YOUR RODS ARE OVERLOADED
Compound Rod Stresses in Deviated Wells

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ABSTRACT & INTRODUCTION

Sucker rods are simple in form and function; however, they operate in a sophisticated engineered system over great lengths without direct visibility. Because of this, we as engineers must do our best in predictive efforts to provide the best configuration of sucker rods and the rod string design for a highly dynamic system.

Rod string design software utilizes complex math to compute stress loading throughout the system. The rod string is constantly moving, and experiences variable loads and forces throughout each cycle. Design software with deviation included incorporates well-bore geometry to estimate ancillary loads throughout the design, i.e.: side-load & drag load from rod-on-tubing contact.

An improved method of accurately evaluating system and rod string stresses has been developed and computed. This new method combines the industry standard Gibbs wave equation and trusted rod-string loading computation as its base. Taking it a bit further, the evaluation of the deflection behavior of sucker rods throughout the deviated wellbores is incorporated, which computes additional stresses based on arc deflection.

In instances of deviation and molded rod guides, sucker rod behavior changes dramatically. Dog-leg severity combined with *most* traditional molded rod guides creates intermittent sucker rod rigidity and assumed perfect alignment with the tubing. This then causes the ability of increased bending moments and bending stresses during moments of compression and sucker rod instability, leading to pre-mature failure. Fiber loading[1] around the circumference of the sucker rod is then increased, leading to regions of the rod body exterior being subjected to stresses far higher than intended.

By accurately computing bending stresses from deviation and adding them to the Modified Goodman stress computation in rod string design software, loading through the deviated well bore and accuracy in the predictive system is increased, creating additional awareness to the system on a detailed, per rod level.

Despite general assumptions, peak stress loading (due to deviation) is not linear throughout each sucker rod taper. Non-linear loading is understood but not quantified, visualized, or addressed in string design software outputs. The effort of this paper is to provide detailed engineering awareness to rod loading, openly discussing with operators

[1]Fiber loading as an engineering term. This is not a reference to fiberglass sucker rod.

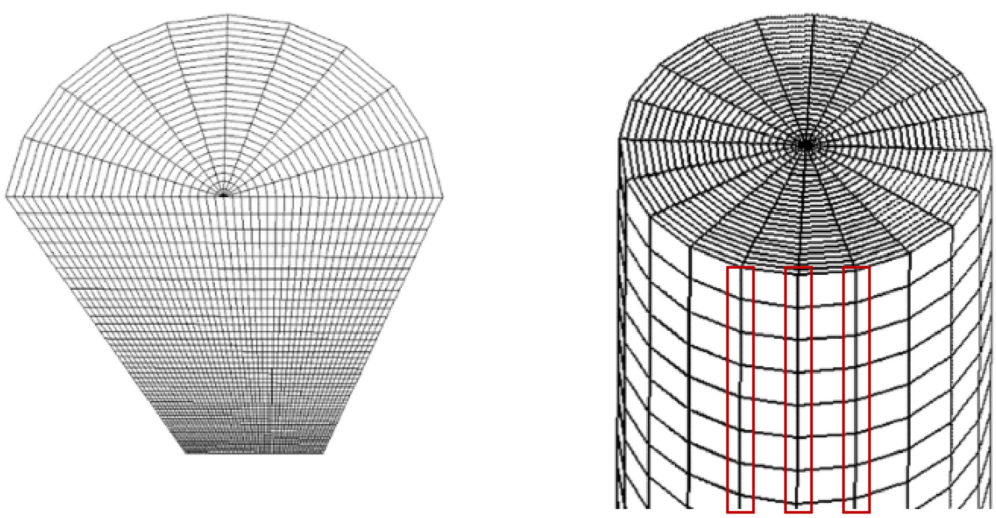
the effects of minor rod deflection and its associated stresses, and how deviation can encourage pre-mature fatigue fractures.

IMPROVED ANALYSIS

Sucker rods encounter various bending stresses in conjunction with tensile loading stresses. Current software in the market does not account for bending stresses in the system. Side-load is typically the main concern when deviation occurs, however, the bending stresses *cannot* be ignored, as they should be compounded with tensile stresses to properly design the rod string system.

