WELL WEIGHING – A LOST ART?

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ABSTRACT

There is no substitute for going out to a pumping unit and gathering data from a dynamometer, amp clamp, motor rpm sensor, and fluid level device in order to fully analyze a well and have as complete an understanding as possible. The physical act of stacking out the well and attaching the horseshoe load cell along with the associated peripherals is becoming a lost art. In this study, we investigate the advantages in obtaining a thorough well analysis the old fashioned way and give examples of how a dynamometer and fluid level analysis outweigh any other type of study that can be performed on a well to obtain quantitative data on all the equipment, from the prime mover to the pump.

INTRODUCTION

When analyzing a well, we suggest the following methodology[1] in order to fully optimize the pumping system. First and foremost, one must investigate the possibility of more production potential. Following this a complete breakdown on equipment loading should be done. This includes calculating the loading on the prime mover, the gearbox, the unit structure, the rod string, and the pump. Using the position and loads that are obtained by the respective transducers, the behavior of the pump can be obtained via a surfeit of wave equation techniques which are now commonplace in the industry. Once the behavior has been quantified, production can be inferred by computing the amount of oil, water, and gas (both free and in solution) that is produced. Fallout from calculating the production from the pump card is pump intake pressure estimation. A fluid level shot should also be made in order to estimate a casing gas rate as well as acquire another measurement of pump intake pressure by different method. While the fluid level is an extremely important aspect of analyzing a well and by far the most widely used means of obtaining a bottom hole producing pressure, due to constraints beyond the authors' control it will not be included in the main part of the discussion of this paper. By combining these measurements and calculations, a very comprehensive analysis can be obtained for the system.

PRODUCTION POTENTIAL

The well's potential should be calculated at the present pump depth if it is not already in a pumped off state. This can be done several ways. The first step is to determine if the well is pumped down and/or could the pump be lowered to increase the well's production. Knowing the correct static reservoir pressure is important in calculating the correct inflow performance relationship for a given well. If the well is assumed to have too high of a static pressure, the predicted production potential will be too low and vice versa. Also the correct method should be used. Vogel's method is used when the well makes at least 15-20% high gravity oil with free and/or dissolved gas, and it is pumping below the bubble point pressure. If the well has a high water cut, say above 80-85%, with little gas, or is pumping above the bubble point, then the constant PI method is the preferred method to determine how much production potential the well has. This method has a tendency to overestimate, especially when used on a gassy well.

When analyzing a gassy well, obtaining the minimum producing pressure on the reservoir may be limited by gas interference. Therefore a good gas separation scheme is often necessary to produce the well more effectively. In addition, these types of wells often have a low productivity index (PI). As mentioned above, the PI of a gassy well is best modeled by Vogel's method which is a declining PI curve. Thus, if these two facts are not properly considered, actual production potential will be substantially less than anticipated.

Take for example a well with an assumed static pressure of 2000 psi that is below the original bubble point pressure and the pump intake is at mid-perfs. We have a production rate measurement of 889 BWPD and 9 BOPD at 1343 psi pump intake pressure with little gas production. Following Vogel's method, the calculated maximum production is about 1779 BPD. Pumping the well down to a pump intake pressure of 100psi would amount to a production rate of 1757 BPD. See Figure 1.

Following the constant PI method, we have a calculated maximum production of about 2734 BPD and a predicted production of 2597 BPD with a pump intake pressure of 100 psi. See Figure 2.

In this case, the well would probably be best modeled by the constant PI method. Both methods were listed here to show the large differences in the predicted production amounts as the well is drawn down to a producing pressure of 100 psi.

If a static reservoir pressure is not known, a table of assumed static pressures versus production rates is often presented. The upper limit of the static pressure is usually based on a salt water gradient to the reservoir depth. The calculated additional production will most likely be less than what would actually be attained if the well was pumped down.

In reality, a well is probably best modeled by a multi-rate inflow performance relationship. Having individual relationships for the oil, gas, and water will allow one to determine if the water cut will increase or decrease as the well is pumped down. For example, if the water seems to be coming from a higher pressure zone than the oil, then the oil cut should increase as the well is pumped down.

Using the results from the inflow performance relationships, one can determine the economical repercussions of whether or not attempting to increase production would be beneficial.

EQUIPMENT LOADING

The loading of all the equipment on the surface and downhole can be computed by incorporating the measurements listed in the introduction. Beginning with the prime mover, we can tell if it is overloaded mechanically by calculating the power required during the stroke and comparing it to the horsepower rating of the motor. We can also determine the thermal loading by calculating the RMS current and comparing that to the amp rating on the motor. It is important to note that a lightly loaded motor less efficient than a heavily loaded motor. If a UHS motor is implemented at the well, confirm that the lowest possible torque mode is used to reduce rod and gearbox loads. The speed changes during a pumping cycle are fallout of measuring RPM. With these values, we can implement an inertia torque calculation which yields a more accurate gearbox loading. These benefits are amplified when using the aforementioned UHS motor or variable speed drive (VSD). This will be discussed later.

