SELECTED BEST RECOMMENDED PRACTICES

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ABSTRACT
Utilizing Best Recommended Practices adds value, this paper will discuss selected Best Practices.

1. Tensile strength is an essential criterion in sucker rod selection. The primary cause of sucker rod failures is **not** due to exceeding the tensile strength threshold but is due to compression from the polished rod velocity exceeding the plunger velocity. Value can be added from loading up rod strings w/sinker bars.

2. Operating sucker rod lifted wells with incomplete pump fillage or in wellbores with severe mechanical friction will result in high failure rates, even when applying best practices to the operation of their wells.

3. A specific design and size of a gas separator determines the maximum liquid rate that will **not** entrain gas into the pump intake. This rate is defined as the Separator Liquid Capacity. If the pump displacement exceeds the separator liquid capacity, then gas separation will fail and the producing efficiency of the well will be poor. Proper gas separator selection requires that pump displacement **not** exceed gas separator liquid capacity. The gas anchor length is a significant factor in the number of incomplete fillage cycles. Too long of a gas anchor results in gas breaking out. This is particularly damaging when operating with pump off controllers. The gas collects below the standing value and is processed at start up.

4. Properly setting sucker rod pump clearances maximizes value. After the Patterson Slippage Equation was developed, industry began opening pump clearances. It was believed increasing pump clearances would improve solids handling and decrease failures. When pump clearances are too large and the pump is filled with liquid, the results are 1) increased energy cost, 2) increased failures from increased polished rod velocities to lift the same volume, and 3) increased capital investment to surface the same volume.

5. It is a common practice for operators to delegate responsibility for down hole sucker rod pumps to pump companies. Inspecting the failing component is usually required to identify and understand the cause(s) of failure. Part of failure analysis is to be present when failed equipment is pulled from the well and to attend to the tear down of pumps when repaired at the pump shop. Delegation

without proper oversight can increase costs. Oversight procedures should be included in Best Recommended Practices. When pump repair volumes are adequate, it may add value for operators to own pump shops.

BEST RECOMMENDED PRACTICES:

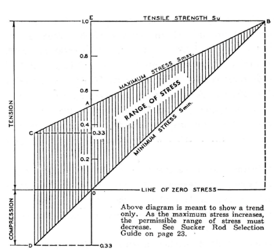
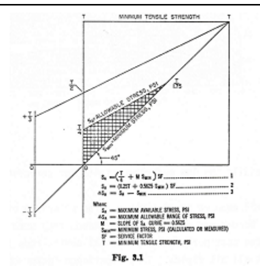
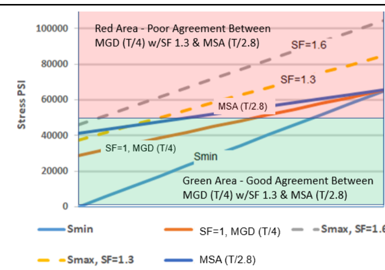
1) BEST RECOMMENDED PRACTICE:

DESIGN METAL RODS STRINGS USING THE HIGH STRENGTH EQUATION (MSA T/2.8) FOR Grade “C” or “D” RODS W/SINKER BARS AND EVALUATE THE COST EFFECTIVENESS OF CHANGING TO GRADE K OR KD RODS, HIGH STRENGTH RODS, OR FIBERGLASS SUCKER RODS.

Discussion:

Sucker Rod Loading

Either rod grades “C” or “D” should be selected. The selection should be based on their predicted rod loading as applied to **the Modified Stress Analysis (MSA) (T/2.8) with a Service Factor (SF) of 1.0**. The **cost effectiveness of all exceptions should be evaluated** before installation and confirmed with look backs.

Stress Fatigue Diagram for Polished Steel Bars Conducted in Laboratory Conditions	**API Modified Diagram (≈1954) For Sucker Rods in Field Applications**	API Modified Good Diagram w/ Various Service Factors and Modified for High Strength Equation
		

The Goodman Diagram was developed by John Goodman and published in 1908. His analysis was based on polished steel bars in laboratory conditions. The purpose was to predict railroad car axle failures from stress fatigue (above left). In 1954, a group of oil industry experts modified the diagram by consensus for application to metal oilfield sucker rods (above center). They based this modification on their subjective judgement from experience. It is worth noting that they 1) eliminated the compression portion and 2) reduced the acceptable Maximum Allowable Stress (Smax) by about half.

The API Modified Goodman Diagram (MGD(T/4)) has been used in predicting the allowable stress range from 1954 until recently. The acceptable Stress Range is from Stress Maximum (Smax) to Stress Minimum (Smin) for any given Smin. The Service

Factor (SF) was used to adjust the acceptable stress range up or down based on the specific well conditions. SFs less than 1.0 were used for years to adjust for corrosion and abrasion but history suggests this is too conservative.

Three SF cases are shown above right (SF= 1.0, 1.3 & 1.6). 1) A SF of 1.0 was the standard when the MGD (T/4) was originally applied. 2) A SF of 1.3 is currently accepted. 3) A SF of 1.6 is being evaluated in those applications where it is felt that a SF of 1.3 is too conservative. 4) the MGD utilizing the High Strength Equation or **Modified Stress Analysis (MSA) (T/2.8)** is an alternative to the MGD(T/4) w/ a SF of 1.3. For now, frequently used design programs do **not** have the **(MSA) (T/2.8)** proposed range of stress prediction.

The graph (above right) shows how higher SFs and MSA (T/2.8) affect Smax. The Stress Range for a SF of 1.30 and MSA (T/2.8) have good agreement for Smax less than 50,000 psi (area shown in green). The agreement for Smax greater than 50,000 psi is **not** good (area shown in red). This illustrates how the use of a higher SF can approximate the New MSA (T/2.8) but does **not** predict the new performance exactly and gives good results as many have taken this approach with success.