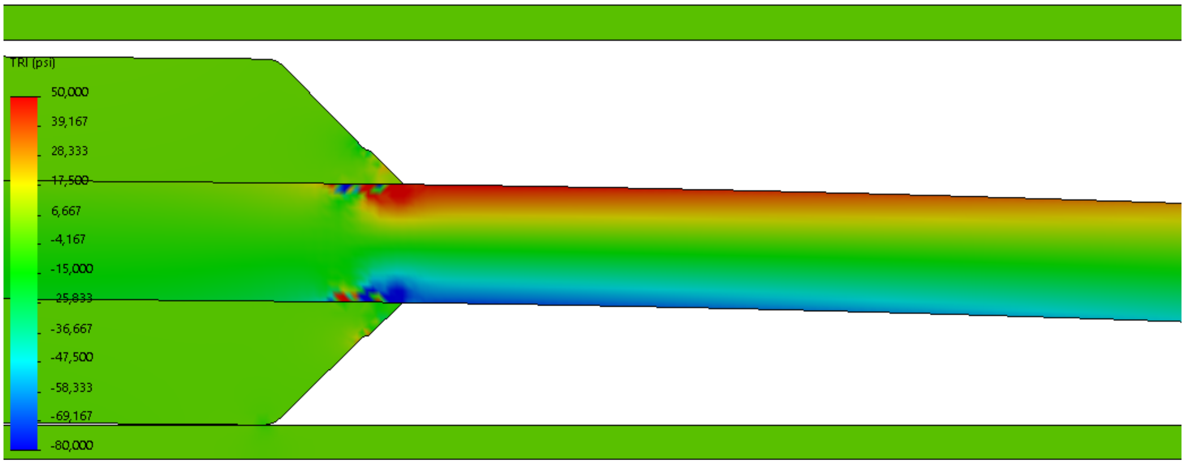
Tensile stresses are basic and computed via general loading of the string weight, acceleration, and fluid load throughout the wave equation.

Although existing software assumes stress loading is consistent along the cross section of the rod, bending stresses create localized 'fiber loading' at different areas of the outer surface of the sucker rod. These bending stresses combined with tensile loading can lead to excessive loading, yielding, and ultimately, micro-fractures which cause pre-mature rod failure.



Nodal analysis of round bar, FEA.

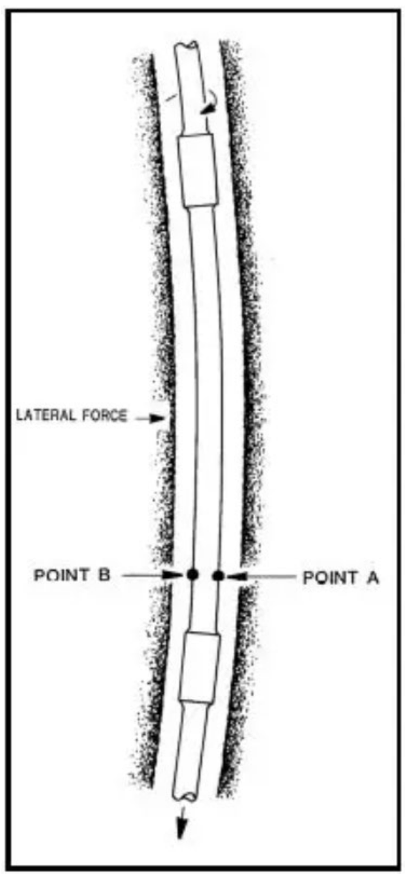
This 'fiber loading' can be visualized through a bending moment at the edge of a rod guide during buckling. In the below graphic, non-uniform stress loading is displayed due to a bending moment at the edge of traditional rod guides during sucker rod buckling. The red represents tensile stresses, and the blue represents compressive stresses. Green, along the neutral axis and plane of the sucker rod, has minimal bending stress.



Bending stresses along a sucker rod.

Deviated well-bores lead to rod deflection and bending stresses throughout the entire. These must be accounted for in string design or accurate computation.

Bending stresses are commonly analyzed in drilling engineering, too.



Bending drill pipe.

As explained on DrillingManual.com:

> [The figure] illustrates how a severe dogleg can cause fatigue failures. Point “A” on the drill pipe is in maximum tension while point “B” is in minimum tension due to bending. If there is no weight hanging below the joint of drill pipe, point “A” would be in tension, and point “B” would be in compression.) As the pipe is rotated, the reference points go through cyclic stress reversals. Point “A” goes from maximum tension to minimum tension and back to maximum tension on each cycle. These cyclic stress reversals will cause fatigue failures.

To compute bending stresses in the system, an assumption is made from DLS/100’ that the rod’s arcing is consistent throughout the 25-foot rod, based on 1/4th (25 feet) of the DLS/100’ value.

