life model.

The combination of a more fatigue resistant thread form and increased box stiffness from a steeper taper gives the NC56 connection a dramatic fatigue advantage over the 6-5/8 REG connection. In addition, the results show the fatigue life of the pins modeled with ¾″ and 1″ SRGs to be the best, with the 1″ groove having a slight advantage. Pins cut with no groove can be expected to have the worst fatigue life, followed surprisingly closely by pins cut with the 1 ¼″ groove.

Introduction

Because fatigue historically accounts for roughly 80% of all drill string mechanical failures and the cost of a failure can be quite substantial, it is in the interest of the oil and gas industry to utilize the most fatigue resistant connections. Operators will benefit from decreased costs associated with fishing and tool replacement, and drilling contractors and rental companies will benefit from increased tool life. In order to determine which designs will be most fatigue resistant, and to quantify the benefits of a particular design, analyses must be conducted that compare available options on a relative basis.

Finite element analysis has been used for many years to determine the stresses present in complex models under multiple loads. Advances in FEA software and computer technology have allowed problems that previously required expensive supercomputers and days of computer run time to be solved with a desktop workstation and only several hours of computer time. As a result of these advances, problems that were run in the past with simplifying assumptions (in the interest of time and money) can be revisited in the pursuit of more accurate results. In addition, new topics previously deemed too complex to analyze can now be investigated.

Given the multiple loads involved (e.g. make-up torque, externally applied tension, bending, etc.), accurately analyzing the stress and strain distribution in a rotary shouldered connection is a challenging matter. In general, the areas that frequently require the most attention are (1) the mesh, (2) the "application" of make-up torque (MUT), (3) the application of a bending load, (4) the stress and strain output, and (5) the fatigue life model. The mesh, application of MUT, and application of bending load are discussed in detail in Appendix A. The stress and strain output and fatigue life model are discussed below as they distinguish these analyses from those previously conducted.

Stress and Strain Output

The stress distribution and maximum stress have traditionally been the means by which the suitability and performance characteristics of a model have been rated. When evaluating the performance of a rotary shouldered connection, however, it is sometimes necessary to focus on the strain instead of the stress, or both. Strain is significant because the stresses in the critical areas of the connection (thread roots and shoulder transition region) often exceed the yield strength of the material. Referring to the stress versus strain curve in Figure 1, it is evident that once the yield strength of the material is exceeded, the stress changes only slightly until the ultimate strength is reached, but the strain continues to increase significantly. Thus, if the point strain is plastic, the strain and strain amplitude are important drivers that should be considered when evaluating connection performance, not just stress.
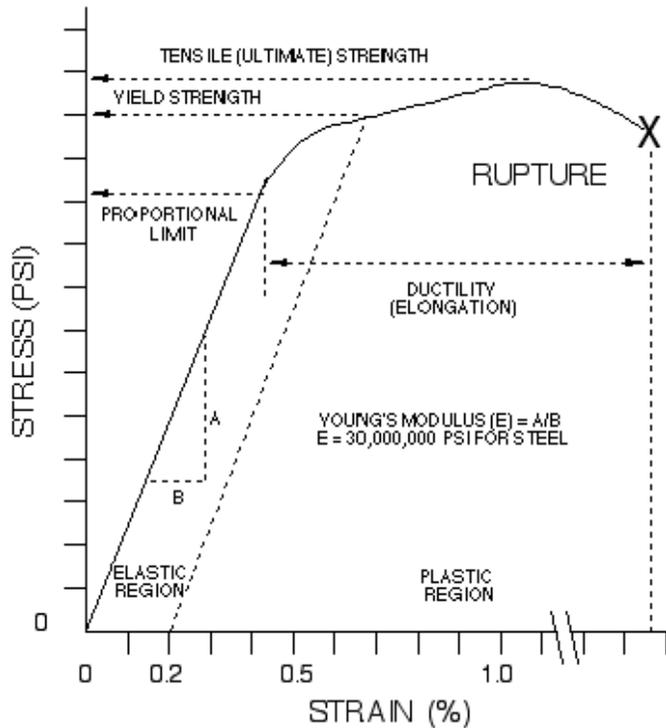
Figure 1 – Stress versus strain curve for typical drill stem tubular steel. After the yield strength of the material has been exceeded, stress increases only a small amount relative to the increase in strain.