Sucker Rod Expert Russel Stevens has been using the MSA (T/2.8) w/ a 0.9 S.F. which approximates an API Modified Goodman Diagram (MGD) (T/4) w/ a S.F. 1.3 for 30 years [1]. Evolving from MGD (T/4) to MSA (T/2.8) has resulted in **no** significant increase in failures.

The MGD (T/4) w/ a SF of 1.6 has been evaluated. 1) Testing was done in the lab in the year 2000 on D grade rods' They were loaded to Smax based on **API MGD (T/4) w/ a S.F. of 1.6** and cycled 10 million cycles in air. No fatigue failures occurred. 2) In addition, operators are loading "D" or "KD" rods to API MGF W/ a S.F. of 1.6 and others are trying MSA (T/2.8) using 1.2 or 1.3 S.F. The positive results indicate that the MSA (2.8) w/ a 0.9 S.F. may be too conservative.

History suggests that sucker rod designs are significantly underloaded. It would be in the operator's best interest to perform destructive testing (loading rods to point of failure) in their specific application and determine Smax based on field conditions. Capital investment and operating expenses can be reduced if Smax determined from destructive testing is higher than Smax used in current design procedures.

Developing a culture that looks back at the cost effectiveness of design recommendations is necessary to avoid operating cost creep. Best Recommended Practices should be reviewed in the specific field applications before accepting.

It is recommended that all rods be designed using MSA (T/2.8) w/ a 1.0 S.F. Since the load range has been greatly expanded in recent years, the application of High Strength Rods has become limited."

Rod Grade Selection

The **old thinking** is that Grade “C” rods be run until their loading exceeds acceptable stress range based on the stress fatigue diagram. When the Grade “C” loading is exceeded, Grade “D” rods should be run. If corrosion is high, then a “K” rod should replace the “C” rod and the “KD” rod should replace the “D” rod. Some companies have used the “KD” rod in what would normally be the “C” application.

Oxy ran a study [2] in 2016-17 that **compared the failure rates and costs of “C” vs. “KD”** in about 300 wells producing from the San Andres formation (≈5,000’) which were effectively chemically treated for corrosion. When the chemical treatment program is effective, then the results of corrosion testing showed there was no discernable difference in the corrosion rates between Grade C and Grade KD rods in those wells.”

During this testing, Oxy material Subject Matter Experts (SME’s) advised that the alloy content (nickel/chromium/ molybdenum) would have to be greater than 5% to have a meaningful impact on corrosion resistance. The American Petroleum Institute (API) Specification 11B only requires 1.15% alloy content for Special Alloy rods. The SME concluded that there was no significant difference in corrosion tolerance between the “C” rod and the “K” rod or the “D” rods and the “KD” rods. **The new thinking is that the cost effectiveness of switching to “C” to “K” or “D” to “KD” when corrosion is excessive should be evaluated.**

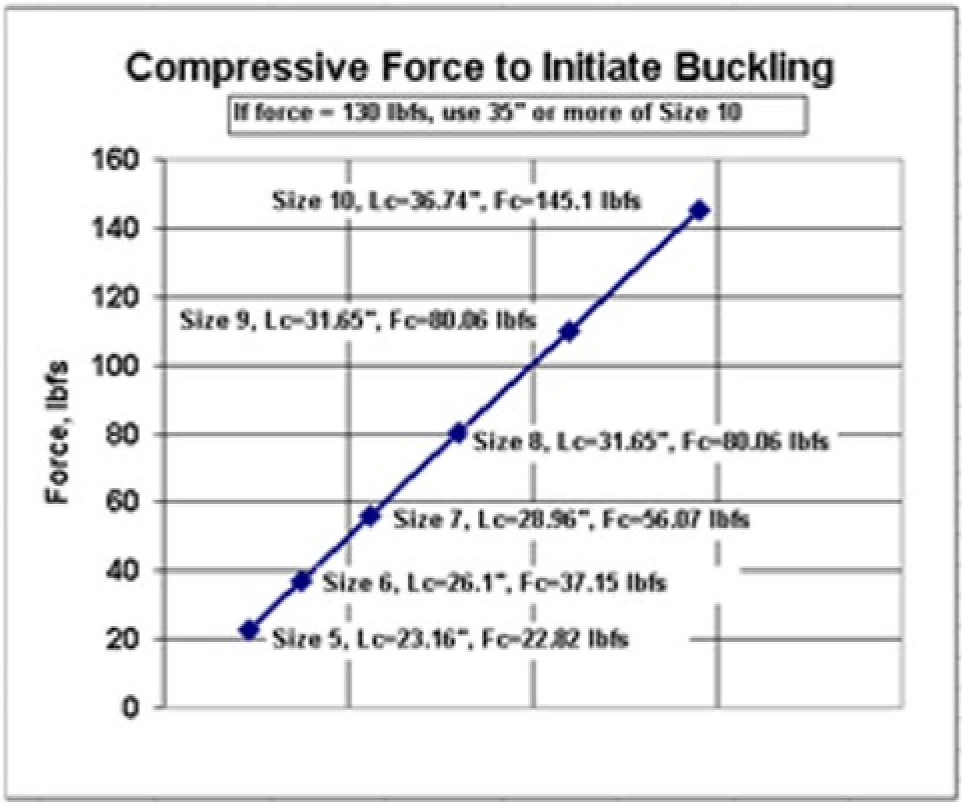
Fiberglass Rods

An accepted procedure for wells responding to water injection and exceeding their production capacity has been to install **fiberglass sucker rods** and keep the existing pumping unit. Oxy studied about 126 wells with fiberglass sucker rods producing from the San Andres (≈5,000’), Gloriette (≈6,000’) and Clearfork (≈7,000’). A lookback evaluated the cost effectiveness of these installations and a paper was presented in 2022 (SPE-209731) [3]. Fiberglass rods are represented to have a life expectancy of 30-40 million rod reversals when properly loaded. Even though properly loaded, the study revealed that approximately 80% of these wells had failures at fewer than 10 million rod reversals and 90% of the fiberglass tapers were replaced with five million rod reversals. Field personnel described **80-90% of recent failures as fiberglass connection failures**. The look back determined that **it was not cost effective to switch to fiberglass sucker rods compared to upsizing the pumping unit and using metal rods**.