INSERT GRAPHIC OF DLS/100’ v DLS/25’.

A rod that is positioned throughout DLS based on static position and the stroke length shall be assumed as the highest DLS in the path region to compute the worst-case bending stresses.

VALIDATION OF EXISTING SOFTWARE LIMITATIONS

A study was completed utilizing common rod string design software in industry. The requirement of this comparative study is to limit the stress-loading computation to two variables.

1. Straight Hole (zero rod deflection)
2. Deviated Hole (rod arc deflection through DLS)

A string analysis was conducted in ‘deviated’ mode with a customer provided survey. The well features plenty of DLS throughout the design. In order to eliminate the factor of friction forces from straight hole versus deviated, a friction coefficient of 1 E-9 was utilized to get as close to zero as possible.

Rod string design					Rod string stress analysis (service factor: 1)				
Diameter (in)	Rod Grade	Length (ft)	Min. Ten. Str. (psi)	Fric. Coeff	Stress Load %	Top Maximum Stress (psi)	Top Minimum Stress (psi)	Bot. Minimum Stress (psi)	# Guides/Rod
1	DS (T/2.8)	2625	115000	1E-	83.9%	40728	13166	7207	0
0.875	DS (T/2.8)	2550	115000	1E-	83.9%	38622	8786	4284	0
0.75	DS (T/2.8)	2176	115000	1E-	83.7%	36443	4358	1453	0
1	DS (T/2.8)	1500	115000	1E-	33.3%	13474	-266	-764	0

NOTE: Displayed bottom minimum stress calculations do not include buoyancy effects (top minimum and maximum stresses always include buoyancy).

A secondary analysis was completed in deviated mode, utilizing a survey which assumes a perfectly straight well-bore, also with a friction coefficient plugged in at 1E -9, like before.

Rod string design					Rod string stress analysis (service factor: 1)				
Diameter (in)	Rod Grade	Length (ft)	Min. Ten. Str. (psi)	Fric. Coeff	Stress Load %	Top Maximum Stress (psi)	Top Minimum Stress (psi)	Bot. Minimum Stress (psi)	# Guides/Rod
1	DS (T/2.8)	2625	115000	1E-	84.3%	40923	13298	7297	0
0.875	DS (T/2.8)	2550	115000	1E-	84.3%	38837	8900	4287	0
0.75	DS (T/2.8)	2176	115000	1E-	83.9%	36538	4350	1455	0
1	DS (T/2.8)	1500	115000	1E-	33.4%	13518	-272	-764	0

NOTE: Displayed bottom minimum stress calculations do not include buoyancy effects (top minimum and maximum stresses always include buoyancy).

Stress loading computation was maintained with a variance of max 0.4%. This is not reflective of bending stress analysis on the individual sucker rods as it relates to DLS.

REACTION FORCES OF VARIOUS SUCKER RODS

By doing mathematical analysis of known changes, linear deflection analysis can be utilized to understand various sucker rod behaviors and the resultant stresses from specific deflection and deviation.

This is the correct approach to understanding rod string dynamics. The rods, regardless of what diameter, are following the path of the production tubing. For this analysis, rods are perceived to be following the central axis of the tubing, i.e.: guided rods. Doing so is a fair assumption because rod guides and centralizers are utilized in times of need for deviated sections in the well-path.

A table is computed below showing common deflection across a 72” section of sucker rod.

Diameter	**Material**	**Modulus, E**	**AMOI**	**Bending Stiffness**	**Force to Deflect, lbs** ***1" Deflection at 72"***	**Bending Stress, psi** ***1" Deflection at 72"***
0.75	Steel	30,500,000	0.0155	473,712	3.81 lbf	6,619 psi
0.875	Steel	30,500,000	0.0288	877,611	7.05 lbf	7,722 psi
1	Steel	30,500,000	0.0491	1,497,165	12.03 lbf	8,825 psi
1	Fiberglass	7,200,000	0.0491	353,429	2.84 lbf	2,083 psi
1.25	Fiberglass	7,200,000	0.1198	862,864	6.94 lbf	2,604 psi

Commonly understood by industry, fiberglass sucker rods are more flexible than steel sucker rods. This is due to their reduced modulus of elasticity. Additionally, the stress put on the fiberglass sucker rods through deflection (deviation) is much less because of their ease of flexibility despite the larger diameter. The reaction force of deflection is far less because of the modulus reduction.