Fatigue Performance and the Morrow Strain-Life Model

Absolute values of stress and strain are often used as measures of how much better one design is relative to another. When only considering overload or determining a factor of safety, this practice is acceptable in most instances. When evaluating differences in the fatigue performance of rotary shouldered connections, however, looking only at the magnitude of the stress or strain does not tell the whole story. An approximation of the fatigue life should be made to better understand the impact that both have on the performance of the connection. As an example, consider the two cases shown in Figure 2[1]. The difference in the magnitude of the stress amplitude between case 1 and case 2 is -20%, but the difference in fatigue life between case 1 and case 2 is 110%. Relatively small differences in stress and strain can have a dramatic impact on fatigue life. Therefore, it is necessary to consider the fatigue life rather that just the stress or strain when comparing design options.
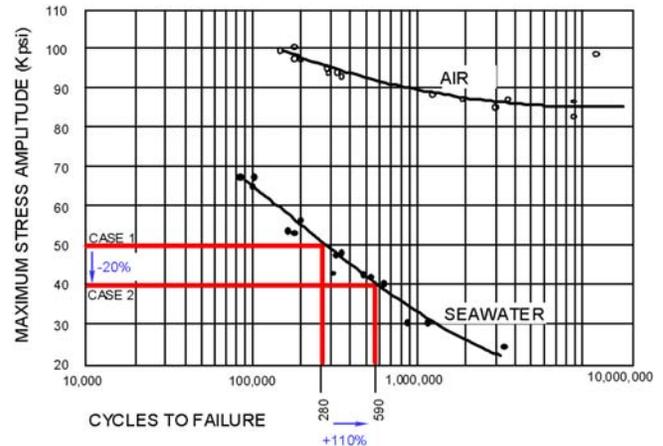

Figure 2 – Stress amplitude versus cycles to failure for typical drill stem steel. A small decrease in stress can net a substantial increase in fatigue life. Therefore, when making decisions about which design to use that concern fatigue, base decisions on relative fatigue life rather than relative stress or strain levels.

For the analyses to be discussed herein (taking plastic strain, stress, and fatigue into account), the Morrow strain-life model[2] is used. The Morrow strain-life model was chosen as it accounts for the fatigue crack initiation phase, not just the crack propagation phase. This is important as the connection geometry (thread design or stress relief groove width) heavily influences fatigue performance. Using a fatigue model such as the Forman model, which assumes an initial crack size, would not capture the effects of connection geometry. Equation 1 is the Morrow strain-life equation used to calculate fatigue life.

$$\varepsilon_a = \frac{\sigma'_f - \sigma_m}{E}(2N_f)^b + (\varepsilon'_f - \varepsilon_{mp})(2N_f)^c \ldots\ldots\ldots (1)$$

Where:

ε_a = Strain amplitude

σ'_f = Material fatigue constant determined experimentally

σ_m = Mean stress (psi), when $\sigma_m > \sigma_{mys}$, set $\sigma_m = \sigma_{mys}$

E = Young's Modulus (psi)
N_f = Number of cycles to failure
b = Material fatigue constant determined experimentally

ε'_f = Material fatigue constant determined experimentally

ε_{mp} = Mean plastic strain, when $\sigma_m \leq \sigma_{mys}$, set $\varepsilon_{mp} = 0$

c = Material fatigue constant determined experimentally

A quick study of this equation shows that if the mean stress in the critical area is above the yield strength of the material (100 KSI in the case of the drill collars used in these studies), the variables that drive the fatigue life calculation are strain amplitude and mean plastic strain. Therefore, the model with the highest combination of strain amplitude and mean plastic strain will have the lowest fatigue life. If the mean stress in the critical area is less than the yield strength of the material, the strain amplitude and mean stress are the variables that drive the fatigue life calculation. In these cases, the model

with the highest combination of strain amplitude and mean stress will have the lowest fatigue life.

