EXAMPLES OF FORCES NOT ACCOUNTED FOR BY THE WAVE EQUATION

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ABSTRACT
Different types of forces NOT accounted for by the wave equation are 1) mechanical friction, 2) piston force acting on the polished rod due to tubing back pressure and 3) true vertical rod weight. Mechanical friction will be discussed from 1) over-tight stuffing box, 2) down hole sticking due to a severe dogleg in the wellbore profile and 3) friction from paraffin along a section of the rod string. The application of these external mechanical forces acting on the rod string impacts measured surface loads, down hole stroke length and plunger velocity, plus the calculated rod loading at the pump or other locations in the rod string.

Damping coefficients are used to subtract out fluid damping as a function of velocity along the rod string using the wave equation. Unaccounted for mechanical friction cannot be modeled by adjusting the damping factors in the wave equation. Mechanical friction impacts both the shape of the pump card and the measured surface dynamometer card loads versus position and, as friction on the rods goes up then the surface load range also changes. Field measured dynamometer data will be used to show examples of these different types of forces NOT accounted for by the wave equation

Introduction

Field measured dynamometer data will be used to show examples of different types of forces NOT accounted for by the wave equation. These forces are 1) mechanical friction, 2) piston force on polished rod due to tubing back pressure and 3) true vertical rod weight. Mechanical friction will be discussed from 1) over-tight stuffing box, 2) down hole sticking due to severe dogleg in the wellbore profile and 3) friction from paraffin along a section of the rod string. The application of these external forces acting on the rod string impacts measured surface loads, down hole stroke length and plunger velocity, and calculated rod loading at the pump or other locations in the rod string.

Mechanical Friction in the well changes the measured rod loading at the surface stroke. The mechanical friction acting on the rods are not modeled by the wave equation[1,2], resulting in the excess frictional loads and horsepower being displayed in the plot of the pump card. The pump card in effect becomes the trash can for any mechanical loads applied to the rod string, including the weight of the rods that rest on the side of the tubing in a deviated wellbore. When the pump card loads are higher than expected on the upstroke or plot below the zero load line and these pump loads are outside the expected load range often indicating the presence of mechanical friction unaccounted for by the wave equation.

Normally the pump slows or stops at the beginning and end of the stroke. When the plunger stops unexpectedly during the up stroke or down stroke, sticking down hole is frequently the cause. The rod stretch spring constant, Kr, can be used to identify the depth to a severe dogleg in the well that is causing the downhole sticking.

Use of a back-pressure regulating valve on the tubing can prevent the liquids in the tubing from unloading, when the tubing liquids are lightened from gas being pumped into the tubing. The surface tubing discharge pressure is higher when compared to the surface pressure if no back-pressure regulating valve were present. A piston force equal to the back pressure times the area of the polished rod reduces the polished rod load that are measured.

The damping coefficients are used to subtract out fluid damping/friction from the loads calculated using the wave equation down the rod string as a function of velocity. When the damping factors are properly adjusted fluid friction on the rods is properly modeled and the pump card load lines during the up and down stroke are generally flat. Unaccounted for friction impacts the shape of the pump card, as friction on the rods goes up then the surface load range also increases. In wells that are deviated, the up stroke loads go up and the minimum loads decrease due to additional friction on the rods. Damping factors are not used to remove mechanical friction on the rod string. Example dynamometer cards will be presented showing examples of these conditions.

Damping Factors

The damping factors (damping coefficients) are used to subtract out fluid damping/friction along the rod string as a function of rod velocity using the wave equation from the surface loads down to pump loads. When the up and down stroke damping factors are properly adjusted the correct amount of horsepower is removed from the displayed pump card loads. One simplistic method to identify if the correct fluid damping has been removed is the pump card load lines during the up and down stroke are plotted horizontally. The horizontal pump card load during the up stroke when the standing valve has opened with the plunger moving upward through the pump barrel is created by a fairly constant discharge pressure acting on the top of the plunger area and a fairly constant intake pressure acting on the bottom of the plunger area. The horizontal pump card load during the down stroke when the traveling valve has opened with the plunger moving downward through the pump barrel is created by very little or no differential pressure acting across the plunger as the well fluids inside the pump chamber are displaced through the traveling valve into the tubing. The pump card area and height depend on the fluid damping acting on the sucker rods being correctly modeled by the proper adjustment of the damping factors.

Fig. 1 displays the same pump card where the loads are plotted using too low and too high damping factors and with the pump card plotted "correctly" with the up and down stroke damping factors properly adjusted. The up stroke damping factor is properly adjusted when the up stroke pump card load line is being plotted flat/constant (called the fluid load), meaning while the plunger is moving on the up stroke a fairly constant discharge pressure is applied to the plunger top reduced by a fairly constant pump intake pressure acting across the bottom of the plunger area. When the up stroke fluid damping coefficients are properly selected, then the fluid load displayed on the upstroke of the pump card would be plotted as a flat line. On the down stroke with zero load applied to the sucker rods by the plunger, when the traveling valve is open and the standing valve is closed the pump card should be plotted as a flat load line on zero load. There is basically no load from the plunger acting on the rods during the down stroke and the pump card should plot as a flat load line, if the down stroke damping factor is adjusted correctly. When the damping factor is too high on the up or down stroke too much horsepower will be removed from the pump card shape. Too much dampening on the down stroke or too much dampening on the up stroke results in the pump card plotted concave inward toward the center of the card. When the damping factor is too low then the pump card balloons outward and the pump card is plotted as convex outward past where the flat pump card load line would be plotted. If the damping factors are set to zero then the area of the surface card and the pump card would be equal, no loss in horsepower occurs when damping factors are set to zero and equal horsepower (card area) work is being done at the pump and at the polish rod. The purpose of the damping factor is to remove the horsepower used to lift the fluid from the pump intake pressure to the surface. This horsepower lost to fluid damping against the rods creates the fluid load, the pump card height (discharge minus intake pressure times plunger area) adjusted by the up stroke damping factor represents the pump discharge pressure required to move the fluid between the rods and the tubing and lift the fluid to the surface. Fluid damping acting on the rod string is used to account for the loss of horsepower from the surface down to the pump required to produce the fluids to the surface.