HUNGER FOR UNDERSTANDING

As mechanical and production engineers for beam lift systems, there is a common goal for putting extensive engineering mathematics in rod string design for accurate prediction. Well bores are more complicated than ever, and it is imperative that we use best practices for selecting the correct diameter and grade of sucker rod in application. This extends now to material selection too, fiberglass and steel sucker rods.

Sucker rods in string design software are analyzed at the per-rod level.

A 3-degree dogleg shows to add the following bending stress to various type of sucker rod. Recall that the grade/hardness of sucker rod does not dictate flexibility or bending stresses. Deflection, modulus of elasticity and diameter are the only driver of bending stress computation.

Diameter	**Material**	**Modulus, E**	**AMOI**	**EI (Bending Stiffness)**	**Force to Deflect, lbs** ***3 deg DLS/100', 25' section length***	**Bending Stress, psi** ***3 deg DLS/100', 25' section length***
0.75	Steel	30,500,000	0.0155	473,712	0.21 lbf	1,498 psi
0.875	Steel	30,500,000	0.0288	877,611	0.38 lbf	1,748 psi
1	Steel	30,500,000	0.0491	1,497,165	0.65 lbf	1,998 psi
1.25	Steel	30,500,000	0.1198	3,655,189	1.60 lbf	2,497 psi
1.5	Steel	30,500,000	0.2485	7,579,399	3.31 lbf	2,997 psi
1	Fiberglass	7,200,000	0.0491	353,429	0.15 lbf	472 psi
1.25	Fiberglass	7,200,000	0.1198	862,864	0.38 lbf	590 psi

Larger diameter sucker rods are stronger than smaller diameter rods due to their cross-sectional area. The lifting load can be increased. However, the bending stress factor also increases because the rod is larger in diameter and resists deflection. Fiberglass sucker rods are a suitable replacement, if appliable to the needs of the well, to reduce bending stresses but still increase lifting load capability. Fiberglass sucker rods must always be kept in tension for successful and acceptable run times.

Relative to the Modified Goodman Diagram for sucker rod loading, these bending stresses are significant, even more so, when more extreme dog-legs are pumped through.

The T/2.8 method for acceptable stress loading is a widespread method for 25-foot steel sucker rods of all grades.

$$S_{\max allowed} = \left(\frac{UTS\ of\ Rod_{min}}{2.8}\right) + ((0.375)(S_{min}))$$

Fiber loading, the outside diameter peak stress in a bending situation, is ADDITIONAL tensile stress put on that exterior region of the sucker rod round bar profile. Accounting for this prior to the computation of the Modified Goodman Stress loading percentage is necessary for accurate Stress Load computation.

In the example used prior, the top sucker rod of the 7/8” DS (T/2.8) taper was loaded as follows:

Rod string design					Rod string stress analysis (service factor: 1)				
Diameter (in)	Rod Grade	Length (ft)	Min. Ten. Str. (psi)	Fric. Coeff	Stress Load %	Top Maximum Stress (psi)	Top Minimum Stress (psi)	Bot. Minimum Stress (psi)	# Guides/Rod
1	DS (T/2.8)	2625	115000	1E-	84.3%	40923	13298	7297	0
0.875	DS (T/2.8)	2550	115000	1E-	84.3%	38837	8900	4287	0
0.75	DS (T/2.8)	2176	115000	1E-	83.9%	36538	4350	1455	0
1	DS (T/2.8)	1500	115000	1E-	33.4%	13518	-272	-764	0

NOTE: Displayed bottom minimum stress calculations do not include buoyancy effects (top minimum and maximum stresses always include buoyancy).

Top Maximum Stress: 38,837 psi
Top Minimum Stress: 8,900 psi
UTS of Rod: 115,000 psi
Stress Load: 84.3%

Adding 3-degree DLS/100’ bending stress to the equation:

Top Maximum Stress: 38,837 psi + 1,748 psi = 40,585 psi
Top Minimum Stress: 8,900 psi + 1,748 psi = 10,648 psi
UTS of Rod: 115,000 psi
Stress Load: 87.0%.

$$S_{\max allowed} = \left(\frac{UTS\ of\ Rod_{min}}{2.8}\right) + ((0.375)(S_{min}))$$

$$S_{\max allowed} = \left(\frac{115{,}000}{2.8}\right) + ((0.375)(10{,}648))$$

$$S_{\max allowed} = 41{,}071 + 3{,}993 = 45{,}064\ psi$$

$$Goodman\ Stress\ Loading, \% = \left(\frac{S_{max} - S_{min}}{S_{\max allowed} - S_{min}}\right) * 100$$

$$Goodman\ Stress\ Loading, \% = \left(\frac{40{,}585 - 10{,}648}{45{,}064 - 10{,}648}\right) * 100$$

$$Goodman\ Stress\ Loading, \% = \left(\frac{29{,}937}{34{,}416}\right) * 100$$

$$Goodman\ Stress\ Loading = 87.0\%$$

Although this example seems marginal, the difference is enough to matter in string design and understanding pre-mature fatigue failure.