## Fatigue life comparison: NC56 versus 6-5/8 REG connections

Regular connections have been cut on down-hole tools for many years. In 1968, the API introduced a series of numbered connections (NC), whose fundamental differences from those that existed at the time were the naming convention and the connections' thread design. The numbers that follow the NC designation represent the first two numbers of the pitch diameter of the connection. For example, an NC38 connection has a 3.808″ pitch diameter. All NC connections were specified with the V-0.038R thread form. Despite the fact that the NC connections were touted as better fatigue performers, other connection types (mainly REG on larger collar sizes) continue to be cut on BHA components.

Three distinct differences between the NC56 and 6-5/8 REG connections are shown in Table 1.

Table 1-NC56 and 6-5/8 REG Connection Dimensions

	Thread Root Radius	Thread Taper	Pitch Diameter
NC56	0.038 in.	3 in./ft.	5.6160 in.
6-5/8 REG	0.025 in.	2 in./ft.	5.7578 in.

The dimensional differences may appear to be relatively inconsequential. However, a drawing of the two connections brings into focus just how different these connections are, and how these differences ultimately impact fatigue performance. Figures 3 and 4 illustrate this point.
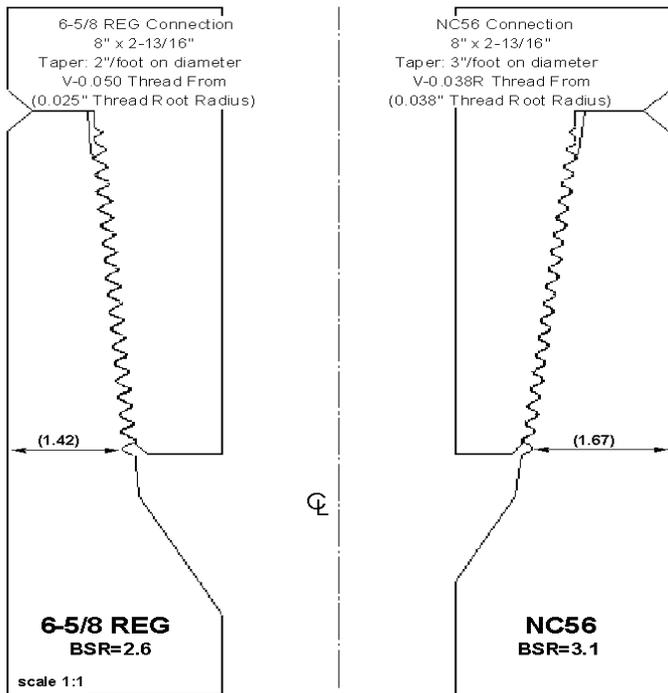
Figure 3 – Dimensional differences between 6-5/8 REG and NC56 connections. Differences in thread taper and root radius drive the difference in box fatigue life. (The difference in thread root radius can be seen more clearly in figure 4.)

The larger root radius of the NC56 connection translates into a lower stress concentration factor, all else equal. Its 3″/′ taper yields more cross sectional area in the back of the box and a smaller "notch" when compared with the 2″/′ taper of the 6-5/8 REG. This is advantageous because most of the bending the connection experiences will occur in this area. A larger cross-sectional area means less bending for a given load, lower stresses and strains, and longer fatigue life.

To quantify the fatigue advantage a NC56 connection has over a 6-5/8 REG, the common 8″ OD x 2 13/16″ ID drill collar was selected. Two dimensional axis-symmetric models were made of each connection type. The API recommended MUTs (49,153 ft-lbs for NC56 and 53,934 ft-lbs for 6-5/8 REG) were applied to the models along with a bending load that induced an equivalent connection curvature of 6°/100′.

Higher stress and strain amplitudes coupled with exposure to corrosive drilling fluids dictate that fatigue failures on large drill collars will occur more frequently in or near the last engaged thread of the box as opposed to the pin. This area should be used to compare relative differences in fatigue life. Table 2 contains a summary of the stress and strain data generated by the finite element analyses. Figure 4 shows in detail the resultant effect of the dimensional differences between the NC56 and 6-5/8 REG connections.