Unaccounted for mechanical friction impacts the shape of the pump card, as downhole mechanical friction on the rods goes up then the surface load range also increases. The wave equation uses the damping factors to account for only the fluid contacting the rods and the loss in horsepower from the pump to the surface due to the sucker rods moving through the tubing fluid lifting the fluid to the surface. Damping factors are not used to remove mechanical friction acting on the rod string.

Mechanical Friction

Mechanical Friction in the well can occur at any location along the rod string from the stuffing box down into the pump. The increase in mechanical friction normally results in higher measured surface loads on the upstroke and lower measured surface loads on the down stroke. The location along the rod string where mechanical friction is applied to the rod string is normally not known. The amount of force acting on the rod string from the mechanical friction is also unknown. Since the location and magnitude of the mechanical downhole friction forces are not known, then the pump card shape is impacted by any mechanical friction loads applied to the rod string that are not removed by the wave equation.

When the mechanical friction forces are not removed, then the resulting plot of the pump card displays abnormal loads and excessive horsepower. There are many potential sources of mechanical friction: 1. paraffin, 2. scale, 3. over tight stuffing box, 4. misalignment between pumping unit and well head, 5. dogleg severity, 6. deviated wellbore, 7. pump friction, 8. crimped tubing, plus many other possible sources. Increased forces acting on the rod string due to mechanical friction result in loss of down hole pump stroke. When damping factors are properly adjusted with up and down stroke loads being plotted flat, the pump card loads are higher than expected on the upstroke or plot below the zero load line, then these pump loads outside the expected load range often indicate the presence of mechanical friction from one of these sources. When the pump card load is impacted by mechanical friction and the pump card plots below the zero load line or above the expected loads on the upstroke (Fo from the Fluid Level), then the amount of load below zero load or above Fo from the Fluid Level is called a friction force unaccounted for by the wave equation.(mechanical friction is not modeled by the wave equation)

The presence of mechanical friction results in less production from the well due to reduced down hole stroke. Reduced pump displacement results in a higher than normal fluid level. Increased friction results in less efficient operation due to increased motor horsepower required to overcome extra frictional horsepower. Increased mechanical friction also results in an increase in downhole failures.

Rod Stretch

Rod stretch is a function of the lengths and diameters of various sections of sucker rods making up the entire rod string. In a vertical well the pump applies the fluid load at the bottom of the rod string, the increasing pump load results in the rod string elastically stretching in response to the applied load. A constant diameter fiberglass or steel sucker rod has a linear elastic change in length, stretch, in response to a proportional equal amount of change in load, this property is called the modulus of elasticity, E. A rod string in a well consists of one or more diameters, with the largest diameter rods typically located at the top of the rod string. The top rod diameter is large enough to elastically stretch to support the weight of all of the rods connecting from the surface to the pump; plus supports the pump loads applied at the bottom of the rod string. The diameter of the rod string decreases at depth as the weight of the rod string decreases, creating a tapered rod string. A normal design of a tapered rod string results in the maximum stresses being equal at the top rod of each of the different-sized sections of the tapered rod string. The tapered rod string is never loaded to more than 25% of the elastic tensile stress range and the sucker rods do not fail in tension due to a maximum load because the rod string is designed for a long operational life with the expected failure mode of fatigue. Stretch of the rod string occurs due to the fluid load being applied by the plunger to the bottom of the rod string during the upstroke. Rod unstretch occurs by the release or transfer of the fluid load from the rod string to the tubing during the down stroke. **Equation 1** defines the Coefficient of Rod Stretch, Kr, for a single rod of area **A** and is defined as the required load in pounds applied to the rod of length **L** to stretch the entire rod string equal to 1 inch.

$$Kr = AE/L \qquad \ldots \text{ Eq. 1}$$

Fig. 2 is an example of a stuck pump where the plunger is stuck at the bottom of the stroke. A 190 lbs/in composite Kr for the rod string can be calculated based on the rod string taper. In this well the plunger motion is negligible with both the plunger stroke (Sp) and plunger velocity equal to zero. The measured surface loads at the bottom of the down stroke are equal to the weight of the rods in fluid and the measured surface loads at the top of the stroke are equal to the weight of rods in fluid plus the force required to stretch the rod string equal to the stroke length. The surface pumping motion is sinusoidal with a total surface stroke length, S, of 64 inches. Starting at the beginning of the upstroke, point A, the polished rod will move upward in a sinusoidal fashion. The pump, however, will remain stationary, and the rods will gradually stretch based on the 190 lb/in spring constant for the entire rod string. In this example the amount of rod stretch of 64 inches will occur at the top of the stroke at point B. The pump is stationary from time at point A to time at point B and the plunger has zero velocity and zero displacement. At the beginning of the down stroke, point B, the pump still remains stationary, while the rod stretch decreases. At point A the rod stretch becomes zero at the bottom of the stroke, the cycle is complete and the stroke process is repeated. The rod string does not fail in this example of a stuck pump because the rod loading does not exceed the yield strength of the rod string.

Application of Down Hole Mechanical Friction Force

Fig. 3 is an example of a pump almost stuck due to downhole mechanical paraffin friction force, where the calculated pump loads are 3,350 lbs greater than the expected 7,040 lbs fluid load determined by the fluid level. A 111 lbs/in composite Kr for the rod string is calculated based on the rod string taper. This surface dynamometer card has a similar appearance to the stuck pump shown in **Fig. 2** where the surface dynamometer card load and position on the up and downs stroke plot parallel to the 111 lbs/in spring constant. In this well the mechanical friction due to the downhole paraffin mechanical friction is resulting in an 82% loss of the surface stroke to additional rod stretch. After this well was effectively treated for paraffin the down hole stroke increased from 18 inches to 75 inches with a corresponding increase in 91 barrels per day increased pump displacement.

Fig. 4 is an example surface and pump dynamometer card showing a downhole sticking force applied at some unknown depth, but not at the pump. At 105 inches on the up stroke the pump card load increases by approximately 1000 lbs, this increased pump load is calculated by the wave equation by transferring a mechanical force applied up hole through the rod string down to the pump depth. A 214 lbs/in composite Kr for the rod string is calculated based on the rod string taper. This 1000 lbs force is not applied at the pump and is not equal to 1000 lbs, but what is known is the polished rod load and position plot shows a different rod stretch. This additional downhole mechanical force causes the surface rod string stretch to change from 214 lbs/in to 339.9 lbs/in. The composite rod stretch can be calculated versus depth from the surface to any depths along the rod string, **Fig 5**. A 339.9 lbs/in composite Kr for the rod string is calculated based on the rod string taper in the well going from the surface to a depth of 5109 feet. The location of this mechanical downhole friction force would be applied at a depth of 5109 feet from the surface if the sucker rods were to stretch only from the surface to the depth of 5109 feet.