Sinker Bars

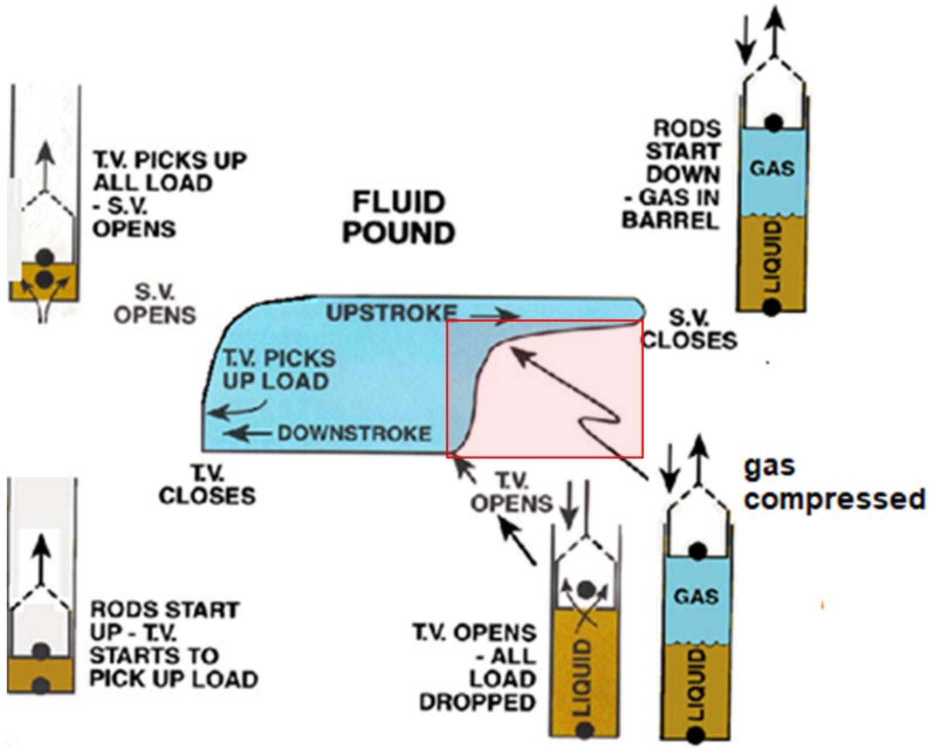
The Stress Fatigue Diagram for Polished Steel Bars shown above includes a section in which the minimum stress is less than zero (going into compression) and the maximum stress is positive. This suggests that steel bars can have negative loads without experiencing stress fatigue. The MGD (T/4) shown in the middle eliminated that portion of the curve. Experts felt that sucker rods would have a better life if there were no compressive forces on the sucker rods. Many rod pumping cycles experience compressive forces immediately above the pump. It is believed that compressive forces will not result in significant loss of rod life unless the compressive forces exceed the compressive forces in Initiate Buckling shown in a chart below.

Below are the (compressive) forces required to buckle rods:



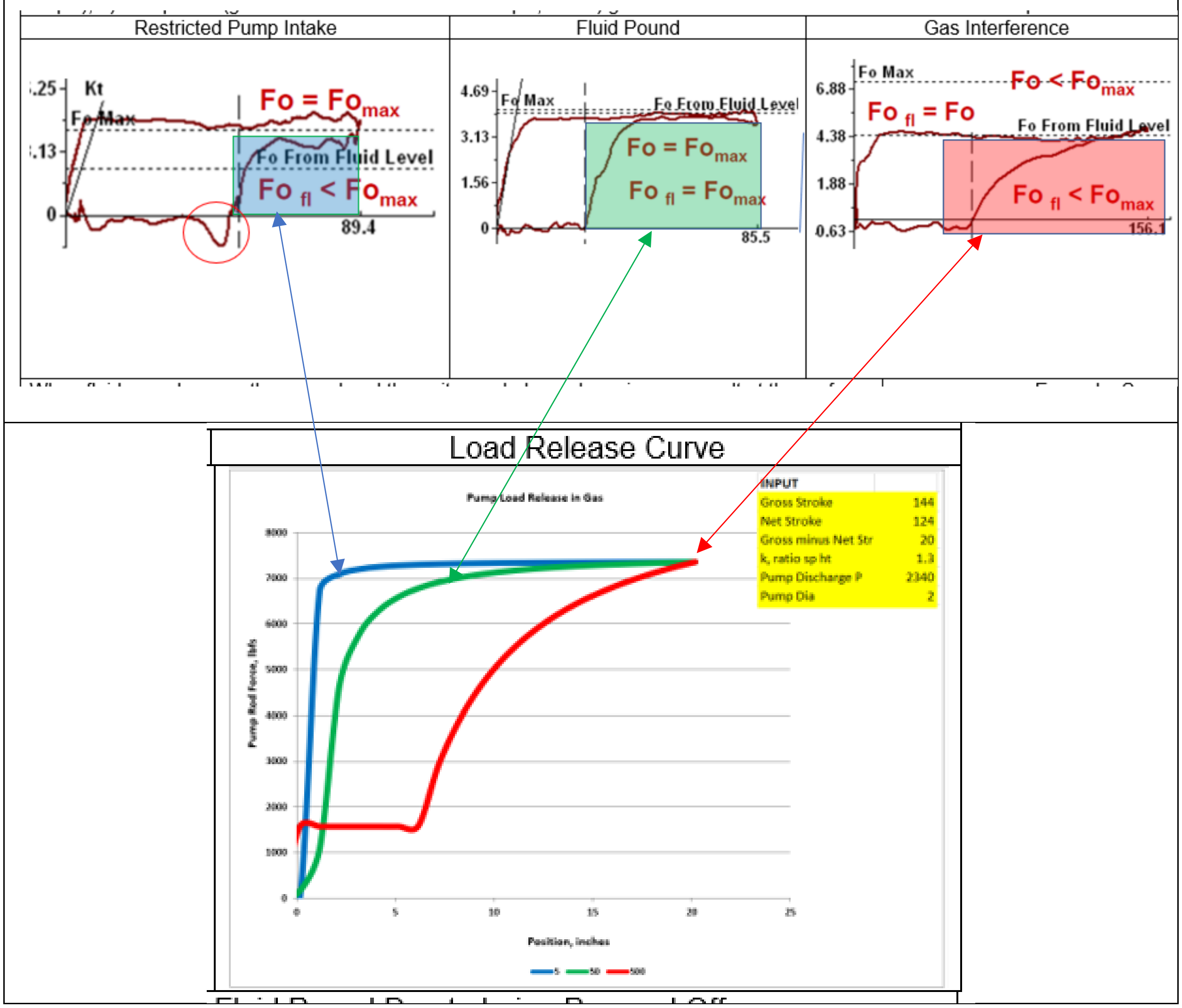
It is an industry accepted practice to run **sinker bars** or the largest rod that will fit in the tubing above the pump. It is important to be able to Identify **Negative Effective Minimum Loads** when evaluating the running sinker bars. Mechanical friction applied to the rod string at any location from the surface to the pump, results in the wave equation calculated pump card displaying negative loads on the down stroke making difficult/impossible identifying **Negative Effective Minimum Loads**. Sinker bars can be run to manage the decreased loading that occurs on the rods above the pump during the downstroke of a rod pumping cycle or when compressing gas. If the compressive loads exceed the load that buckles the rods, then rod-on-tubing wear and bending occurs and the rods/tubing will rapidly fail.

The below illustration shows the **major events for rod loading for the downstroke when gas is in the pump barrel**.



First, the load is picked up (on left) and then the plunger strokes upward. Second, the downstroke starts and the gas under the plunger is compressed on the **load release** until the force under the TV is such that the TV opens. (Load Release Curve Area is shown in pink area.) Third, the plunger continues to complete downstroke. For the simple illustration above, there is no negative loading on the rods at the pump on the downstroke because the diagnostic wave equations calculated 0 load at the end/bottom of the rod string by removing the weight of rods in fluid, fluid damping on rod string and dynamic forces. Although some think that negative loading can occur 1) on the load release due to plunger slowing to open the traveling valve, 2) due to pump friction from tight clearances, and 3) differential pressure change across the plunger.

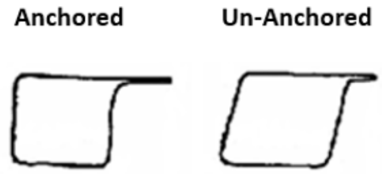
Example: Of How Various Gas Compression Cards Relate to the Load Release Curve
The Shape of the Load Release Curve is related to the pressure inside the pump barrel at the start of the downstroke. In most cases (except restricted pump intake), this is the Pump Intake Pressure and correlates to fluid level. The example below shows how the load release curve relates to 1) restricted pump intake (blue load release line Pressure in Pump at start of downstroke ≈5 psi), 2) fluid pound (green load release line PIP ≈ 50 psi), and 3) gas interference (red load release line PIP ≈ 500 psi).



Many believe that Fluid Pound occurs when the plunger strikes fluid, and a negative load occurs. However, the example of the Restricted Pump Intake card above illustrates that negative loads can be seen on the downhole card when the pressure inside the pump at the start of the down stroke is very low. The Fluid Pound Card and the Gas Interference card both have adequate pump intake pressure to allow the gas to compress and avoid negative loads.

When fluid pound occurs the ground and the unit can shake and a noise can result at the surface. Fluid pound is damaging but what is happening?

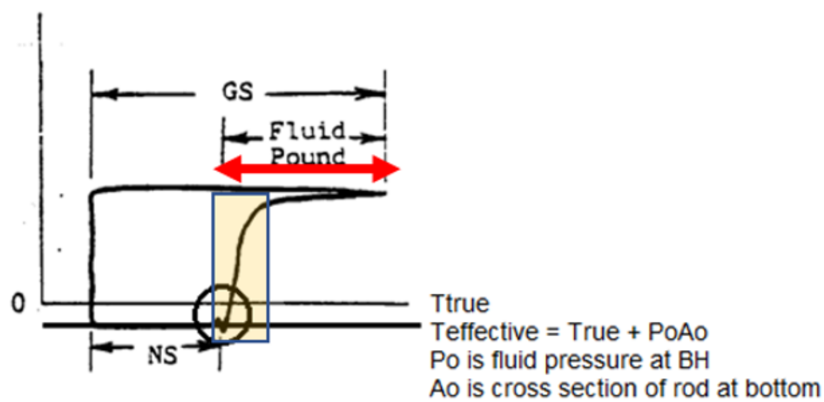
Example: Severe Fluid Pound Due to being Pumped Off



The authors of the two cartoons (right) of the bottom hole dyno card say fluid pound is occuring on the well with the quick load release. The load release can be quick on a well that is pumped off. The load release (with gas) at high intake pressures is more gradual and not as quick. The illustrations imply “fluid pound” but do not have compression or negative loading. So, there is no tendency for buckling (the dynomometer cards are in Effective Loading or plot on the zero line on the down stroke). One concludes fluid pound is not from negative loading when the plunger hits liquid in the barrel.