Examining the surface dynagraph, the maximum polished rod load can be compared to the structural rating of the pumping unit to determine the structure loading.

The torque loading on the pumping unit gearbox can be measured in two ways. A simple approximate way is to obtain torque-speed curves for the motor in use and multiply the instantaneous torque measured at the motor by the total reduction to obtain the instantaneous torque at the gearbox slow speed shaft. The primary method that is easily executed in the field is using the geometry of the pumping unit along with a measurement of the existing counterbalance effect at the polished rod. This effect can be translated into a moment by using the geometry and the polished rod position at which the measurement was taken, along with knowing whether the unit was on the upstroke or the downstroke. Using this moment, we can calculate the net torque on the gearbox during the entire stroke using the API approved formula for net torque on the gearbox. This method is a conservative approach. To further improve the measurements on the gearbox torque, using motor speed variation and geometry of the unit, the rotary and articulating inertias of the pumping unit components can be calculated, as mentioned above. This generalizes the API method and generates a more precise and realistic account of net gearbox torque[2]. The advantage to using this method becomes apparent when using a heavily loaded Nema D or UHS motors or when a VSD is in use.

A third way that can be used to eyeball whether or not the gearbox is overloaded and/or out of balance is by using permissible load diagrams. The permissible loads are two separate curves that can be overlaid with the corresponding measured surface dynagraph. These diagrams show the allowable loads at the polished that fully loads the gearbox throughout the upstroke and downstroke. This is a very useful way to get a qualitative feel for gearbox loading, unit balance, and pumping unit geometry performance. These diagrams are illustrated in Figures 5 and 6 and are discussed in an example below.

Generally speaking, unit balance should be within 20% of ideal balance if the gearbox is not overloaded. However, when heavily loaded more precise balancing is required to avoid overload. If the gearbox is overloaded, when possible, the correct counterbalance moment should be calculated in order to balance the unit. Ideally, the crank arms are able to hold enough counterweight for the unit to achieve the required moment. Secondly, it should be noted whether or not this balanced condition results in an overloaded situation or if it resolves the overloaded condition as well as the balance issue. If in fact crank arms cannot bear enough counterweight or there is not enough counterweight available to balance the unit, consider shortening the stroke. It is an easy proof to show that the counterbalance effect is increased as the stroke is shortened. This option should only be implemented when economically feasible. In other words, either the well was over displaced and thus a shorter stroke would still make the same production or the unit stroke should be shortened along with an appropriate increase in strokes per minute to maintain current production. There are times when shortening the strokes by a few crank holes and increasing the speed can actually lower gearbox loading and at least maintain production at the same time. These possibilities should be investigated during the analysis.

If the gearbox remains overloaded when balanced correctly and the pump is filling completely, there are other more expensive solutions to the problem instead of shortening the stroke and adjusting pumping speed. Decreasing pump capacity is an alternative to lowering gearbox loading, since the load that the pump is moving is directly proportional to the surface load, which is an influential variable in calculating rod torque. Another possibility is moving pumping units around and placing a larger unit at the well to keep the production rate the same. Finally, the operator may decide to run the unit in the overloaded and balanced condition, knowing that the gearbox life will be shortened. However, this last option is not always a bad idea if the economics of more production is weighed against the cost of failure. Be sure to note that the balancing requirements of the pumping unit will change if the pump is lowered or the speed is changed.

Once the surface dynagraph has been recorded and the rod string description is obtained, the loading on the rods can be calculated by the following methods. For API grade K, C, and D rods, the recommended practice is to calculate the stress loading as a ratio of the existing stress range to the allowable stress range using the API Modified Goodman Diagram. For ultra high strength rods and fiberglass rods, the manufacturers provide their recommended stress range diagrams to follow. It is highly recommended that fiberglass rods do not enter a compressive state because in doing so a catastrophic failure may occur.

When designing a recommending string for use in the well, shoot for equal loading at the top of each rod interval. However, if there are fiberglass rods in the design, some say that fiberglass loading should not exceed say 70% of loading in a design program. In addition, the loading of sinker bars or a heavy rod interval at the bottom is typically much lower than in the main rod string.

In addition to equal taper loading, the proper service factor should be used in calculating rod loading. The correct factor is normally determined by the operator's field experience. Lower service factors are typically used as the well environment gets more corrosive.