Fig. 6 is an example surface and pump dynamometer card for a well having excessive downhole mechanical friction applied to the rod string. The calculated 15,715 lbs pump load card load range is 48% greater than the expected pump load of 10,610 lbs. A 142 lbs/in composite Kr for the rod string is calculated based on the rod string taper, shown in **Table 1 Rod String Taper**:

	Top Taper	Taper 2	Taper 3	Taper 4	Taper 5
Rod Type	EL	EL	EL	SB	SB
Length	3162.00	4125.00	3025.00	32.50	300.00
Diameter	1.000	0.875	0.750	1.000	1.500
Weight	9145.1	9124.4	4910.9	94.0	1958.9

Fig. 7 is the same well's surface and pump dynamometer card shown in **Fig. 6** except the pull rod for the pump parted and no load is being applied to the rod string by the pump. The excessive down hole mechanical force is being applied at some unknown point in the rod string. The rod string is still attached because the average of the horse shoe load cell measured surface dynamometer loads appears to be equal to the weight of rods in fluid. The measured surface dynamometer card Kr slope is equal to a line drawn between the two points circled on the left side of the surface dynamometer card load versus position plot and equal to 1470 lbs/inch ((27430-16670)/7.27). The depth along the rod string to the point of the application of the excessive down hole mechanical friction force is where the measured surface Kr equals the Kr calculated from the surface down the rod string (Measured Kr = Rod Kr). To calculate the composite Kr remove rod segments from the bottom of the rod string in steps for the remaining rod string from the surface to the end of the rod string. The step by step removal of rod segments from the bottom toward the top is a quick technique to determine the depth to the point of the application of the excessive down hole mechanical friction force. **Fig. 8** shows the calculated rod spring constant of 1479 lbs/in at a depth of 1350 feet from the surface in the 1 inch sucker rod string section closely matches the 1470 lbs/in Kr determined from the surface dynamometer loads versus position. A defect in the tubing, casing or wellbore profile should be expected near the 1350 foot calculated down hole severe mechanical force location. **Fig. 9** shows that a 12,000 lb mechanical friction force is being applied to the rod string at a depth 1350 feet from the surface. This 12,000 lbs friction force is much larger than expected and the total load applied at the 1350 depth is the weight of the rods in fluid plus the mechanical friction force. Inspection of the deviation survey on 100 foot intervals shows at a measured depth of 1371 feet the dogleg severity to be 1.96 deg/100 ft. The location of the severe mechanical force being applied to the rod string can be accurately determined, by finding the depth to where the measured Kr is equal to the Kr calculated. Severe doglegs in the upper sections of any well should be avoided because the increased mechanical friction forces due to the dogleg severity will result in increased failures and high operating cost.

Stuffing Box Friction

Fig. 10 and **Fig. 11** display the measured surface dynamometer card acquired on a well after excessive stuffing box mechanical friction was applied at the surface by over tightening the stuffing box. The operator used a crescent wrench to apply excessive force to packing inside the stuffing box, increasing the normal polished rod horsepower by 15 percent from the normal 8 to the displayed 9.2 horsepower. The excessive stuffing box friction force applied to the polished rod at the surface increased the upstroke loads by approximately 500 lbs. and decreased the downstroke loads by approximately 500 lbs. If the friction force on the polished rod is not removed, then in **Fig. 5** the pump horsepower is increased from the normal 6 to 7.3 horsepower. Visually examining the pump card in **Fig. 5** the stuffing box friction has not been removed and the pump card loads on the upstroke are higher than the maximum expected fluid load and the pump card loads are plotted below the zero load line. Negative pump card loads plotted below the zero load line are usually thought to represent compressive loads acting at the pump and increased pump friction should increase downhole failures due to increased rod on tubing wear. In this example the negative pump loads are not acting at the pump, but are the extra stuffing box friction force applied at the surface that was not removed by the wave equation. When mechanical friction is not removed the shape of the pump card is changed from normal to show excessive loads on the up and down stroke of the pump card. These excessive upstroke and down stroke pump loads are called “unaccounted for friction forces” and this 500 lbs. extra friction force displayed in the pump card loads indicates the friction in the system is excessive. The stuffing box friction force is applied opposite to the direction of motion of the polished rod. In **Fig. 10** before the measured surfaces loads are processed using the wave equation, a software option is used to reduce the measured polished rod loads by 500 pounds of force on the upstroke when the polish rod is moving up and increases the polished rod loads by 500 lbs on the down stroke when the polished rod is moving down. This “500” pound force is entered by the user as an estimate to account for excessive stuffing box friction acting on the polished rod before processing the surface dynamometer card using the wave equation. The **Fig. 10** pump card shape with the appropriate amount of excessive stuffing box friction removed shows a normal shaped pump card with the loads near the expected maximum load and zero load lines.

In the analysis and identification of excessive mechanical friction acting on the rod string, if the stuffing box friction is not removed, then the pump card shape shows excessive loads on the up and down stroke. If the downhole pump card shape appears normal by the removal of the stuffing box friction and the pump card shape appears to be normal, then presence of excessive friction is not apparent. By using the stuffing box friction software option to remove this mechanical friction force, the pump card shape appears to show that there is no excessive friction problem during the operation of the well. Where excessive friction is present in the system, the use of the software option to remove the display of the excessive stuffing box friction, hides the visual fact that excessive friction is present.

Impact of Tubing Back-Pressure on Measured Polished Rod Loads

The practice of increasing the surface tubing pressure above the normal flow line pressure by using a back-pressure regulating valve to maintain a high producing tubing pressure is typically done to prevent unloading of gassy tubing liquids out of a sucker rod lifted well. When using the back-pressure regulating valve the surface tubing discharge pressure is higher as compared to the surface pressure if no back-pressure regulating valve were used. Increase tubing back-pressure allows pump action to be maintained when differential pressure across plunger is low due to tubing liquids lightened from gas entering (pumped into) the tubing. The higher surface tubing back-pressure results in increased pump discharge pressure, in increased tubing fluid gradient, in increased rod loading and in additional input horsepower from the prime mover. BUT using a back-pressure regulating valve on the tubing can prevent the liquids in the tubing from unloading and stopping pump action, when the tubing liquids are lightened to the point of flowing due to free gas being pumped into the tubing.

