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Compressive Forces Causing Rod Buckling in Sucker Rod Pumps and Using Sinker Bars to Prevent Buckling

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Compressive Forces Causing Rod Buckling in Sucker Rod Pumps and Using Sinker Bars to Prevent Buckling

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Dedication

To my parents, brother and devoted fiancée for their endless love and support

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Abstract

Compressive Forces Causing Rod Buckling in Sucker Rod Pumps and Using Sinker Bars to Prevent Buckling

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Sucker rod pumps has been the most commonly used pumps in the petroleum industry. Therefore, many studies associated with sucker rods focuses on maximizing the rod life. Rod buckling is a leading problem which causes concentrated wear on tubing wall, immediate failure in rod strings, and shortens fatigue life of the string. This study fundamentally consists of a review of the literature on compressive forces causing rod buckling in sucker rod pumps and using sinker bars to prevent buckling.

The study initially addresses defining rod buckling, and then continues with the studies on analyzing the static forces acting near pump in the literature. Subsequently, the critical loads causing rod buckling, and the various approximations to estimate these critical loads are discussed. Then, the comparisons of the measured and calculated critical loads in the literature are presented. Next, the two of the most commonly experienced

buckling types in the sucker pumps, sinusoidal buckling and helical buckling, are discussed. An example study on developing a model to estimate compressive forces acting on the pump plunger is reviewed to illustrate the importance of the parameters, such as surface roughness of pump, valve diameter, and pump geometry. Lastly, using sinker bar which is the most practiced method in the industry to prevent rod buckling is extensively discussed and demonstrated.

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Chapter 1: Introduction

Texas, the second largest state in the United States of America, is also well-known for its energy resources. The hundreds of oil wells can be seen while driving across Texas. These wells are mostly produced by a common type of pump which is called the beam lift pump or the horse head pump because of its shape looking like a horse head.

The beam lift pump consists of three main mechanical parts, surface pump, rod string (sucker rods), and subsurface pump (pump plunger and barrel). The surface pump is the visible part of the beam lift, located at the surface, which drives the rods up and down through an oil well by converting the electric power into motion energy. Rod string is composed of hundreds of 25-feet steel or fiber-reinforced plastic (FRP) rods, and transmits the motion of the surface pump to the pump plunger. The total length of rod string can be thousands of feet. In Figure 1, a short sucker rod and its parts are illustrated. This transmitted strokes cause the pump plunger firstly to move the fluid into the pump barrel and then to push it up from the pump barrel. Figure 2 represents the schematic of a conventional beam lift pump and its major parts.

The life of rods and pumps plays an important role for the profitability of an oil well because oil-producing companies have a natural desire to profit more by using rods and pumps for a longer period of time. However, the continuous up-down motion of these mechanical parts weakens their endurances in time due to the rapidly changing forces on rods and pumps. These forces, mostly depending on the frequency of the motion and the materials which are used to manufacture, determine the life of these beam lift pumps. Therefore, selecting the most suitable material and estimating the optimum pumping speed for each unique well by considering that the faster the pump moves, the easier the pumping system tends to fail are necessary to maximize the life of rods and pumps.

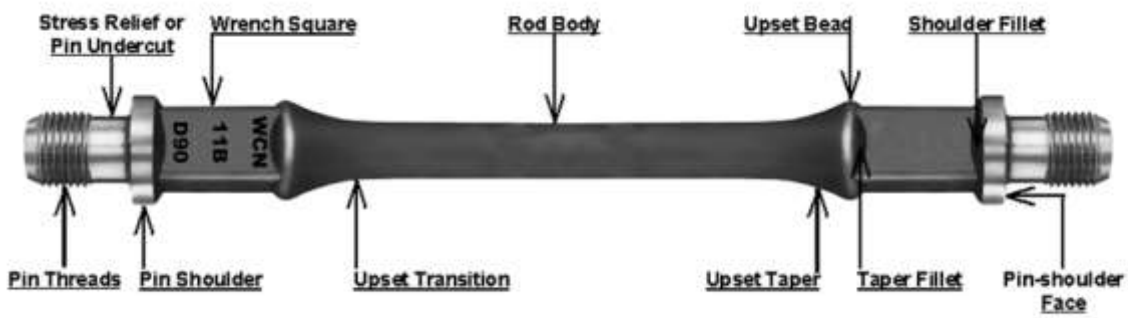

Figure 1: Sucker Rod Terminology (The Beam Lift Handbook by Paul M. Bommer and A. L. Podio, 2015)

The analysis of the compressive force acting on pumps, which has been widely covered in the petroleum industry, is one of the most substantial issues in sucker rod pumps which directly affect the rod and pump life, and failures in pumping systems. This study focuses on the compressive forces associated with rod buckling and the methods to prevent buckling.

The study begins with defining rod buckling, and proceeds to a review of the analysis of the static loads acting near the pump. Then, the study addresses the critical loads causing rod buckling, and the approaches in the literature to calculate these loads. Comparing the measured critical load data and the calculated critical load values by different approaches, the reliability of these methods is investigated. Next, sinusoidal buckling and helical buckling are discussed as helical buckling is assumed as the most common form of rod buckling in the literature. Later on, the effects of the parameters, such as viscosity, valve design, and surface roughness are discussed in the perspective of the models developed to predict the dynamic forces acting on pump. Analyzing the compressive forces acting on the bottom of the sucker rod, the most commonly used methods to prevent rod buckling are presented. Finally, the method of using heavy sinker bar to prevent rod buckling is explained in detail, and the studies in the literature are discussed.