### Table 2-NC56 and 6-5/8 REG Stress and Strain Data

	Max Stress (PSI)	Mean Stress (PSI)	Stress Amplitude (PSI)	Max Strain	Mean Strain	Strain Amplitude
NC56	103,755*	73,471*	30,284*	0.003152*	0.002213*	0.000940*
6-5/8 REG	123,005	80,935	42,070	0.004035	0.002739	0.001296

Numbers in red are values critical in determining fatigue life.
Numbers with * indicate the lowest value in the set (best for fatigue)
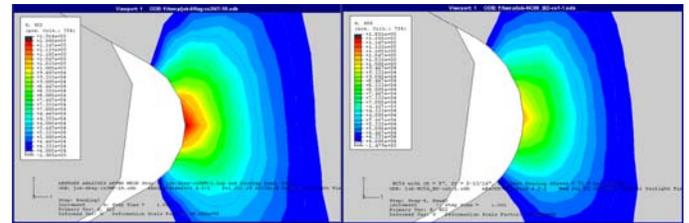
Figure 4 – Maximum stress plot of the last engaged 6-5/8 REG box thread root (left) and NC56 box thread root (right). Note the distinct difference in the magnitude of the stress concentration in the 6-5/8 REG box thread root relative to the NC56 box thread root (120 ksi (red) / 100 ksi (yellow) = 1.2).

The NC56 connection clearly has much lower stress in the last engaged box thread root. Lower stress translates into longer fatigue life, all else equal. To quantify the relative difference in fatigue life, the stress and strain data from the FEA models were used in equation 1. Since the mean stress in the critical area is less than the material yield stress, the strain amplitude and mean stress drive the fatigue life calculations. The resulting relative fatigue lives of the boxes are shown in Figure 5.

NC56 vs. 6-5/8 Reg
Relative Box Fatigue Life at 6 deg / 100 ft DLS
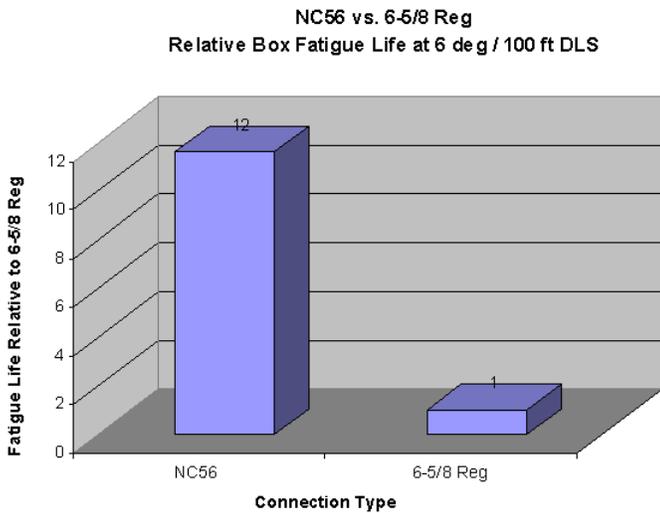
Figure 5 - Relative fatigue life. Although not absolute, the predicted relative fatigue life yields a general understanding of which connection has the best fatigue performance. The NC56 connection has superior fatigue performance compared with the 6-5/8 REG connection under the same load conditions.

Based on this analysis, the NC56 connection has a distinct fatigue performance advantage when compared with the 6-5/8 REG connection under the same loading conditions.

Fatigue life comparison: Pin stress relief grooves (no groove, 3/4˝, 1˝, 1 1/4˝)

Stress relief features have been employed on BHA connections of various types and sizes for many years. These features include a stress relief groove on the pin (SRG) and a boreback box (BBB), and are typically cut on NC38 and larger connections. The purposes of stress relief features are (1) to remove unengaged threads that act as stress concentrators near the pin shoulder and in the back of the box, and (2) to provide flexibility in the connection outside of the threaded area so bending occurs across the grooved areas and not in the threaded areas. Employing stress relief features can have a dramatic positive impact on the fatigue life of BHA connections.