Fig. 12 shows a simplistic well head cut away, where the piston force created by the tubing back-pressure is acting across the seal of the stuffing box on the polished rod cross-sectional area. This piston force is equal to the tubing back pressure times the area of the polished rod and this force reduces the polished rod load measured at the surface (the measured polished rod load is reduced by this piston force equal to the back pressure times the area of the polished rod, **Fig. 12**). When the tubing pressure is low this piston force is not normally seen, because the weight of the polished rod is approximately equal to tubing pressure multiplied times the polished rod area. **Fig. 12** shows a 1.5 inch diameter 25 foot length polished rod weighing approximately 150 lbs, the weight of the polished rod is equal to the piston force at the stuffing box if the surface tubing pressure is equal to 85 psig. Once the tubing back-pressure exceeds approximately 85 psig, then a decrease in the measured polished rod loading is usually observed. This force piston should be accounted for when using the wave equation to calculate the pump card loads when the tubing pressure is greater than 85 psig, plus the polished rod length should be added to the rod string.

Table 2 – Back-pressure Test Surface Dynamometer Measurements

Tubing Head Pressure (Psig)	Card #	Peak Polished Rod Load Lbs	Min Polished Rod Load Lbs	% Rod Loading	Polished Rod HP	Strokes per Minute
None	2	30292	17186	77.1	23.0	4.02
200	6	30361	16625	77.3	23.6	4.02
400	6	30547	16019	77.8	25.0	4.02
600	6	30753	15480	78.3	25.9	4.00
800	6	31057	15190	79.1	27.0	4.00
Ratio		1.025	0.884	1.026	1.174	0.995

To study the impact of applying different amounts of back-pressure on the measured polished rod load a field test was performed using an accurate calibrated horseshoe load cell to acquire dynamometer data on a well with a special low friction stuffing box design to not apply excessive friction to the polished rod as tubing pressure was increased. A back-pressure regulating valve was used to increase the tubing pressure from the normal 54 psig to 200, 400, 600, 800 psig. **Table 2** shows as back-pressure increases, the peak polished rod increases by 2.5% and the minimum polished rod loading decreases by 11.6%, percent rod loading increased by 2.6%, polished rod horse power increased by 17.4% and motor speed slightly decreased by 0.5%.

Table 2 – Back-pressure Test Pump Card Calculations

Tubing Head Pressure (Psig)	Card #	FoUp Lbs	FoDn Lbs	Fluid Load on Pump (Lbs)	Effective Pump Stroke Inch	Effective Pump Disp. BPD	Input KW	System Eff %
None	2	5676	-384	6060	272.8	287.5	25.7	19.7
200	6	5779	-580	6359	270.5	285.2	26.4	19.2
400	6	5903	-1013	6916	267.7	282.2	27.6	18.4
600	6	6062	-1376	7438	264.1	277.2	28.6	17.8
800	6	6216	-1784	8000	260.0	272.8	29.7	17.1
Ratio		1.095	4.646	1.320	0.953	0.949	1.156	0.868

Table 2 shows as back-pressure increases the calculated pump load on the up stroke, FoUp, increased by 9.5%. The pump load on the down stroke, FoDn, became more negative by a factor of 4.646 or decreased from a -384 to a -1784 lbs. The height of the pump card (FoUp-FoDn) increased by 32% from 6060 to 8000 lbs. The effective plunger stroke was reduced by 4.7% due to the additional rod stretch required to pick up the increased pump load, resulting in a 5.1 % reduction in the effective pump displacement. Increased tubing back-pressure increased the input KW to the motor by 15.6 resulting in a 13.2% reduction of the overall system efficiency. As the tubing back-pressure is increased from no back-pressure to 800 psi, then the wave equation calculated pump card loads on the down stroke shift to a negative 1784 lbs below the zero load line. This negative shift results from the measured surface loads being lightened by the piston force acting on the polished rod not being accounted for by the wave equation calculations.

Table 4 – Back-pressure Polished Rod Piston Measured Compared to Calculated

	Average Pump Card Dnstroke Loads KLbs	Difference in Min Pump Load at Back-pressure Compared to Min Pump Load @ Normal Line Pressure	Average Tubing Pressure Psig	1.5 Inch Polished Rod Force PsiXArea Lbs	Piston Force on 1.5" PR Due to Back-pressure above Line (Psi x Area PR)
No Backpressure - #2	-0.515	0	53.6	94.7	0.0
200 Psig Backpressure - #6	-0.714	200	182.0	321.6	226.8
400 Psig Backpressure - #6	-1.122	608	362.4	640.4	545.7
600 Psig Backpressure - #6	-1.472	957	557.3	984.8	890.1
800 Psig Backpressure - #6	-1.847	1333	759.8	1342.7	1248.0

Table 4 shows the applied tubing back-pressure varies during a stroke as the back-pressure valve open flow area changes to maintain the spring-force set tubing pressure at the surface, for example 800 psi was the intended amount of back-pressure, but the average tubing pressure during stroke #6 was 759.8 psig. In **Table 4** the measured pump card piston force is equal to the pump card FoDn negative load shift at a specific back-pressure compared to the pump card FoDn load with no back-pressure. The average pump card load with no back-pressure was -535 lbs and the pump card load with 800 psig of tubing back-pressure was -1847 lbs, then the pump card piston force not accounted for by the wave equation is a negative 1333 lbs (the pump card load shift is due to a surface applied -1333 lbs force reducing the surface measured polished rod loads). The measured surface loads would be 1333 lbs higher if there were the normal 54 psig tubing pressure. This tubing pressure piston force is reducing the measured surface loads during both the up and down stroke. To calculate the surface piston force due to tubing back-pressure the measured or input tubing pressure dynamometer acquisition should be multiplied times the polished rod area. Then this force should be added to the measured surface loads prior to processing by the wave equation. **Fig. 13** plots the unaccounted for shift in the wave equation calculated pump card load on the down stroke compared to the tubing piston force applied to the polished rod area by the tubing back pressure. **Fig. 14** shows the impact on the pump card loads resulting from including the tubing back-pressure polished rod piston force. In **Fig. 14** the piston force is

not accounted for the pump card shifts below the zero load line by the tubing piston force indicating excessive friction on the rod string, when accounted for the pump card height increases due to additional fluid load acting on the rod string. Without accounting for the piston force acting on the rod string the sucker rod loading will be incorrectly calculated below the surface of the well, because the rod loading will be higher than measured.