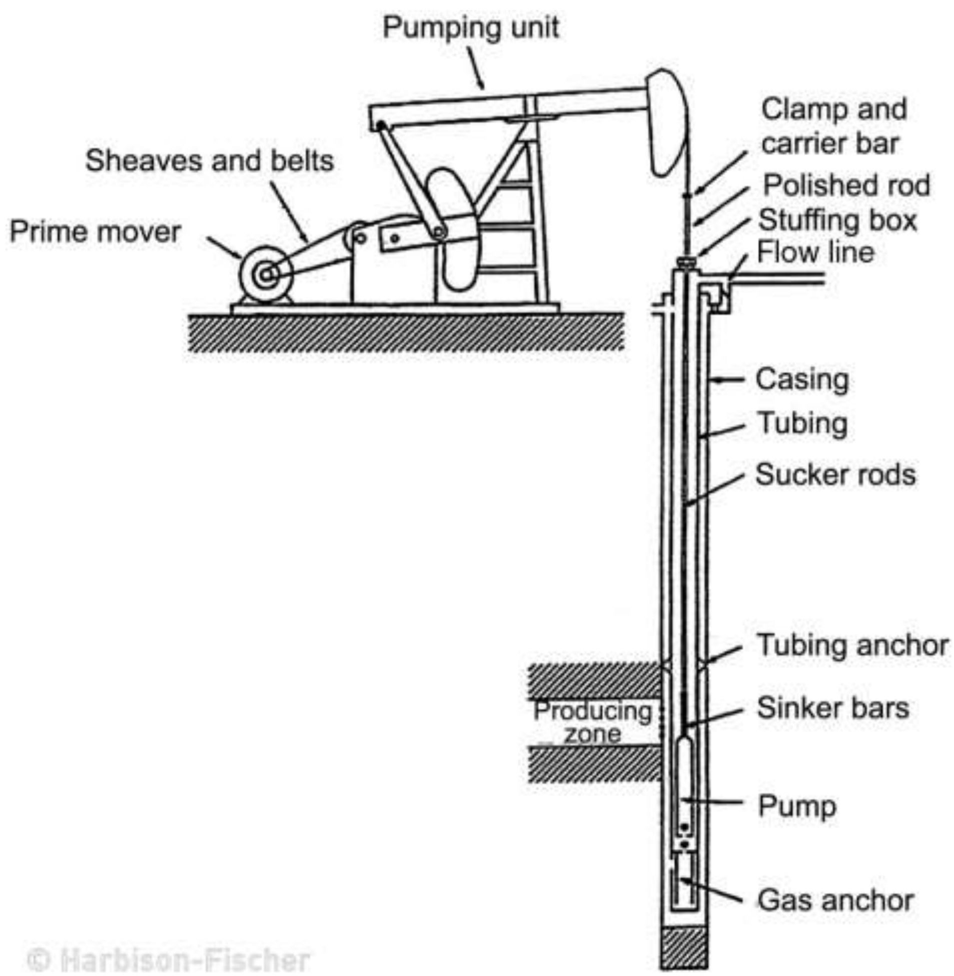

Figure 2: Schematic of conventional pumping unit with major components of the sucker-rod-lift system (Courtesy of Harbison-Fisher).

Chapter 2: Rod Buckling

Rod buckling, a leading cause of pumping failures, is a universal concern of all oil-producing companies and operators. It has been experienced that rod buckling could easily engender lost revenue during downtime and expensive operating costs. The most of the companies and the operators aim to reduce failures in rod pumping systems in order to diminish the effects of costly failures. Therefore, rod buckling has been extensively studied in the literature.

As sucker rods are manufactured to handle thousands of tensile stress, a small compressive load can easily bend the sucker rod string. This bending motion is called as rod buckling. In this chapter, the previous researches on defining rod buckling, static loads on pump plungers, critical buckling loads, buckling types, hydraulics of sucker rod pumps, compressive forces acting on the bottom of the sucker rod pumps, and the methods to prevent rod buckling are extensively investigated.

2.1 DEFINING ROD BUCKLING

In order to define buckling, determining the all aspects of buckling is needed. The motion of rod alone does not cause buckling. Therefore, the consideration of buckling should include both the pump's dynamic and rod inertia. The dominant contributing forces which can buckle rods are the loads caused by the motion of the pump, fluid properties, and pump clearance. Most of the work in the literature proves that hydrostatic has no impact on true compressive force acting on the rods (Lubinski et al. 1962). In other words, there is equilibrium between the load which tends to straighten the rod, and the load which acts to compress the rod. As a result, the equilibrium abolishes the hydrostatic effects in the rod buckling (Mendenhall and Ott, 1995).

The resistance occurred at the subsurface pump is one of the main factors which directly affects buckling. Since the rod string is not loaded by fluid during down-stroke, the effect of this resistance mostly appears in the lower level of rod string while the rest upper part of the rod string is held in tensile stress because of the gravitational forces on the attached rods in the lower rod string. The point splitting these upper and lower portions of rod strings is called as neutral point, which is illustrated in the Figure 3. As a secondary effect, the compressive load propagates through rod string which is called force wave propagation. This compressive load takes place in the lower level of neutral point (Bishop and Long, 2010; Mendenhall and Ott, 1995).

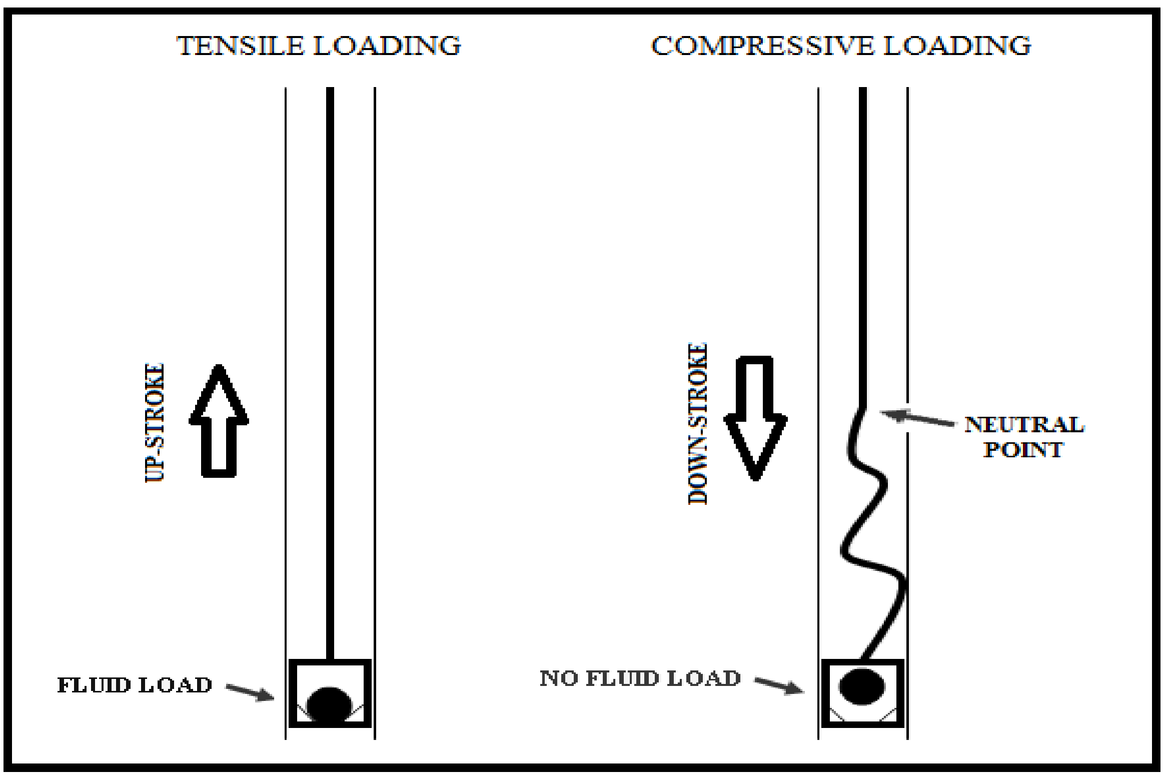

Figure 3: Load cycle of stroke and illustration of neutral point (Bishop and Long, 2010).