Analyses have been conducted in the past to study the effect different pin stress relief groove widths have on the fatigue performance of rotary shouldered connections. Previous analyses on this topic that the authors are familiar with have been conducted with axially applied tensile approximations of MUT, assume linear-elastic material behavior, and do not include actual bending loads[3]. In addition, the stress state alone was used as the means to measure the relative differences between groove widths. For the current analyses, actual bending loads were applied to better simulate down-hole conditions (build or drop, dogleg, or buckling) and elastic-plastic material behavior was recognized. The NC56 connection with a box outer diameter of 8˝ and a pin inner diameter of 2-13/16˝ was selected for this study. Four different NC56 connection models were generated. Geometrically, only the width of the pin stress relief groove varied between models. The connection models included no groove, 3/4˝ groove, 1˝ groove, and 1 1/4˝ inch groove. The loads applied to each model included the API recommended MUT (49,153 ft-lbs for no groove, 45,547 ft-lbs

for cases with a stress relief groove) and a bending load that induced an equivalent connection curvature of 6°/100´. Table 3 contains a summary of the stress and strain data generated by the finite element analyses.

Table 3 – NC56 Stress and Strain Data

	Max Stress (PSI)	Mean Stress (PSI)	Stress Amplitude (PSI)	Max Strain	Mean Strain	Strain Amplitude
No Groove	139,529	116,014	23,515	0.019159	0.018429	0.000731
3/4˝ Groove	128,522	114,157	14,365	0.007436*	0.006980*	0.000457
1˝ Groove	125,692*	111,859*	13,833*	0.007669	0.007222	0.000447*
1 1/4˝ Groove	140,161	122,159	18,003	0.010959	0.010959	0.000562

Numbers in red are values critical in determining fatigue life.
Numbers with * indicate the lowest value in the set (best for fatigue)

The 1˝ groove and 3/4˝ groove display nearly the same magnitudes of mean strain and strain amplitude. The model with no groove has relatively high strain followed closely by the model with the 1 1/4˝ groove. The strain contours resulting from MUT and bending are shown in Figure 6.
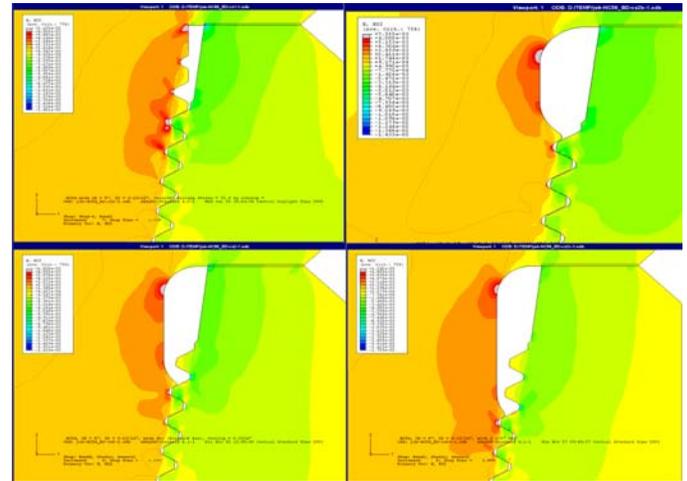
Figure 6 – No groove (top left), 3/4˝ groove (top right), 1˝ groove (bottom left), 1 1/4˝ groove. Note the axial strain is the highest (gray = plastic) in the last engaged thread roots, unengaged thread roots, and in the transition near the shoulder.

Instead of using stress alone as the basis of comparison, the stresses and strains generated from the FEA were used in equation 1 to calculate the fatigue life of each connection. The calculated fatigue lives of the models with stress relief grooves relative to the model with no SRG are somewhat surprising. Since the mean stress in the critical regions of the models is above the material yield strength, the strain amplitude and mean plastic strain drive the calculated fatigue lives. Figure 7 displays the magnitude of the relative fatigue lives of each of the grooved connections relative to the connection with no stress relief groove.