Rods on the Side of Tubing in a Horizontal Wellbore

The pump card in effect becomes the trash can (unaccounted for wave equation loads are shown at the pump card) for any mechanical loads applied along the rod string not removed by the wave equation. An example of a negative shift in the pump card loads is not accounting for the weight of the rods that rest on the side of the tubing below the kickoff point in a horizontal wellbore. The wellbore deviation survey is required input information and the deviation survey should be input into the diagnostic software used to analyze measured surface dynamometer data acquired on a deviated well. **Fig. 15** shows the deviation survey for a well where the standard API 1.5 inch diameter pump is set 800 feet past the kickoff point into the horizontal section of the wellbore. The weight of the entire rod string in fluid is 13460 lbs with 800 feet of rods past the kickoff point there is 621 lbs (4.6%) of rod weight in fluid resting on the side of the tubing. If the deviation survey were not input into the diagnostic software then the pump card loads would shift downward by this extra 621 lbs of weight. When using the vertical well wave equation model, if the weight of each rod is adjusted to only include the vertical component of the rod string weight, then the pump card will be positioned on the down stroke resting on the zero load line as shown in **Fig. 15**. The weight of rods past the kickoff point in a horizontal well is usually small when compared to the total weight of the rod string. The mechanical friction from rod-on-tubing contact below the kick-off point is normally small because the rod weight below the kickoff point is small. If no adjustment of the rod weight is made, then the pump card will plot below the zero load line by the weight of the rod in fluid resting on the side of the tubing. This negative pump card down stroke load shift will be considered to represent increased downhole friction, but the pump load shift below the zero load line will only be due to the weight of the rods resting on the side of the tubing below the kickoff point not being accounted for in the application of the vertical wave equation model.

Conclusion

This paper discusses mechanical friction, true vertical rod weight, and the tubing back-pressure piston forces. The vertical wave equation model is not designed to account for these mechanical forces. The deviated wave equation model and/or the vertical wave equation model do not properly account for a mechanical force applied to the rod string, when the mechanical forces are due to down hole sticking at a point along the rod string. The magnitude and location a force applied, like a stuffing box friction force, must be known to remove the force acting on the rod string. Mechanical forces applied to the rod string by a severe dog-leg are normally large, if the sticking/drag occurs near the surface where the rod loadings are large. Frictional forces from paraffin can act along the rod string length near the surface, at the cooler temperatures where paraffin deposition can occur. OR a paraffin mechanical friction can be applied at a point where a dogleg plus paraffin deposit can result in down hole sticking. If the force is not accounted for in the application of the wave equation, then the unaccounted for force ends up in pump card loads and can be called “a force unaccounted for by the wave equation”. These external forces impact measured surface loads, down hole stroke length, horsepower and plunger velocity, plus calculated rod loading at the pump and other locations along the rod string.

References

1. Gibbs, S. G., and Neely, A. B.,” Computer Diagnosis of Down-Hole Conditions In Sucker Rod Pumping Wells”, SPE1165, Journal of Petroleum Technology, January 1966.
2. Gibbs, S. G., “Design and Diagnosis of Deviated Rod-Pumped Wells” SPE22787, Dallas, TX, October 6-9, 1991.

Nomenclature:
Fo - Fluid load applied to Rods by plunger - Lbs
S - Surface Stroke Length - Inch
Kr - Rod String Spring Constant – Lbs/In
A – Area of a Sucker Rod
E – Modulus of Elasticity
L – Length of a rod string segment