To conclude, it is necessary to estimate the compressive force in order to achieve a useful rod buckling analysis. Defining and analyzing rod buckling require better understanding of neutral point because the principle of preventing buckling is related to the methods to keep the neutral point at the lowest level of rod string.

2.2 ANALYSIS OF SUCKER ROD LOADS

Prior to estimating the compressive force, it is essential to analyze the forces acting on rod string. Since the working principle of the sucker rod pumps is based on reciprocating in the well, load on the sucker rods are fluctuating. Let us consider that pump plunger begins moving up from the lowest level which it can reach in the pump barrel. As pump moves up, travelling valve tends to close, and fluid begins being lifted by pump plunger. Summation of fluid weight, buoyed rod weight and dynamic loads gives the total load during the upstroke. Therefore, the maximum (peak) load occurs during upstroke which is shown in the Textbook of Bommer and Podio as given in Equation (2.1).

$$W_{max} = W_{rf} + F_o + W_{Dup} \quad \text{(2.1)}$$

where W_{max} is the maximum rod load during the upstroke, W_{rf} is the buoyed weight of rod, F_o is the weight of fluid, and W_{Dup} is the dynamic load during the upstroke.

When the pump plunger begins moving down in the barrel, standing valve closes, and increase in pressure occurs in the barrel. At some point of early time of downstroke, pressure in the barrel reaches a critical value which opens the travelling valve, and fluid load is transferred to the standing valve. Since fluid weight is hold by the standing valve, and the direction of the dynamic load changes due to the downward motion of the rod string, the minimum load supported by the sucker rods is shown as in Equation (2.2).

$$W_{min} = W_{rf} - W_{Ddown} \quad (2.2)$$

where W_{min} is the minimum rod load during the downstroke, and W_{Ddown} is the dynamic loads during the downstroke.

W_{rf} can be calculated based on Archimedes' Principle. According to Archimedes, the buoyed weight of the rods is equal to the weight of the rods in air minus the weight of the fluid displaced by the rod string. The displaced fluid weight and the buoyed rod weight can be computed in Equation (2.3) and Equation (2.4), respectively.

$$W_{FD} = W_r \frac{\rho_f}{\rho_r} = W_r \frac{62.4\gamma_f}{\rho_r} \quad (2.3)$$

$$W_{rf} = W_r - W_{FD} = W_r \left(1 - \frac{62.4\gamma_f}{\rho_r}\right) \quad (2.4)$$

where W_{FD} is the weight of the fluid displaced by the rods, W_r is the weight of the rod in air, ρ_f is the density of fluid, ρ_r is the density of rod, γ_f is the specific gravity of fluid referenced to fresh water.

The weight of the fluid, which is lifted by the travelling valve during upstroke and the standing valve during downstroke, can be estimated by multiplying the static pressure of the produced fluid on the pump plunger by the cross-sectional area of the plunger. Equation (2.5) gives the fluid load acting over the plunger.

$$F_o = \frac{62.4}{144}\frac{g_c}{g}\gamma_f L_n A_p = 0.34\gamma_f L_n {D_p}^2 \quad (2.5)$$

where g is acceleration due to gravity, g_c is gravitational conversion constant, L_n is net fluid lift from the working fluid level, A_p is plunger area, and D_p is plunger diameter.

Since determining the dynamic forces acting on sucker rod pump will be discussed in detail in the following sections, Equations (2.1) and (2.2) can be considered as general expressions for the maximum and minimum loads. Dynamic forces basically consist of the frictional and viscous forces inside the tubing string, such as the mechanical pump friction, the friction between the rods and fluid, the fluid and tubing, the tubing and rods, and the friction due to the fluid flowing through valves (Bommer and Podio 2015).

2.3 CRITICAL BUCKLING LOADS

One of the questions whose answer is so valuable for the industry is how compressive load causing rod buckling can be determined. According to Mendenhall and Ott (1995), measuring, modeling, or empirically determining are the only ways to determine the magnitude of the compressive load which depends on the specific rod string installation. Although the most accurate way of determining the load is considered as measuring the loads as a function of time and position at the pump by downhole load cell device (DHLC), it is not feasible enough to perform routine analysis because it is both expensive and time consuming. However, modeling, which is the most popular alternative, needs to be calibrated by DHLC measurements. Therefore, well-calibrated software can be utilized effectively for routine field works. The third method, empirically determining, is less expensive and time consuming than the other two methods, but it is not recommended because the complex dynamics of rod string is disregarded.

Leonhard Euler was the first mathematician investigating buckling and critical load required to buckle an ideal, elastic slender column. In his investigation, it is assumed that an ideal slender column is perfectly straight and compressed by load acting through the centroid of the cross section. According to Euler's study, this compression can be described by one of three forms of equilibrium; stable, neutral, and unstable.

In stable equilibrium, the force not exceeding critical load is applied, and the perfect straight shape of column does not alter. Under these conditions a small lateral force will bend the column which will return to its straight form if the lateral force is removed. If the applied force reaches to the critical force, a lateral force will produce a deflection, and will not disappear when the lateral force is removed. This equilibrium shapes neutral buckling. Equal to or greater loads than the critical load cause instability in column which is called unstable buckling (Long and Bennett, 1996).