Relative Fatigue Life vs. Stress Relief Groove Width
NC56 in a 6 deg / 100 ft dogleg
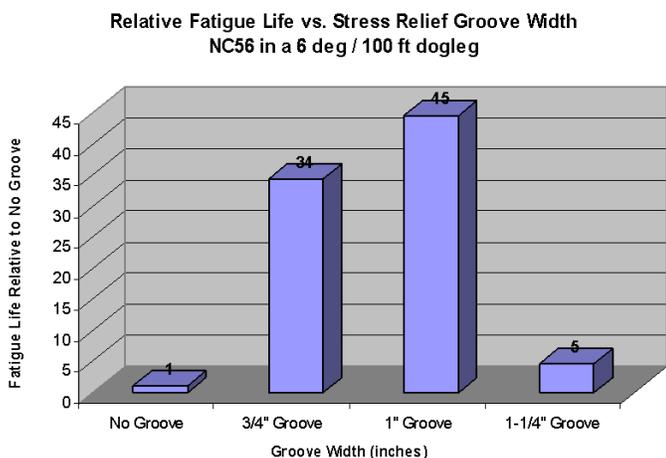
Figure 7 – Relative fatigue life. Although not absolute, the predicted relative fatigue lives yield a general understanding of which groove widths perform best. The 3/4″ groove and 1″ groove are clearly superior to no groove and 1 1/4″ groove.

As a check, a similar analysis was conducted using a 6-5/8 REG connection with the same outside and inside diameters. API recommended MUT and a 6°/100′ equivalent dogleg were applied to each model. The trend and magnitude of the relative fatigue lives calculated for the SRG configurations closely matched the analyses completed on the NC56 connections.

Based on the results of these analyses, cutting stress relief grooves at 3/4″ +9/32″, -1/32″ will yield the best fatigue performance.

Summary

- Choose the NC56 connection over the 6-5/8 REG connection on 8″ components for better fatigue performance. Larger area and a smaller "notch" in the back of the NC56 box, coupled with a larger thread root radius drive the improved fatigue performance.
- Cut 3/4″ +9/32″, -1/32″ stress relief grooves on BHA connection sizes NC38 and larger. The 3/4″ and 1″ grooves yield similar fatigue results, but are far superior to connections cut with no groove or a 1 1/4″ groove.
- When making relative comparisons between connections, it is important to consider strain since the material near the thread roots (where fatigue cracks form) is plastic due to MUT and externally applied loads. Use a fatigue life model to make comparisons, as just comparing the magnitude of stress and strain does not accurately represent the difference in fatigue performance between connections.

References

1. Rollins, H.M., "Drill Pipe Fatigue Failure", *Oil and Gas Journal*, 1966
2. Dowling, Norman E., <u>Mechanical Behavior of Materials: Engineering Methods for Deformation, Fracture, and Fatigue.</u> Prentice Hall, 1993, pp. 621-637.
3. API RP7G, "Recommended Practice for Drill Stem Design and Operating Limits", Sixteenth Edition, American Petroleum Institute, December 1, 1998.