Figure 1 – Damping Factor Correct Adjustment

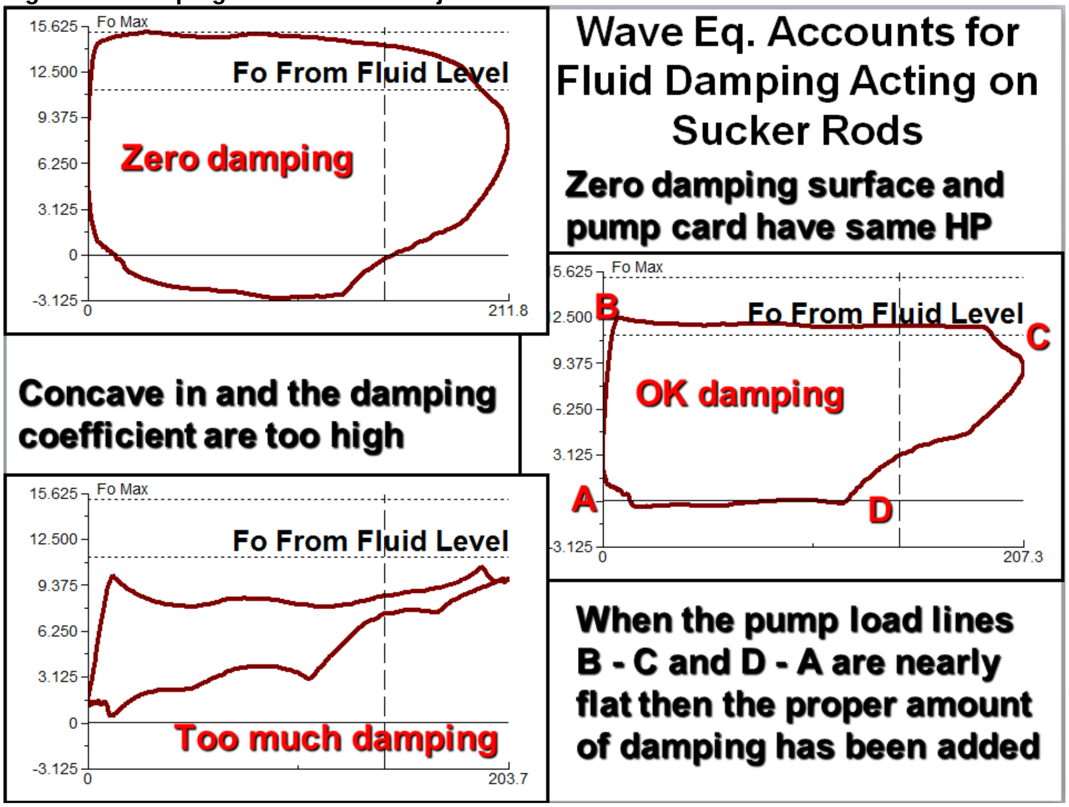

Figure 2 – Rod Stretch of a Stuck Pump

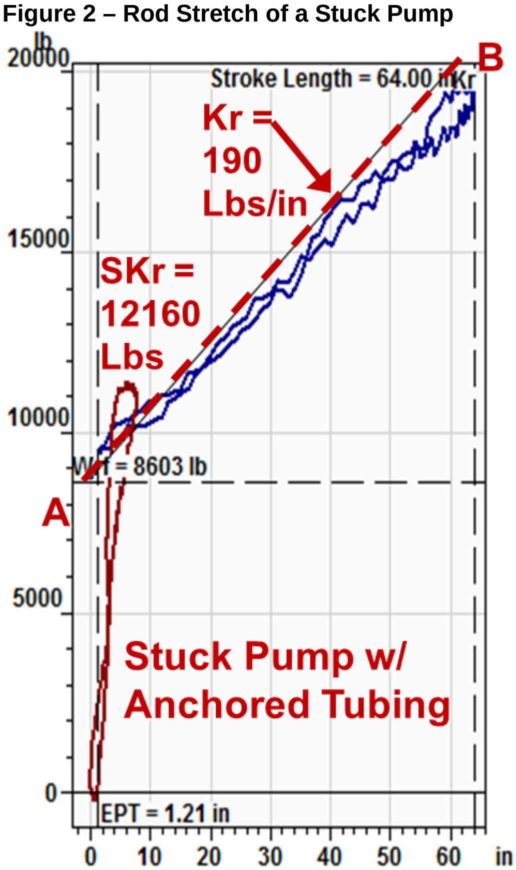

Figure 3 – Paraffin Almost Sticks Pump

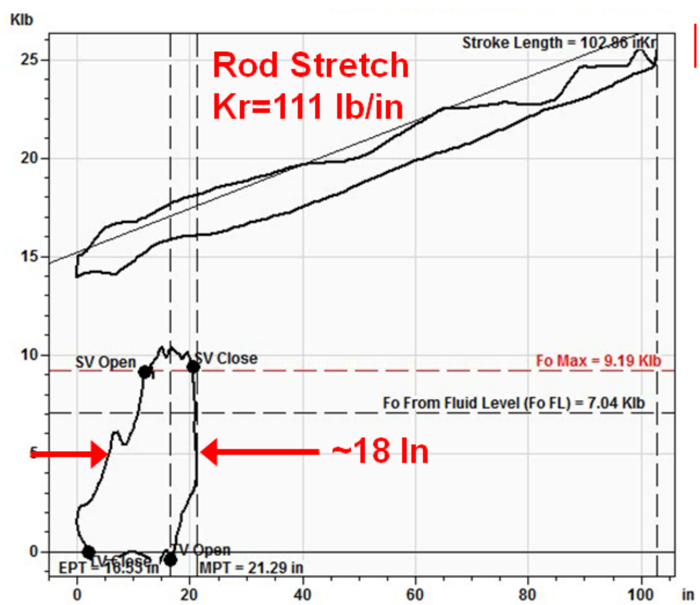

Figure 4 – Mechanical Friction Force Not at the Pump

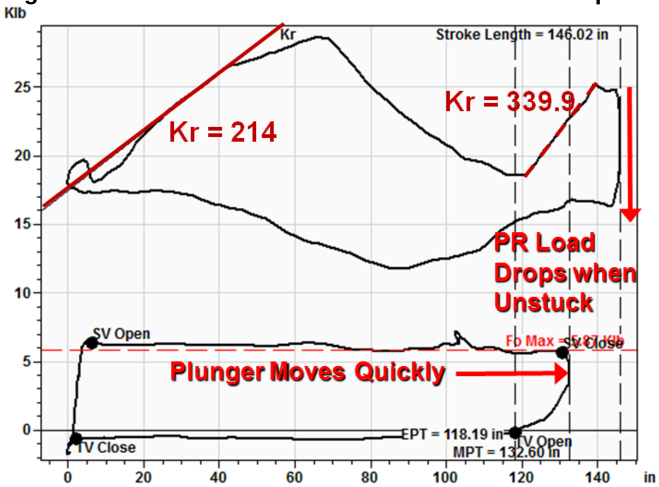

Figure 5 - Rod String Kr versus Depth

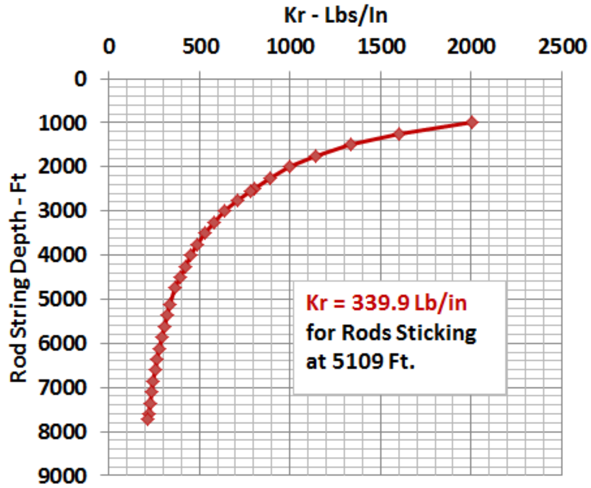

Figure 6 – Excessive Mechanical Friction Force