Euler solved the differential equation for the rod deflection curve with the free-fixed boundary conditions assuming that the one end of the column is fixed while the other one is free to move radially. The best condition describing the physical shape of the bottom rod is free-fixed condition. The general expression for the critical load defined by Euler is represented in Equation (2.6):

$F_{CR} = (\pi^2 E\ I)/(4L^2)$,……..…………….….……………………………………..…(2.6)

where E is modulus of elasticity, I is moment of inertia, L is the length of column. The moment of inertia for a circle can be expressed as a function of the diameter of circle in Equation (2.7):

$I = (\pi D^4)/64$,……………………………………………………………..………(2.7)

According to Long and Bennett, it is possible to develop Equation (2.6) based on the end configuration of the column. Figure 4 illustrates these configurations whose critical loads are as follows:

$F_{CR(fixed)} = (4\ \pi^2 E\ I)/(L^2)$,……………..……………………………….………..(2.8)

$F_{CR(hinged)} = (\pi^2 E\ I)/(L^2)$,…………………...…………...…………………………...(2.9)

where $F_{CR(fixed)}$ is the critical load for the fixed end configuration, $F_{CR(hinged)}$ is the critical load for the hinged end configuration.

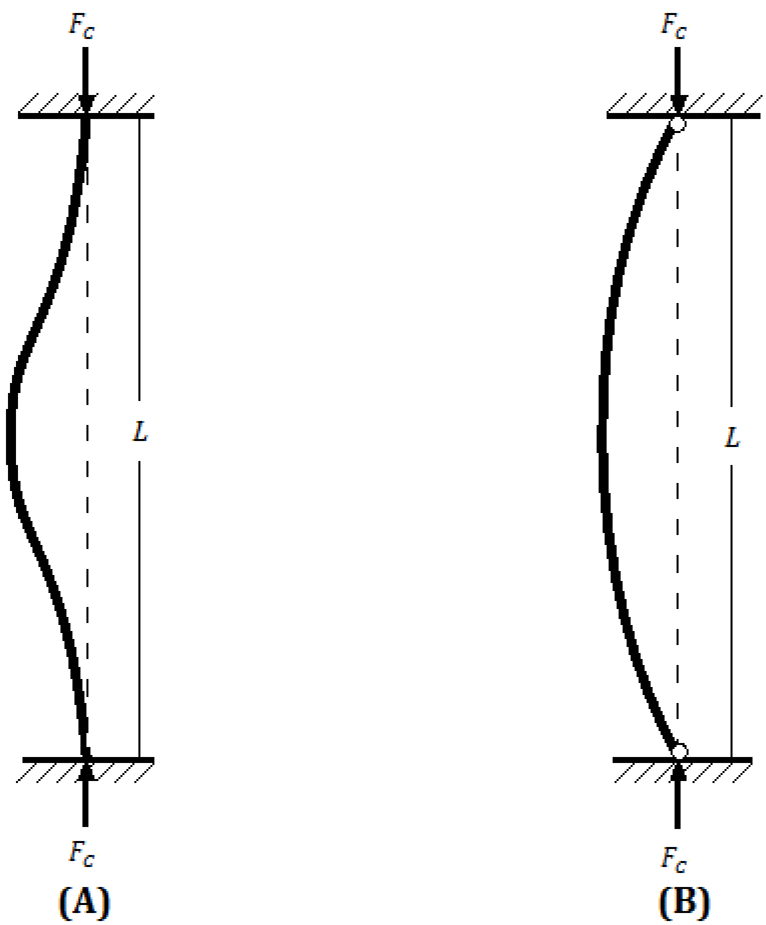

Figure 4: Configurations of fixed end and hinged end. (A) and (B) respectively illustrate fixed end and hinged end configurations (Long et al., 1996).

The Equations above also reveal that the critical load of a fixed-end configuration is four times greater than the critical load of a hinged-end configuration which can be seen in Table 2.1. On the other hand, Leo, Pattillo, and Studenmund present Equation (2.10) to estimate the critical compressive force to cause a rod to buckle.

$$F_{CR} = -[0.795\pi^2 EIw^2]^{\frac{1}{3}}, \qquad (2.10)$$

where w is the buoyed weight of the sucker rods per foot.

Rod Diameter	Buckling Loads in water (lbf)	Calculated Euler Loads (lbf) Fixed End	Calculated Euler Loads (lbf) Hinged End	Measured Buckling Loads (lbf)
1/2''	-	-41	-10	-
5/8''	-22.9	-100	-25	-
3/4''	-37.2	-208	-52	-23
7/8''	-56.2	-385	-96	-162
1.0''	-80.2	-657	-164	-
1-1/8''	-109.8	-	-	-
1-1/4''	-145.5	-	-	-
1-3/8''	-186.0	-2348	-587	-641
1-1/2''	-230.0	-3325	-831	-
1-5/8''	-	-4579	-1145	-

Table 2.1: Comparison of calculated and measured buckling forces, and Euler loads.

In Table 2.1., it is indicated that calculated buckling loads by Equation (2.10) result in less values than measured buckling loads by Long and Bennett except for the ¾'' rods. Furthermore, the calculated Euler loads for fixed-end rods are much greater than measured loads as the calculated Euler loads for hinged-end rods closely approximate the measured loads. Ott and Mendenhall assert in their conference paper that the predicted critical load values by commercially available software programs are much more than those measured by the downhole tool with strain gauges which provides actual forces at the pump.

2.4 BUCKLING TYPES

2.4.1 Sinusoidal Buckling

Euler's critical load equation describes sinusoidal buckling which takes form of sine wave. Also, Equation (2.11) is an expression for any other sinusoidal shape assumed by the rod.

$$F_{CS} = [(2n-1)\pi/(2\,L)]^{\,2} E\,I\,, \qquad (2.11)$$

where F_{CS} is the critical load for sinusoidal buckling, and n is the number of full sine waves in rod section.

Euler's equation for the rod with free-fixed conditions is applicable to a finite length rod; however, Equation (2.12) developed by Dawson and Paslay applies to an infinite rod in an inclined borehole by assuming same boundary conditions (Mendenhall and Ott, 1995).

$$F_{CS} = E\,I\,(n'\,\pi/\mathrm{L})^2 + \{w\ \sin(a)\,/r\}\{L/(n'\,\pi)\}^2\,, \qquad (2.12)$$

In Equation (2.12), a is the borehole inclination from vertical, and n' represents the number of half sine waves in the buckled rod section, and can be expressed as follows:

$$n' = L/l_w, \qquad (2.13)$$

$$l_w = \pi\,(E\,I\,/\,k)^{1/4}, \qquad (2.14)$$

$$k = w\ \sin(a)\,/r, \qquad (2.15)$$

where l_w is the length of half-wave for rod string, k is elastic foundation constant, and r is radial clearance between rod diameter and tubing wall.