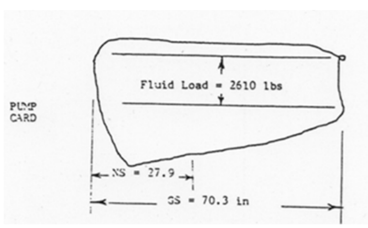
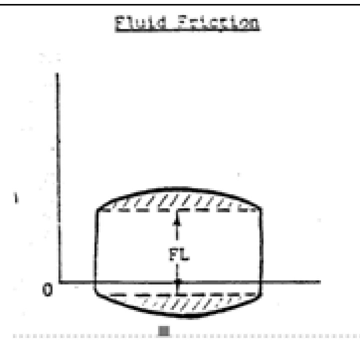
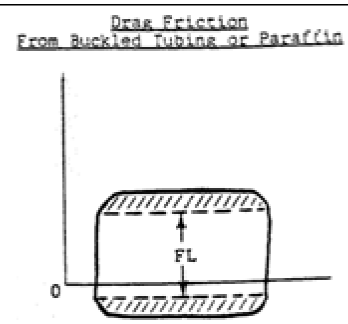
Fluid pound is due to the rapid load release of the tension In the rods which sends a shock throughout the system.

Example: Severe Fluid Pound w/ Negative Load. (Note: the illustration shows the fluid pound over the entire Load Release. However, the fluid pound is only over a portion of the Load Release Curve.



It is easy to see that fluid pound occurs during the load release but “fluid pound” seems short lived and may occur over a shorter time during the load release (see actual load release in orange rectangle but artist description of load releaes in red douple arrow above.). This curve plots below the zero line and implies negative loading. This is because the plot is in **“true force”** as opposed to **“effective force”.** The true force curve may or may not have negative loading. The above example is a true force curve with a very slight possible **negative loading (in the circle)** resulting in a tendency to buckle? In order to buckle the rods, the negative loading must be greater than the force necessary to buckle the rods.

<table>
<tr><td colspan="3">Three examples with negative loading on the dyno cards at the pump but there is no fluid pound.

Below are several case of negative loading with out Fluid Pound..</td></tr>
<tr><td>Negative loading on the downstroke caused by sand and viscous oil
</td><td></td><td></td></tr>
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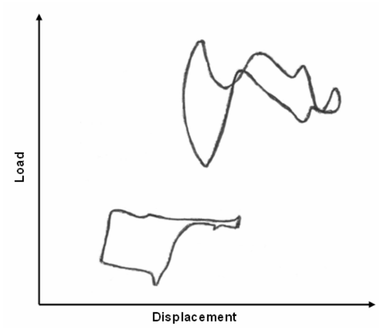
Below is an example of a bottom hole fluid pound card with negative loading near the the area of the TV opens. Negative loading translates to buckling if the load exceed the negative load to initiate buckling. The onset of negative loading is most likely the plunger hitting fluid. Echometer says they have **not** seen this negative load in a pump off card when the TV opens unless the pump intake is blocked and/or NO FREE gas is inside the pump chamber to compress. In a well where the pump intake is completely blocked, then pump fillage is due to slippage from the tubing above the plunger; resulting in there being NO FREE gas inside the pump chamber to compress on the down stroke. In a pumped off well with an absolute vaccuum on the casing, then NO FREE gas exist in the casing and gas cannot enters the pump intake or partially fill the pump chamber. When any free gas is present inside the partially liquid filled pump chamber on the down stroke traveling valve opens by compressing the free gas to the discharge pressure, while compressing the gas the plunger velocity incrementally decreases as the free gas pressure increases above the intake pressure. Compressing the pump chamber gas results in the plunger slowing down, when the pump intake pressure is low then the plunger velocity decrease is more sudden and occur closer to the point where the TV opens. When NO FREE gas is inside a partially liquid filled pump chamber, then on the down stroke to open the traveling valve the plunger velocity is approximately equal to the polished rod velocity and the plunger velocity does not decrease since NO FREE gas exist inside the pump chamber; resulting in the plunger hitting (pounding) the liquid inside the pump chamber at a very high velocity. Once the plunger pounds the liquid then typically the plunger stops. If NO FREE gas exist inside a partially liquid filled pump, only then does fluid pound exist when the plunger slams into the liquid in the pump chamber at a very high velocity.



Example of severe fluid pound w/o negative loading due to well being pumped (w/ anchored tubing).¶
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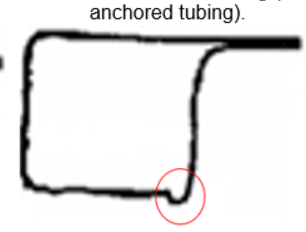
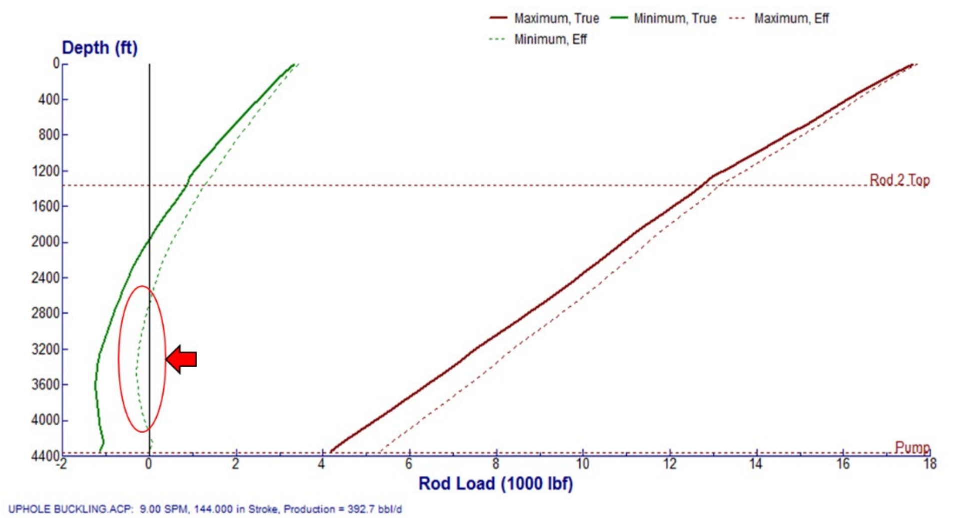
Example of severe fluid pound with some negative rod loading which can lead to buckling (w/ anchored tubing).