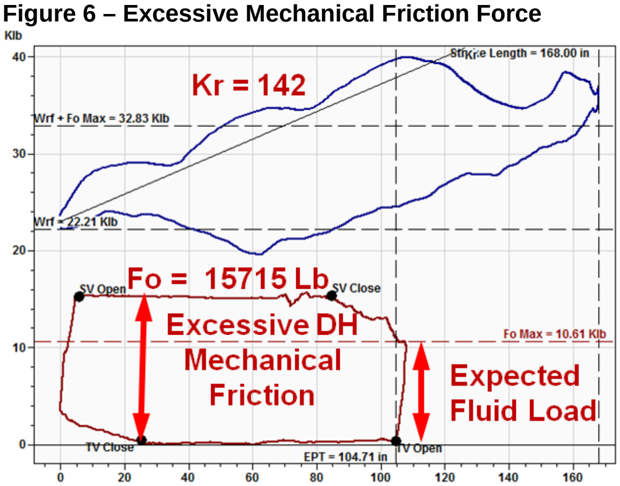

Figure 7 – Pull Rod Parts at the Pump

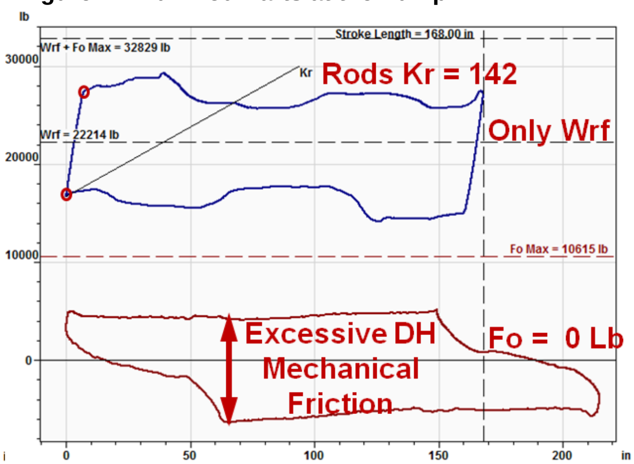

Figure 8 – Kr Measured Equals Kr Calculated

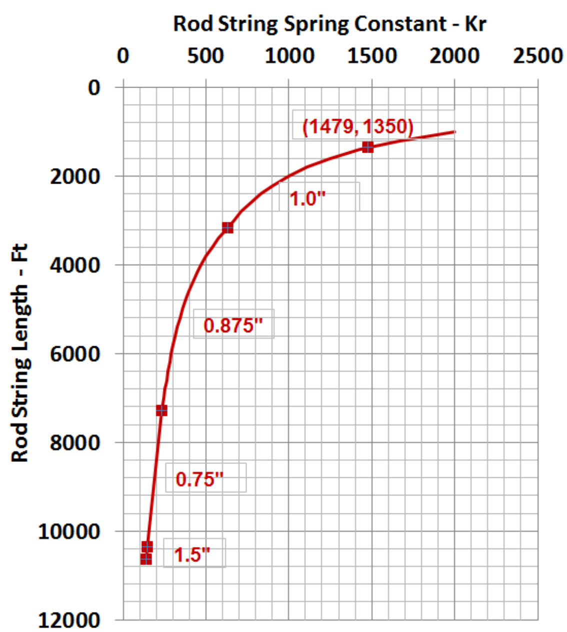

Figure 9 – Rod Load at Depth of 1350 Feet

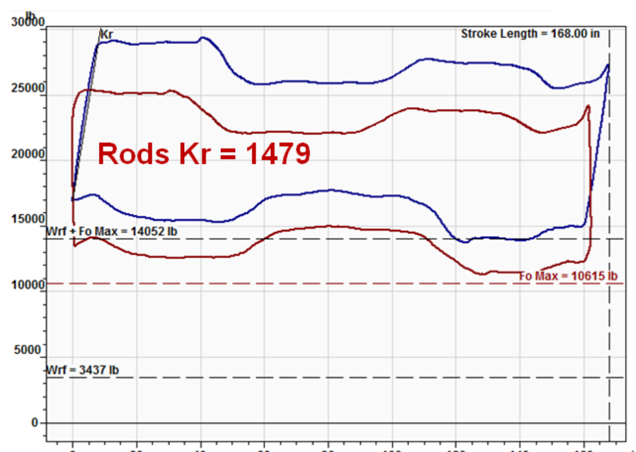

Fig. 10 – Adjusted 500 Lbs. Stuffing Box Friction

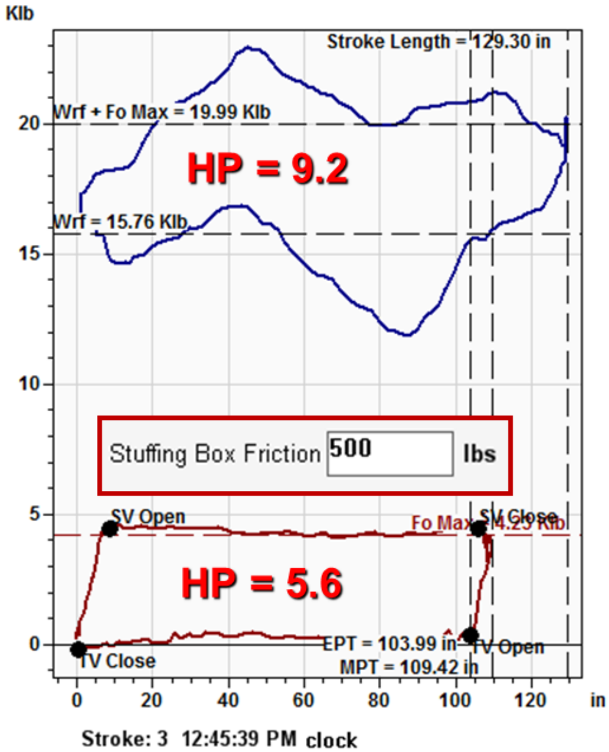

Figure 11 – Not-Adjusted Over Tight Stuffing Box