2.4.2 Helical Buckling

The force required for transition of sinusoidal buckling to helical buckling is negligible, so it is commonly assumed that rods buckle helically (Mendenhall and Ott, 1995). The minimum load buckling the rod into a helix is given as

$$F_{CH} = (8\ E\ I\ w\ \ sin(a)/r)^{\ 1/2}, \qquad (2.16)$$

Dawson and Paslay discuss the two experimental solutions to critical helical buckling force presented by Dellinger et al. in 1983, and Lubinski and Woods in 1953. Dellinger's equation, Equation (2.17), gives results close to Equation (2.16) in the range of fairly high stability as its results increase to 50% higher than Equation (2.16) in the range of lower stability.

$$F_{CH} = 2.93\ (E\ I)^{0.479}\ w^{0.522}\ \ (sin(a)/r)^{0.436}, \qquad (2.17)$$

The experiential equation of Lubinski and Woods, Equation (2.18), also verifies the theoretical solution (Mendenhall and Ott, 1995).

$$F_{CH} = 2.85(E\ I)^{0.504}\ w^{0.496}\ \ (sin(a)/r)^{0.511}, \qquad (2.18)$$

The difference between these two equations is most probably caused by the fact that Woods and Lubinski used the data directly while Dellinger et al., reconstructed the experimental data to fit the equation (Dawson and Paslay, 1984).

2.5 HYDRAULICS OF SUCKER ROD PUMP

The hydraulics of the sucker rod pumps generally trigger major problems in pumping system, such as partial pump fillage, gas interference, fluid pound, and compressive loads on the valve rod. However, these effects are integrated into pump friction constants in rod string design software because of lack of sufficient models. These frictions can be examined in three categories; resistance against the downward motion of the pump plunger due to resisting fluid flow through the inside of the plunger, mechanical frictions caused by metal to metal sliding, and fluid resistance between the plunger and the barrel (Cutler and Mansure, 1999). Therefore, Cutler and Mansure discussed dynamic and geometric details to estimate the frictional losses in the model which was developed by Sandia National Laboratories, and their effects on the calculations.

The first investigated issue in their study is pressure drop across valves which has critical effect on pump filling, gas leakage, and compression loading of the valve rod. By testing various sizes of balls and seats in the valves with open and closed cages, they conclude that using a smaller ball and larger seat ID decreases the pressure losses in both closed and open cage designs, and that using the high efficient valves adequately improves the performance of the valves compared to the less efficient valves. As pressure drop across entire pump is investigated, the pressure loss in the travelling valve is, unsurprisingly, found more than in the standing valve because the area allowing fluid flow in the travelling valve is comparatively smaller than in the standing valve. According to Cutler and Mansure; however, losing more pressure in the pump exit than in the travelling valve was not an expected situation, and can be seen as an opportunity to improve the pump design.

As most parameters in downhole are full of uncertainties, the uncertainty of fluid viscosity in downhole and the effect of viscosity have always been extensive study area to researchers. Cutler and Mansure notice that at low to medium flow rates changing the viscosity from 0.1 cP to 100 cP has very little effect on pressure losses calculated in the model. While this effect becomes greater at high flow rate, it still stays in a small range of change. Although it was stated that only a few experiments investigating the effect of the viscosity had been conducted in that study, replacing the water at 1 cP with Weeks Island Crude at 15 cP only increased the pressure loss by 6%.

Other issues investigated by Cutler and Mansure are effects of surface roughness, valve design, and ball chatter. First of all, according to the results of their model, surface roughness has little effect if the absolute roughness does not exceed 1000 microinches. For the roughness values greater than that limit, pressure loss increases significantly.

Secondly, in 1984 Allen and Svinos asserted that enlarging the I.D. of the valve seat, using open cages rather than closed cages, and increasing I.D. of the pump plunger reduced the resisting forces during downstroke. As the results obtained from the model by Cutler and Mansure are consistent with this study, it also emphasizes the connection between reducing the changes in flow area and the decrease in the resisting forces. Therefore, for an accurate prediction of pressure drop along with pump, the fine details of the components in the pump should be included in models, and the significant effects of these details in pressure drop should be considered when designing new pump components.

Thirdly, the effect of ball chatter was tested by Cutler and Mansure for various combinations, such as cases with the ball and seat, with just the seat, and without the ball or the seat. The results show that pressure drop rises up rapidly at low flow rates and decreases after reaching a critical flow rate. This flow rate can be defined as a critical level at which the ball stops chattering. It can be concluded that the size of the check valves should be determined to prevent the valve from partially opening. The partially opened valve is one of the factors increasing the pressure losses significantly.

To sum up, developing a model for analyzing flow in sucker rod pumps provides better understanding of hydraulics of pumps. The effects of the variable parameters, such as viscosity, valve design, and surface roughness can be analyzed to more accurately predict pressure drops and to reveal the reasons causing the rod buckling.

2.6 COMPRESSIVE FORCES ACTING ON THE BOTTOM OF THE SUCKER ROD

In the previous sections, sucker rod loads were analyzed, and the maximum and minimum loads were expressed in Equation (2.1) and Equation (2.2), respectively. To analyze the tendency of the sucker rod to buckle, more detailed expression is required for the dynamic force acting near the bottom of the rod string. This section will discuss the dynamic forces caused by fluid friction (drag) and mechanical friction in the annulus between the pump plunger and barrel. During upstroke, these forces cannot result in buckling because the forces are tensional, but same forces result in compressive effect on the bottom of sucker rod during downstroke since these dynamic forces act opposite to the sucker rod motion.

Some of the load analyses in the literature ignore the friction force in the travelling valve due to the pressure differences in the pump plunger (Lea et al., 1995). However, tests and model calculations show that pressure difference is required to move the fluid through travelling valve (Cutler and Mansure, 1999). Therefore, this force during downstroke is equal to the cross-sectional area of the plunger times the pressure difference between the ends of the pump plunger as shown in Equation (2.19).

$$F_{TV} = (P_b - P_a)A_p = -\pi {D_p}^2 \Delta P/4, \qquad (2.19)$$

where P_b is the pressure below the pump plunger, P_a is the pressure above the pump plunger, and ΔP is the pressure difference across the plunger.

To include the viscous fluid drag between the sides of the plunger and the barrel, Lea and Nickens use the following formula:

$$F_d = -\pi R_i L_p \frac{\partial P}{\partial z} C_R - 2.088 \times 10^{-5} \frac{2\pi R_i L_p \mu}{C_R} V_p, \qquad (2.20)$$

where R_i is the radius of the wall corresponding to inner wall of the plunger/barrel gap, L_p is plunger length, $\partial P/\partial z$ is the pressure gradient of fluid, C_R is radial clearance, μ is fluid viscosity, and V_p is plunger velocity.