This is an example of true and effective loading w/o negative effective loads

Above is a plot of true and effective forces the the upstroke and the downstroke. The plot has no negative effective loads throughout the rod string.

This case shows negative effective loads above the pump on the down stroke but not at the pump.



The above case shows negative effective loads on the down stroke. The section in the red circle above the pump shows negative effective rod loads due to rod dynamics. If the loads are high enough, there would be buckling over this rod section above the pump but not at the pump. If you see this, you would want to adjust pump design (i.e. change pump size of SPM) until there is no predicted bucking in the rods over the pump.

Some companies have a policy of running sinker bars in every well. Other companies run sinker bars when they are experiencing negative effective minimum loads. Current design recommendations are that the bottom rod above the sinker bar be in 300 psi to 650 psi tension.

The most common sinker bars diameters are:

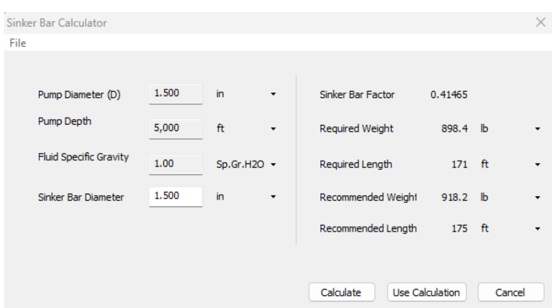
- 1 3/8 in. (1.375 in.)
- 1 1/2 in. (1.500 in.)
- 1 5/8 in. (1.625 in.)
- 1 3/4 in. (1.750 in.)

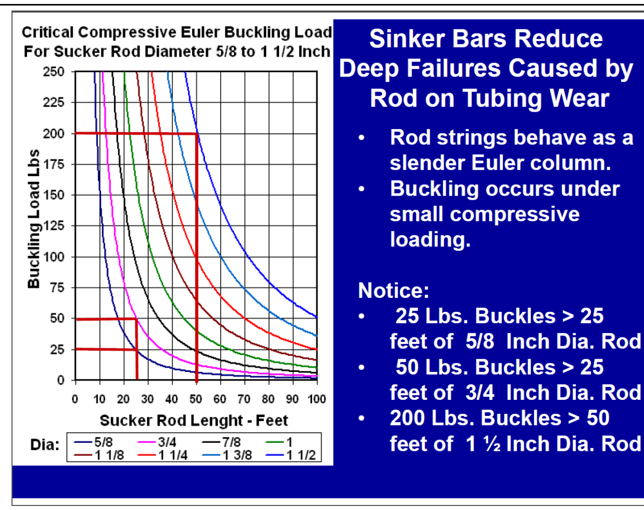
All four-sinker bar diameters can safely operate in 2 7/8-in. outer diameter (OD) tubing. Only the 1-3/8-in. (1.375-in.) and 1-1/2-in. (1.500-in.) diameter sinker bars can safely operate in 2 3/8-in. tubing.

Install sinker bars that are one diameter smaller than normal if the fluid volumes are greater than 700 BFPD to reduce the piston effect and reduce fluid velocity. Delayed closing of the traveling valve can result, when the oil is less than 25 API gravity then oil viscosity between weight bars and tubing can also create the piston effect and it is recommended to install sinker bars that are one diameter smaller than normal. This will aid with corrosion inhibitor filming and corrosion inhibition.

Some commercially available rod design programs have a recommended quantity of Sinker Bars calculated utilizing the Wave Equation. Experts are concerned that this recommendation is dependent on the 1) assumed pump fiction and 2) the accuracy of the wave equation immediately above the pump. QROD uses an alternative technique which is independent of the wave equation (below right). A sinker bar calculator (below left) is available in QROD and may be used to obtain a second opinion to other design programs.

Sinker Bar Calculator From QRod	Calculation Method Used in QROD





<u>2) BEST RECOMMENDED PRACTICE:</u>
<u>MINIMIZE THE QUANTITY AND INTENSITY OF INCOMPLETE FILLAGE CYCLES.</u>

Discussion:

Oxy conducted a study of Polished Rod Velocity (PRV) (SPM × stroke length) vs. **Failure Frequency (FF)** on 6,000 Permian Basin wells mostly producing from the San Andres (≈5,000') and Clearfork (≈7,000') in 2018 [4]. At the same time, Lynn Rowlan was suggesting that FF was driven by the difference between the PRV and the Plunger Velocity. The primary reasons for the Plunger slowing down in relation to the Polished Rod are 1) incomplete pump fillage and 2) severe mechanical friction in wellbore.