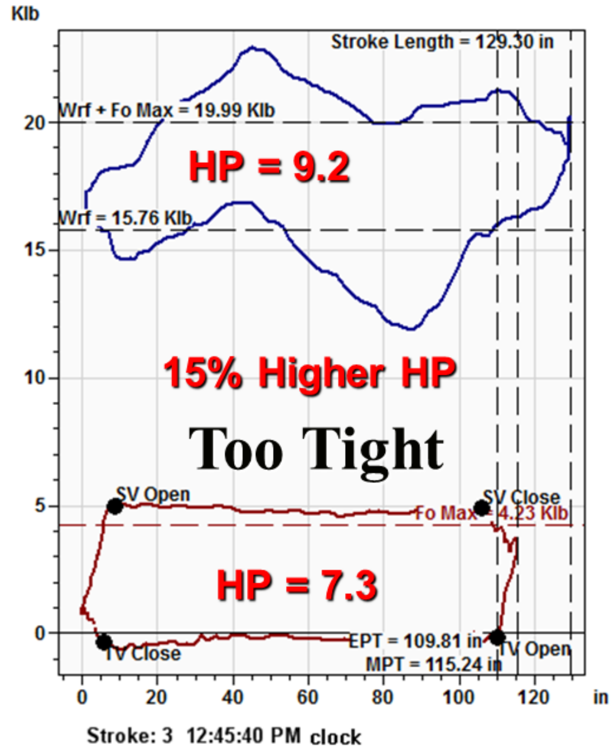

Figure 12 – Well Diagram of Piston Force versus the Polished Rod Weight

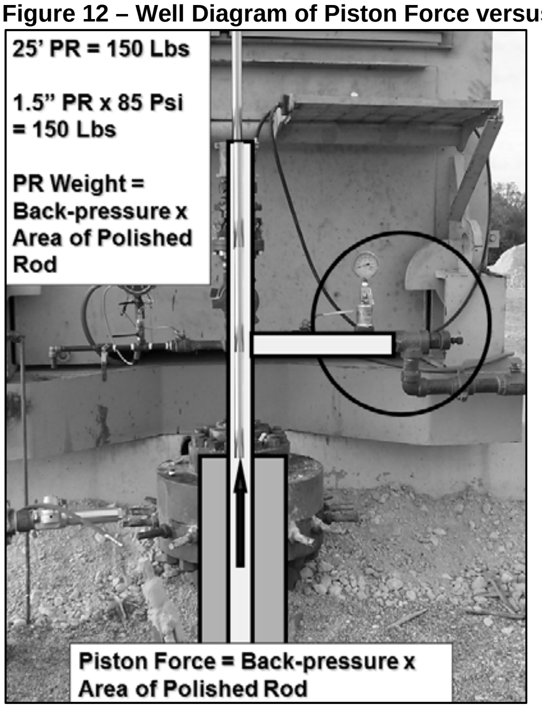

Figure 13 – Compare Measured and Calculated Piston Force

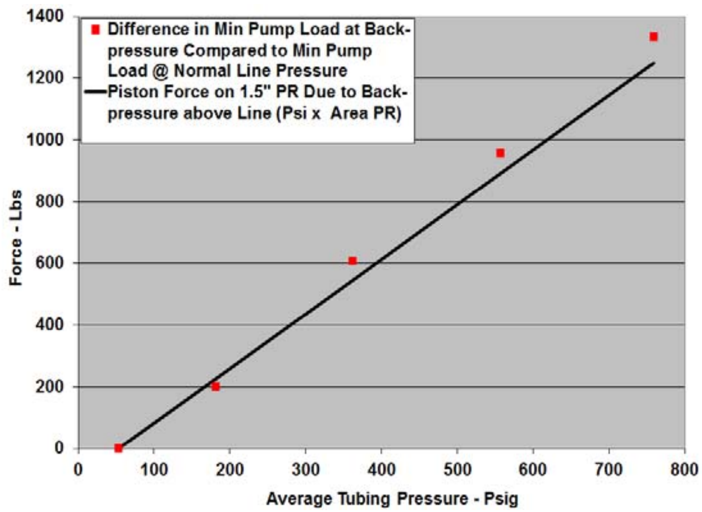

Figure 14 – Rod Loading Below the Surface Not-Corrected versus Corrected for Piston Force

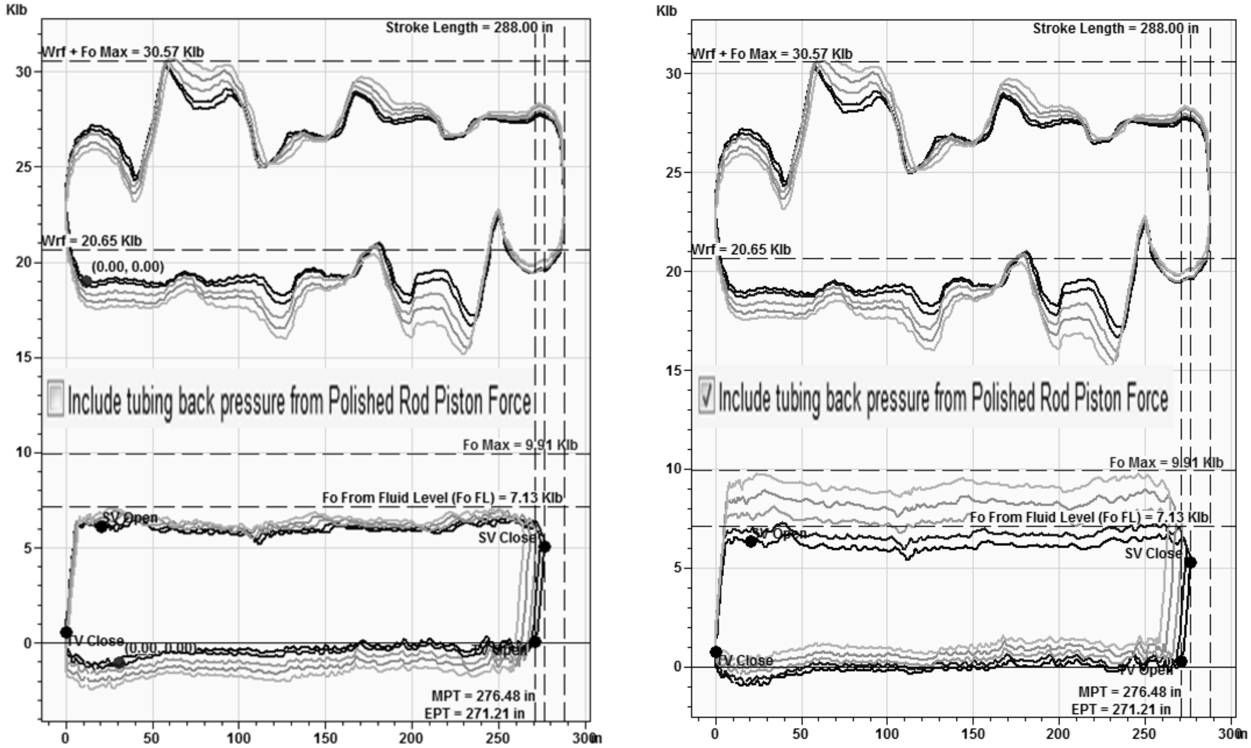

Figure 15 – Must Reduce the Rod String Weight by Weight of Rods Resting on Tubing Wall