After the force due to the pressure difference through travelling valve is added to Equation (2.20), the compressive buckling force, F_c, is as follows:

$$F_c = -\pi {D_p}^2 \Delta P/4 - \pi R_i \frac{(D_b - D_p)}{2} \Delta P - 1.312 \times 10^{-4} \frac{D_p L_p \mu}{(D_b - D_p)} V_p, \qquad (2.21)$$

where D_b is the diameter of pump barrel.

$\pi R_i(D_b\text{-}D_p)/2$ can be replaced by $\pi(D_b^2\text{-}D_p^2)/8$ because the $\pi R_i(D_b\text{-}D_p)/2$ product approximates the one half the area of the annulus in the parallel plate. Then, Equation (2.21) reduces to Equation (2.22).

$$F_c = -\pi \frac{(D_b^2 + D_p^2)}{8} \Delta P - 1.312 \times 10^{-4} \frac{D_p L_p \mu}{(D_b - D_p)} V_p, \qquad (2.22)$$

Equation (2.22) has two terms which are directly proportional to the pressure difference and the fluid velocity. Cutler and Mansure prefer using the pump discharge rate rather than ΔP and V_p, so these terms are rearranged in the form of a function of the rate. After testing pressure drop across the travelling valve and plunger with changing flow rates, an empirical equation is created as follows:

$$\Delta P \approx K Q^2, \qquad (2.23)$$

where the coefficient, K, is in psi/(bbl/day)^2, pressure drop, ΔP, is in psi, and pump discharge rate, Q, is in bbl/day.

To determine the maximum buckling rod, the peak velocity should be taken into account in the calculation. However, if sinusoidal motion is assumed, the peak velocity can be related to the average velocity as in Equation (2.24).

$$V_{average} = \frac{V_{peak}}{2\pi}\left(\int_0^{\pi} \sin[\theta] d\theta + \int_{\pi}^{2\pi} 0 d\theta\right) = \frac{V_{peak}}{\pi}, \qquad (2.24)$$

where $V_{average}$ is average velocity, V_{peak} is peak velocity, and θ is the coordinate of sinusoidal motion of surface pump.

Now, the peak plunger velocity can be rewritten as $V_{average}$ is equal to Q/A_p.

$$V_p\Big|_{peak} = \pi \frac{Q}{A_p} = \frac{Q}{26.7\, D_p^2} \frac{ft\, in^2 day}{\sec bbl}, \qquad (2.25)$$

where the peak plunger velocity is ft/sec, the pump discharge rate is in bbl/day, and the plunger diameter is in inches. The substitution of Equation (2.23) and (2.25) into Equation (2.22) gives the final form of the buckling force due to pressure and plunger velocity effects.

$$F_c \approx -\pi \frac{(D_b^2 + D_p^2)}{8} K Q^2 - 1.312 \times 10^{-4} \frac{D_p L_p \mu}{(D_b - D_p)} \frac{Q}{26.7\, D_p^2}, \qquad (2.26)$$

A slightly modified form of this equation is provided in the Beam Lift Handbook by including $-F_m$ for the mechanical (Coulomb) friction between plunger and barrel in addition to Equation (2.27).

$$F_c = -\pi \frac{(D_b^2 + D_p^2)}{8} \Delta P - 4.906 \times 10^{-6} \frac{L_p \mu Q}{(D_b - D_p) D_p} - F_m, \qquad (2.27)$$

Contrary to the experimental approach of Cutler and Mansure to the pressure drop through travelling valve, Bommer and Podio present Equation (2.28) to estimate the pressure loss by using Bernoulli's equation to account for changes in elevation, flow velocity, and irreversible frictional losses.

$$\Delta P = 2v^2 \frac{L_p}{D_{TV}/12} f \frac{\rho_f}{g_c(144)} + \frac{1}{2} v^2 e_v \frac{\rho_f}{g_c(144)} + 0.433\gamma_f L_p, \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.28)$$

where v is maximum fluid velocity through plunger, L_p is the overall plunger length, D_{TV} is the travelling valve seat diameter, f is the Fanning friction factor, ρ_f is the fluid density, g_c is the mass to force conversion, γ_f is the liquid specific gravity, and e_v is the entrance and exit friction factor which is represented by Equation (2.29).

$$e_v = 0.45(1-\beta) + \left(\frac{1}{\beta} - 1\right)^2, \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.29)$$

where β is the ratio of travelling valve diameter to barrel diameter that can be calculated by using Equation (2.30).

$$\beta = {D_{TV}}/{D_b}, \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.30)$$

Cutler and Mansure (1999) also developed a model to predict the pressure drop due to flow through the pump. The five steps of this well-explained model in the Appendix of their paper are dividing the pump into logical nodes at each change in geometry, determining Reynolds number at each node for each flow rate, determining the Friction Factor at each node for each flow rate by solving iteratively, finding the irreversible fluid friction losses between each node by considering sudden or gradual contraction and enlargement in the pump, and finally determining the total pressure loss between each set of nodes.

2.7 PREVENTING BUCKLING

The determination of the compressive force is necessary to compare with the critical buckling force. If the maximum value of compressive force exceeds the critical buckling load, whether the shape of buckling is sinusoidal or helical should be defined. Based on the buckling shape, the possibility of fatigue can be evaluated by estimating the stresses on the rod string, and it can be concluded whether or not the compressive loads are acceptable. In such situation that the compressive loads are unacceptable, surface pumping parameters should be initially investigated because some minor changes in surface pumping parameters, such as slowing pumping speed or reducing stroke length can reduce the rod buckling effect. Also, these changes can usually provide a better solution in a financial point of view and can be more cost effective than changing down-hole parameters. However, changing the surface pumping parameters is only practical if it can be accomplished without significant reductions in oil and gas productions (Bishop and Long, 2010).

If these changes do not resolve the rod buckling problem, other alternatives, using a larger clearance, shorter barrel, larger TV, redesigning the rod string, or adding sinker bars should be taken into account to try to avoid possible buckling (Lea et al., 1995, Bommer and Podio 2015).

Chapter 3: Using Sinker Bar to Prevent Buckling

This chapter focuses on using sinker bar to prevent buckling and its effects on compressive force at the top of the pump plunger. The purpose of this common method is to replace the lower part of rod string with heavy weight sinker bars to increase the rigidity of rod string (Bishop and Long, 2010). In Equation (2.7), the moment of inertia for a circle is proportional to its diameter. Therefore, larger diameter of same material tends to have higher moment of inertia. In other words, increasing the diameter and weight of rods drastically heightens the critical buckling load as the relation can be seen in Equations (2.6) and (2.7). Furthermore, sinker bars can reduce the length of the rods affected by compressive force since their weight drives the dynamic neutral point closer to the pump plunger. Because, in general, it is difficult to model short sections of heavy weight sinker bar, a well-calibrated software program should be utilized for modeling the dynamics of the rod string to insure whether there have been inappreciable changes in the system, or not (Mendenhall and Ott, 1995).