APPENDIX A

Mesh

An example of the type and size of mesh used is shown in Figure A1. Each model that is evaluated is "broken up" into discrete pieces or "elements." The mesh is the lattice that is the sum total of all of the elements. In order to accurately describe the geometry and shape of a model, the elements must be small enough to closely approximate the contours present in the specific area(s) of interest. In the case of a rotary shouldered connection, the threads and the stress relief features are the prime areas of interest. Therefore, it is important to have a dense mesh in these regions. Regions of the connection away from these critical areas may have a coarser mesh as the stresses and strains here are generally not of much interest.
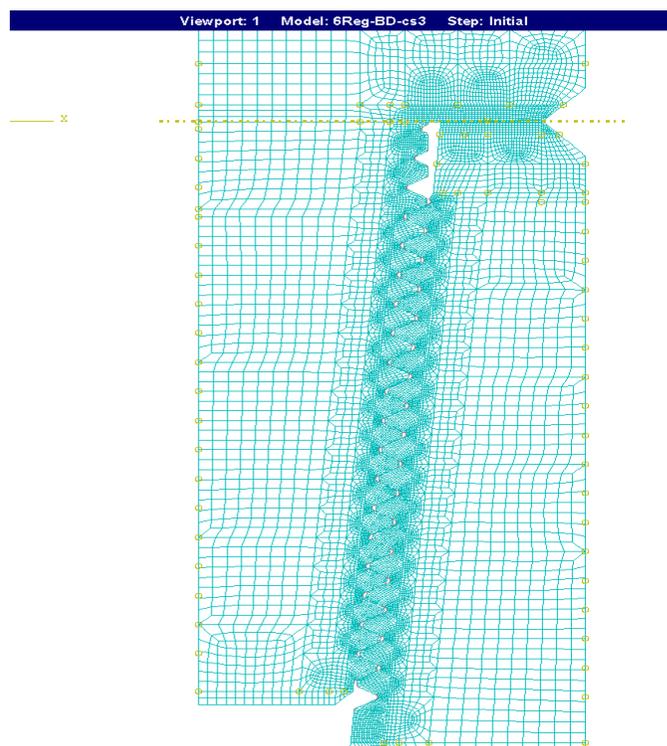

Figure A1 - 6-5/8 REG axis symmetric connection model. Smaller elements are necessary in the regions of interest, while coarser mesh is allowable in non-critical areas.

Application of MUT

Applying torque to a computer generated model of a rotary shouldered connection is much more difficult than loading an actual connection into an iron roughneck and pressing the "GO" button. Successfully modeling a three-dimensional connection (getting a convergent solution) can be a daunting task because of the amount of sliding contact between the thread flanks, the helical nature of the thread path, and the boundary conditions. As a result, rotary shouldered connections are most easily modeled in two dimensions. Two dimensional axis-symmetric models with asymmetric

(bending) loads must be created. MUT is approximated by either assuming a certain amount of thread and shoulder overlap, or by applying a thermal gradient (to expand the material thereby creating thread flank and shoulder interference) to the pin and/or box. The bulk stresses induced from applying MUT to an actual connection are known or calculable in three key areas (Figure A2). Using these known or calculable values, correctly approximating the MUT on the two dimensional connection model can be achieved by adjusting the overlap or thermal layer temperature, and comparing the resultant stresses to those defined to be present in an actual connection of the same type and dimensions.
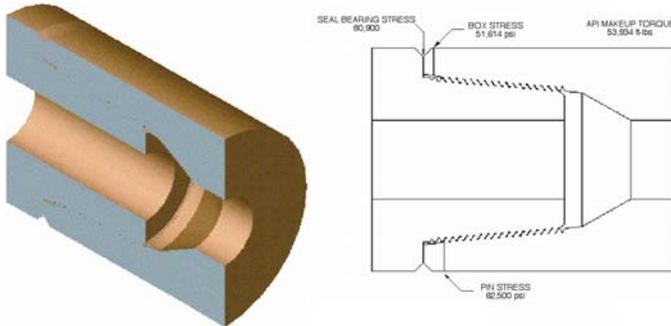
Figure A2 – 6-5/8 REG 8˝ x 2-13/16˝ rotary shouldered connection critical stress areas: Seal area, 3/4˝ from shoulder in pin, 3/8˝ from shoulder in box.

Bending Load

Axial loads and pressure are generally simple to apply. Applying a bending load to a two dimensional axis-symmetric asymmetric model generally requires the use of a special element type. When modeling bending in a rotary shouldered connection, it is important to bend the connection multiple times in order to generate the best approximation of the stress state at the thread roots. During connection make up, the last few engaged pin threads, any remaining unengaged pin threads, and some areas of a stress relief groove (if present) are usually plastically deformed. Since the stresses are plastic, bending the connection once will not give accurate stress and strain values. The connection must be bent multiple times to allow for the effects of stress relieving and stress redistribution. In an actual connection made up and run down hole, this will occur once the connection is rotated more than one revolution while bent or buckled.