It is assumed that wells producing less than 24 hours on Pump Off Controllers (POC) cycle off due to incomplete fillage and wells producing 24 hours on POC do **not** experience incomplete fillage. Oxy employee Chris Cavazos deduced that Lynn's hypothesis could be confirmed by comparing the PRV vs. FF for 1) "24-hour wells" (wells running 23.5± hours per day) to 2) wells that produce less than 24 hours. This comparison was made (FIGURE #1 below) and has a strong correlation. Figure #1 shows that 40% of failures occur from normal circumstances and 60% of failures are related to incomplete fillage. **Properly managing the intensity and quantity of incomplete fillage can greatly reduce operating expenses**. **[5].** A FF of ≤1.0 is reasonable if incomplete fillage can be avoided.

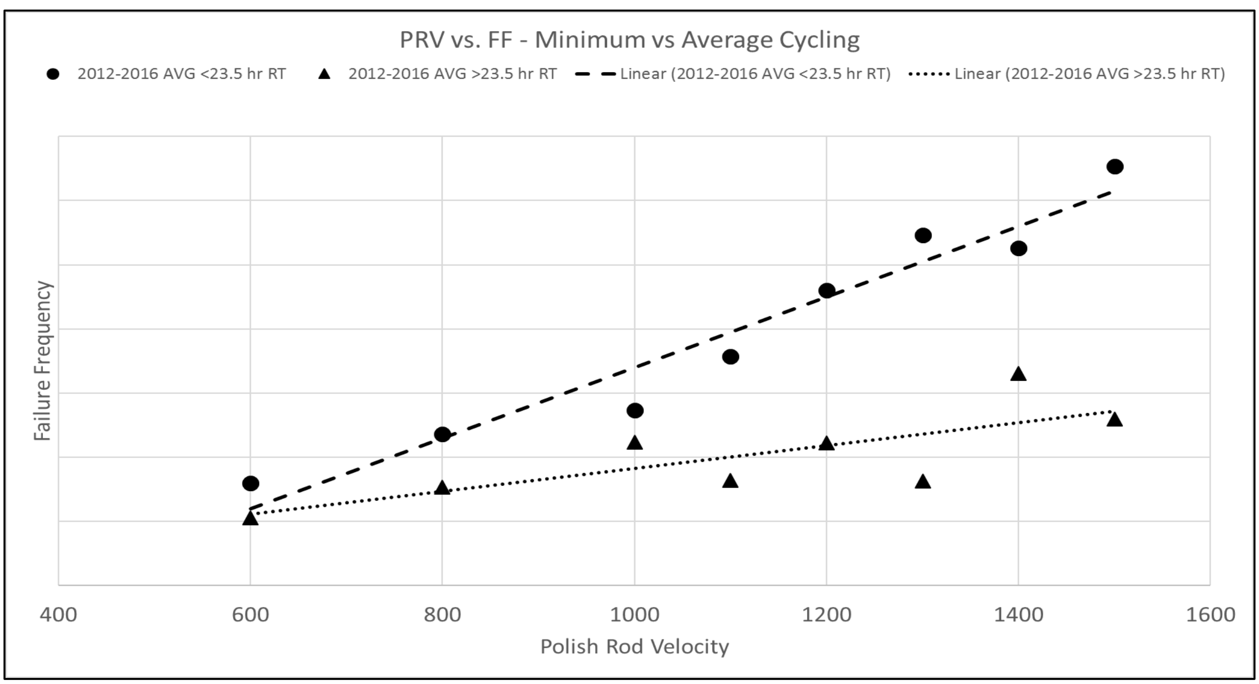


Figure 1: Polished Rod Velocity vs. Failure Frequency

The data from the PRV vs. FF was reformatted to develop a Mean Time Between Failures (MTBF) chart. Figure 2 shows the relationship between polished rod velocity and MTBF. Figure 3 shows the relationship between stroke length, pumping speed, and MTBF. There has been a disagreement over the recommended practice of operating sucker rod lifted wells “short and fast” or “long and slow.” Figure 3 shows that **for any given PFV and incomplete fillage there is a given FF and that FF can be obtained by pumping long and slow or short and fast** [6].

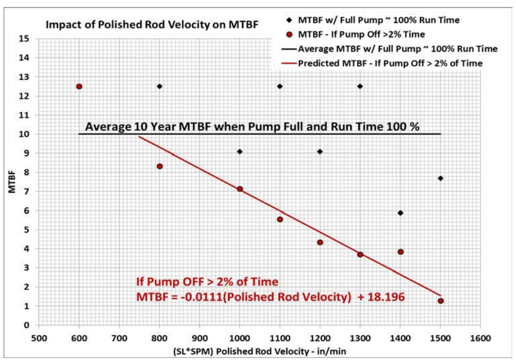


Figure 2: Relationship Between Polished Rod Velocity and Mean Time Between Failures

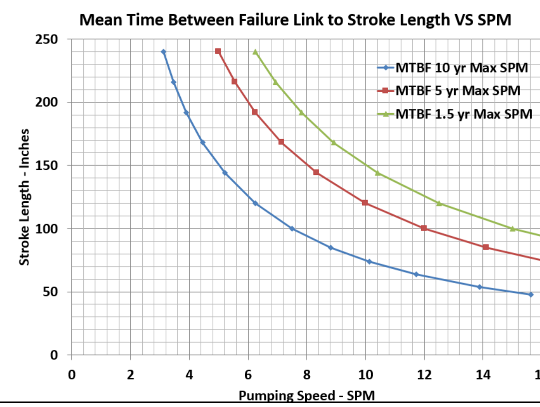


Figure 3: Relationship Between Stroke Length, Pumping Speed, and MTBF

3) BEST RECOMMENDED PRACTICE:

CONDUCT FIELD STUDIES USING THE EQUATION (DIP TUBE LENGTH in. = 200/SPM.) TO IDENTIFY THE GAS ANCHOR LENGTH THAT MAXIMIZES VALUE.