A recent string design was reviewed with a major operator. The operator elected to utilize a 1.125" co-rod section through a 5-degree dogleg in a high kick.

Rod string design					Rod string stress analysis (service factor: 1)				
Diameter (in)	Rod Grade	Length (ft)	Min. Ten. Str. (psi)	Fric. Coeff	Stress Load %	Top Maximum Stress (psi)	Top Minimum Stress (psi)	Bot. Minimum Stress (psi)	# Guides/Rod
1	WFT HD	100	140000	0.2	85.3%	49972	15654	15949	0
1.125	ERod ND	2450	115000	0.1	116.1%	39488	12438	7043	0
1	ERod ND	2550	115000	0.1	113.1%	36626	8125	4309	0
0.875	ERod ND	2000	115000	0.1	105.1%	32664	4547	1373	0
1	ERod ND	1751	115000	0.1	59.7%	16879	-377	-764	0

NOTE: Displayed bottom minimum stress calculations do not include buoyancy effects (top minimum and maximum stresses always include buoyancy).

Using the same analysis but adding the 5-degree DLS bending stress with T/4 stress method:

Top Maximum Stress: 39,488 psi
Top Minimum Stress: 12,438 psi
UTS of Rod: 115,000 psi
Stress Load: 116.1%

Top Maximum Stress: 39,488 psi + 3,740 psi = 43,228 psi
Top Minimum Stress: 12,438 psi + 3,740 psi = 16,178 psi
UTS of Rod: 115,000 psi
Stress Load: 124.8%

An increase of ~9% Stress Load is realized by incorporating bending stress on the continuous-rod section. This is nearly a 7.5% computational error based on the lack of inclusion as it relates to deviated well-bores and how the bending stresses affect the system.

CONCLUSION

Black Mamba and dv8 Energy have worked together to include a stress-loading fact checking tool for evaluating bending stresses in the system, combined with standard Gibbs and Goodman computation of stress loading (See Presentation).

Many operators refuse to use the *Service Factor* function of rod string design as it manipulates stress loading to an intentionally misrepresented number. There are many avenues of incorporating safety factors into the system design through string design software. Dampening coefficients, friction coefficients, stress methods, service factors, etc.

All of these are ways to address ‘noise’ in the system which adds and subtracts stress to the rods and the system. In an ideal world, the predictive software in its base form evaluates and computes accurate stress loading for the environment and application so a real, honest picture can be understood. From there, the operator can apply safety-nets to their preference, rather than a universal “*We don’t know what is actually happening down-there”* catch all.

Engineers need to be aware that complicated well-bores are undoubtedly working sucker rods harder than clean-tangent drills or well-bores which are considered vertical. An inclusion in bending system stresses will only help us in industry understand rod string dynamics in a more complete fashion.

Additionally, the industry continuously designs around **minimum** Ultimate Tensile Strength, the UTS rating from sucker rod manufacturers. Regarding UTS tolerance, rod manufacturers hover above the minimum UTS rating to make sure they are not shipping rods too weak in strength or soft in hardness. This margin and mechanical safety factor in manufacturing assists in proper string design safety net, too. Taking our 7/8” sucker rod analysis, an improvement of UTS from the minimum 115ksi to 120ksi results in a Stress Load computation reduction of ~4%, making the bending system stress analysis counteracted through an improvement of UTS.

Other ways to limit the influence of bending-system stress includes a sucker rod rotator, so long as it is properly working. The rods will spin and rotate, distributing the cyclic fatigue of increased stress throughout the exterior surface of the sucker rod. This can minimize the effects of the additional stress on the sucker rod.

This papers intent is to open the eyes to production engineers and rod string design professionals. Many little improvements, however small, both on the predictive side and in the installation and application of various products for the system, can add up to become a significant change in the reliability of the rod lift system.

REFERENCES

[1] Drilling Manual. *Drilling Pipe Fatigue.* DrillingManual.com, 2022.