Sinker bars can be considered as sucker rods with larger diameter which help keep the rod string straight and in tension by providing concentrated weight above the pump. Table 3.1 shows the difference between the weights of sinker bars and sucker rods. It can be seen that the weights of sinker bars are quite greater than those of sucker rods which are indeed proportional to their cross-sectional areas. Since this extra weight due to replacing the sucker rod with sinker bars can overcome the compressive load acting at the bottom of the string, sinker bars are commonly used to prevent buckling in the sucker rod pumps.

The general idea of using sinker bars is to move the neutral point downward to the closest possible point to pump plunger. As it is discussed in Section 2.1, the neutral point is a point at which negative loading begins to occur in rod string. Figure 5 illustrates the effect of using sinker bar on the position of neutral point. As the neutral point approaches to the bottom of rod string, the effects of rod buckling will be reduced.

Size	Weight in air (lbf/ft)	Weight in water (lbf/ft)	Moment of Inertia (in^2)	Critical Length (ft)	Critical Buckling Force (lbf)
5/8"	1.114	0.971	0.007	23.3	-22.6
3/4"	1.634	1.425	0.016	26.1	-37.2
7/8"	2.224	1.939	0.029	28.9	-56.1
1"	2.904	2.532	0.049	31.6	-80.1
1-1/8"	3.676	3.205	0.079	34.2	-109.7
1-1/4"	4.172	3.638	0.120	37.8	-137.4
1-3/8"	5.049	4.403	0.175	40.2	-177.2
1-1/2"	6.008	5.239	0.249	42.6	-223.4
1-5/8"	7.051	6.148	0.342	45.0	-276.6
1-3/4"	8.178	7.131	0.460	47.3	-337.0
2"	10.68	9.313	0.785	51.7	-481.1

Table 3.1: Critical Buckling Conditions of Sucker Rods and Sinker Bars.

It can be seen in Table 3.1 that the critical buckling force of a sinker bar in a fluid with a specific gravity of 1 is much greater than that of a sucker rod. For example, the compressive load of 100 *lbf*, which buckles a 1" rod string, can be overcome by a 1-1/4" sinker bar. As Equation (2.10) gives the critical buckling forces, the critical lengths necessary to buckle a rod can be calculated as shown in Equation (3.1).

$$L_C = \left[0.795\pi^2 \frac{EI}{w}\right]^{\frac{1}{3}}, \qquad (3.1)$$

Then, for a specified rod string, neutral point can be calculated by Equation (3.2) if the compressive force is predicted by a model or measured where F_c is the axial compressive force. Comparing the critical length of this specified rod and the neutral point leads to discover if the rod will be buckled under the compressive force. Applying to the same example given above, the neutral point of the 1" rod string is calculated as 39.5 feet above the bottom of rod string with the assumption of a compressive force of 100 lbf. The critical length of 1" sucker rod is estimated as 31.6 feet in Table 3.1, and it is less than the neutral point; therefore, the lower steel rods are predicted to buckle.

$$L_{NP} = \frac{|F_{CR}|}{w_R(1-0.127\gamma_f)}, \qquad (3.2)$$

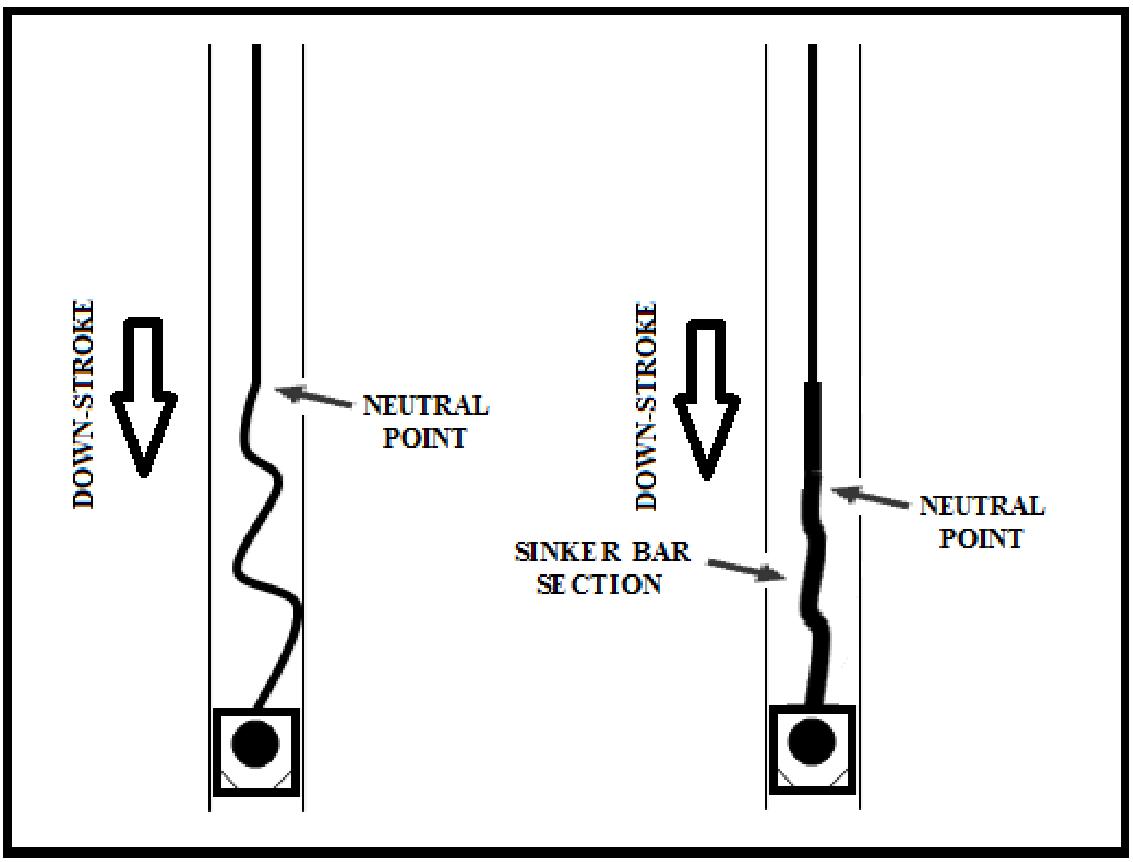

Figure 5: The position change of neutral point with using sinker bar section (Bishop and Long, 2010).