Discussion:

It is easy to falsely assume that lengthening the Gas Anchor or Dip Tube will improve separation efficiency in a downhole gas separator. Many operators make a poor boy gas separator from a 30’+ joint of pipe. It is a common practice to fully utilize the 30’ joint of pipe by installing long gas anchors (10’-20’). However, a gas anchor longer than necessary 1) increases friction loss, 2) lowers separator pressure, 3) increases the volume of gas coming out of solution, and 4) decreases separation efficiency.

An excellent paper presented at the SWPSC in 2017 helps us understand [7].

Gas Separator Length

"Another factor to consider is the length of the gas separator. Many existing designs utilize from one to three pump volumes as the proper volume to contain in the gas separator between the inlet ports and the lower opening in the dip tube. Visual studies performed at the University of Texas lab having clear casing and a clear gas separator indicate that the rate at which gas bubbles of air migrate upwards in the water liquid column and the pumping speed are two of the controlling factors. If a well is pumping 10 strokes per minute, the pumping cycle time is 6 seconds. Three seconds occur on the upstroke and three seconds occur on the downstroke. If gas bubbles are drawn down into the gas separator annulus during the upstroke, these gas bubbles should be liberated from inside the gas separator, when the pump is on the downstroke. **On the downstroke, these bubbles will migrate upward at a rate of approximately 6 inches per second. If the downstroke time is 3 seconds, gas bubbles will migrate upward 18 inches. This suggests that a dip tube should extend at least 18 inches below the gas separator inlet perforations for a well pumping 10 strokes per minute.** Longer length dip tubes results in additional friction loss and the release of free gas evolving from the oil flowing up the dip tube into the pump. A long stroke unit operating at 4 strokes per minute has a downstroke time of 7-1/2 seconds which will allow gas bubbles to flow upward approximately 45 inches suggesting a dip tube length of 45 inches. The collar-size gas separator has a dip tube that extends about 55 inches below the bottom of the outer barrel inlet ports which is sufficient to satisfy the great majority of pumping well conditions."

4) BEST RECOMMENDED PRACTICE:

USE THE PATTERSON SLIP EQUATION TO IDENTIFY THE DOWNHOLE ROD PUMP CLEARANCE. ALWAYS KEEP CLEARANCES AT OR ABOVE 0.003". FIELD EXPERIENCE SUGGEST SINKER BARS SHOULD BE RUN WHEN CLEARANCES ARE 0.003".

Discussion:

The Patterson Slippage Equation (PSE) was published in 2007 [8] and estimates the down-hole-rod-drawn pump fluid slippage. It was developed in a cooperative effort between industry and Texas Tech. The Texas Tech University test well, Red Raider #1. Was used in developing the equation.

In 2012 a table was published [9] recommending clearances for 1) a pump located above 8,000' with 2) a 48" plunger with a "+1 Barrel". The table made broad assumptions trying to simplify designing pumps as compared to running PSE for each pump design. Industry broadly applied this table which resulted in poor pump performance (excess slippage). **The excess slippage 1) increased energy cost, 2) increased failures from increased polished rod velocities to lift the same volume, and 3) increased capital investment to surface the same volume.**

Lynn Rowlan made a presentation later highlighting the problems with using the Table and encouraging people to **use the equation instead of the table**. He also expressed concerns about using clearances less than 0.003”. Field experiences suggest that running sinker bars on pumps w/ 0.003” clearance reduces failures. It has been difficult to get industry **not** to use the table. A SWPSC paper [10] was presented in 2019 recommending a pump clearance based on certain criteria and calculated the incremental cost of opening up clearances more than the recommended. The calculator estimates the increased cost from electricity and failures for the increased displaced volumes.

One commonly held belief is that the clearances on pumps should be opened when solids are present. Pump clearances maximum of 0.009” is a common practice in USA. During the life of the well pump clearance should change in response to changes at the well. The slippage calculator should be used to determine the resulting decrease in pump displacement due to increased slippage related to increasing the pump clearances. Opening pump clearances has been shown to reduce damage by solids causing pump sticking and by solids causing abrasion/excessive wear to the plunger and barrel. In high-rate wells reduction of pump displacement by in excess of 30% due to slippage is common. As production from a well naturally declines the reduction in pumping speed by variable speed controller or at slower SPM with 0.009” pump clearance slippage can become excessive with reduction of pump displacement in excess of 50% due to slippage. (Slippage is reduced in wells with API gravities of 30-32 Deg API or lower and the viscosity of the oil water mixture should be used in the calculation of pump slippage. When API gravities are lower and wells produce solids, then pump clearances can be increased above 0.009” without significant loss in production. When oil percent drops below 20-30% then water viscosity should be used and tighter pump clearances in the 0.009” or less is required because of excessive slippage.)

5) BEST RECOMMENDED PRACTICE:

MAXIMIZE THE VALUE OF DOWNHOLE ROD PUMP PROGRAMS.

Discussion:

The goal of the operating company is to maximize value to its shareholders. The goal of a pump company is to maximize value to its shareholders. The interests of the operating company and the pump company may **not** be aligned because the profits of the pump company may reduce the profits of the operating company. Managing the relationship between the pump company and the operating company is required to better align interest. Defining and enforcing rules and procedures is one way to better align interests.

When pump volumes are adequate, it may be prudent for operating companies to own their pump shops.

Effective managing a downhole pump program require 1) designating an operating company representative to have single point responsibly/accountability for optimizing downhole pump programs, 2) defining equipment replacement guidelines and standard repair procedures, 3) conducting periodic audits and training, and 4) conducting random audits.

CONCLUSION

The goals of service/supply companies and the operating companies **do not** naturally align. It is in the best interests of operating companies to analyze their data and determine the Best Practices in their specific applications.

Several Recommended Best Practices have been discussed in the paper. Best Practices should be evaluated in specific field applications before being included in design procedures. Value may be added by implementing the procedures discussed.

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