If using sinker bar is considered as the method to prevent buckling in this situation, the length of an alternative sinker bars can be estimated by Equation (3.3). This equation is based on the difference between the compressive force and the critical force necessary to buckle the rod above the sinker bar. On the other hand, Lea et al. (1995) provide a similar equation to calculate the length of the sinker bar. Because it is based on only compressive force, Equation (3.4) provided by Lea and others overestimates the necessary length of the sinker bar.

$$L_{SB} = \frac{|F_c - F_{CR}|}{w_{SB}(1-0.127\gamma_f)}, \quad (3.3)$$

$$L_{SB} = \frac{|F_c|}{w_{SB}(1-0.127\gamma_f)}, \quad (3.4)$$

If we go back to the previous example, the length of 1-1/4" sinker bars which 1" sucker rods is replaced with is estimated as 5.5 feet by Equation (3.3) while Equation (3.4) gives a value of 27.2 feet for the length of the sinker bars. The reason of these different results is that the length calculated by Equation (3.3) targets to lower the compressive force acting at the bottom of 1" rod taper to its critical buckling force, -80.1 lbf.; however, Equation (3.4) completely cancels out this compressive load.

In theory, these two values do not seem making a difference, but the effects of using longer sinker bars due to the additional viscous force in annulus between sinker bars and tubing is open to further investigations and experiments. This additional viscous force increases the rod load during up-stroke, and decreases it during down-stroke, therefore; the length of the sinker bars directly has an impact on the rod life. It may even cause buckling in sinker bars because of increasing compression. On the other hand, the effects of sinker bar diameter on tubing failures are studied by Bishop and Long (2010).

In their paper, it is stated that utilizing the largest diameter sinker bar for the available internal diameter of the tubing resulted in reduced tubing wall loss. Therefore, minimizing the length of sinker bar is as significant as maximizing the rod diameter which increases the contact area between tubing and rod, and reduces the concentrated wear on tubing wall. However, it must be also investigated if decreased flow area in annulus increases the drag forces, so compressive forces acting on the bottom of the rod string.

Chapter 4: Conclusions

As low oil price has been a great concern in the petroleum industry for the last two years, rod buckling can easily result in the excessive well expenses by causing tubing wear, fatigue failures, and decreased rod life. There are many studies in the literature which investigate the compressive forces causing rod buckling and solutions to prevent rod buckling.

In this report, it is concluded that using sinker bars to increase the rigidity of rod string is the most practiced method in the industry. It has been shown that sinker bars reduced the effect of buckling if changing surface parameters, such as pumping velocity and stroke length was not practical which would be less expensive and time consuming. However, determining the true compressive forces has a key role to adjust the length of sinker bar. Several commercial software have been developed by the industry to estimate the compressive force; however, they should be calibrated with field data and downhole measurements to be able to represent each specified oil well.

On the other hand, there are issues about using sinker bars because viscous force between tubing and sinker bar can cause additional compressive force that can be a topic for further research. These viscous loads need an extensive study since it has been asserted in the literature that using the largest available sinker bar diameters reduced tubing wear and failures significantly.

Glossary

a: borehole inclination from vertical [degrees]

A_p: plunger area [inches^2]

C_R: radial clearance [inches]

D: diameter of column [inches]

D_b: barrel diameter [inches]

D_p: plunger diameter [inches]

D_{TV}: travelling valve seat diameter [inches]

e_v: entrance and exit friction factor [-]

E: modulus of elasticity [30.5x10^6 psi for steel]

f: Fanning friction factor [-]

F_c: total buckling force [lb_f]

F_{CH}: critical load for helical buckling [lb_f]

F_{CR}: critical load [lb_f]

$F_{CR(fixed)}$: critical load for the fixed end configuration [lb_f]

$F_{CR(hinged)}$: critical load for the hinged end configuration [lb_f]

F_{CS}: critical load for sinusoidal buckling [lb_f]

F_d: drag force [lb_f]

F_o: fluid weight [lb_f]

F_{TV}: frictional force due to fluid flowing through travelling valve [lb_f]

g: acceleration due to gravity [ft/sec^2]

g_c: gravitational conversion constant, [32.174 lb_m.ft/(lb_f.sec^2)]

I: moment of inertia [inches^4]

k: elastic foundation constant [psi]

K: empirical pressure coefficient [psi/(bbl/day)^2]

l_w: length of half-wave for rod string [inches]

L: length of column [inches]

L_c: critical length [ft]

L_n: net fluid lift from the working fluid level [ft]

L_{NP}: neutral point [ft]

L_p: plunger length [ft]

L_{SB}: total length of sinker bars [ft]

n: number of full sine waves in rod section [-]

n': number of half sine waves in rod section [-]

P_a: pressure above the pump plunger [psi]

P_b: pressure below the pump plunger [psi]

r: radial clearance between rod diameter and tubing wall [inches]

R_i: radius of the wall corresponding to inner wall of the plunger/barrel gap [inches]

Q: pump discharge rate [bbl/day]

v: maximum fluid velocity through plunger [ft/sec]

$V_{average}$: average velocity [ft/sec]

V_p: plunger velocity [ft/sec]

V_{peak}: peak velocity [ft/sec]

w: buoyed weight per ft. of the sucker rods [lb_f/ft]

w_{SB}: air weight per ft. of the sinker bars [lb_f/ft]

w_R: air weight per ft. of the sucker rods [lb_f/ft]

W_{Ddown}: dynamic loads during the downstroke [lb_f]

W_{Dup}: dynamic load during the upstroke [lb_f]

W_{FD}: weight of the fluid displaced by the rods [lb_f]

W_{max}: maximum rod load during the upstroke [lb_f]

W_{min}: minimum rod load during the downstroke [lb_f]

W_r: rod weight in air [lb_f]

W_{rf}: buoyed rod weight [lb_f]

β: ratio of TV diameter to barrel diameter [-]

ΔP: pressure difference across the plunger [psi]

γ_f: specific gravity of the fluid referenced to fresh water [-]

μ: fluid viscosity [cP]

$\partial P/\partial z$: pressure gradient of fluid [psi/ft]

ρ_f: density of the fluid [lb_m/ft^3]

ρ_r: density of the rod [lb_m/ft^3]

θ: coordinate of sinusoidal motion of surface pump [degree]

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