PUMP DYNAGRAPH

When examining the pump behavior, the most widely used method is creating a pump dynagraph, which is a parametric plot of the time histories of the calculated loads and positions. It is commonly referred to as the pump card, and of course, the surface dynagraph is referred to as the surface card. The pump card was first measured by W.E. Gilbert[3] in 1936. He also introduced the utility of the pump card for diagnosing the conditions at the pump. The mathematical solution to obtaining the downhole card was first obtained by S.G. Gibbs when he modeled the rod string behavior with the damped wave equation in both the predictive and diagnostic cases. A paper on the former[4] was published in 1963. The latter[5] was patented in 1967.

From the pump card, a complete analysis of the conditions of the pump can be determined. For example, any severe downhole drag friction can manifest itself on the pump card if it is not removed by the damping term in the wave equation. This effect is not always negative for the simple reason that it is desirable to know whether or not there exists downhole drag friction in the well. Even though the pump card can become distorted, it still gives useful information about the stuffing box friction, rod/tubing drag, and pump friction. For example, stuffing box friction distorts the downhole card, but the friction is also observable on the surface card at the top and bottom of the stroke by load changes without a position change. The load change on the top and bottom stroke should be averaged and then half of this average should be used as the new value for the stuffing box friction. In doing this, a more well-defined pump card can be obtained thereby allowing other characteristics of the pump to emerge that may not have been observable from the distortion created by the stuffing box friction. This method will be outlined in detail by the example below.

The friction from rod/tubing drag will also distort the shape of the pump card as well as the perceived pump stroke and load at the pump. The lower the rod/tubing drag occurs, the better defined the pump card becomes, but it will show what can be perceived as a higher than expected fluid load. Some causes of rod/tubing drag are a crooked hole, buckling tubing due to set down weight on a tubing anchor or packer, or paraffin. If a crooked hole is the culprit, use a fluid level shot to determine if the pump can be moved above the problematic dogleg or kick-off point to eliminate the friction. If the friction appears seemingly out of nowhere, it is possible that the tubing anchor is set in compression. If there is a history of paraffin, consider hot oiling or hot watering the well if there is a higher than normal amount of friction shown at the pump. Small amounts of paraffin deposits cannot be detected with the dynamometer yet will cause severe problems when pulling the pump through the deposits.

Arguably one of the most common and most useful observations gleaned just from glancing at the pump card is a qualitative indication of whether or not the pump is in good mechanical condition. The slippage between the plunger-barrel fit and/or past the traveling valve is represented by a pump card resembling an upside down bowl. Likewise, a leaking standing valve is resembled by a right side up bowl. The slower the pumping unit speed, the more evident the leakage will become on the pump card. The exact rate of leakage in barrels per day is not a value that can be accurately determined from the pump card, instead leakage should be computed by using one of the methods listed below in the valve check routine.

What makes the diagnosis of pump conditions so challenging and enjoyable is the fact that the shapes of the pump dynagraphs that have been identified as representing certain pump conditions can be combined to produce a hybrid card which is a combination of all of the things that are occurring at the pump. This is due to the fact that the viscous damped wave equation is a linear hyperbolic partial differential equation which means that the principal of superposition is applicable, thus allowing well weighers the ability to combine solutions. Thus, a well which exhibits more than one phenomenon, when analyzed using a pump dynagraph, will actually display the conditions as if they were merged together.

For example, in Figures 3 and 4 is a C228-250-74 in the second hole (65.5”) with a 1¼” pump running clockwise at 6.5 SPM. The seating nipple is at 8801’, the TAC is set at 8486’, and the pump intake pressure from a fluid shot was measured at about 330 psi. It is obvious that the pump card is distorted. Since the card is distorted and the fluid load is normal, one can conclude that the friction is not at or near the pump. Upon further inspection of the pump card, there is an indication of possible slippage past the plunger and/or a leaking traveling valve. We cannot yet confirm this completely because there is still distortion in the pump card. Looking at the surface card in Figure 3, it can be seen at both the bottom of the stroke and the top of the stroke that there is in fact a considerable load change without a position change. Scaling the ends of the card and averaging the values, we can infer that there is about 1000 lbs of stuffing box friction. We divide this number in half, obtaining 500 lbs and subtract this amount from all the loads on the upstroke and add this number to all the loads on the downstroke. This does not change the way the surface card is plotted, but instead it gives a “new” surface card (plotted within the original surface card in Figure 4) to the diagnostic program which has the stuffing box friction removed. With this cleaner surface card, the pump card in Figure 4 is computed. The distortion is gone and the slippage past the plunger and/or leaking traveling valve is confirmed. This leakage is only a qualitative observation. Valve checks on this well did conclude that there is approximately 11 BPD slippage (leakage).

If one was actually in the field, other indicators of excessive stuffing box friction are squealing of the polished rod as it moves through the stuffing box as well as being hot to the touch. The stuffing box is sometimes tightened to this degree to prevent leakage when there is a tubing head back pressure valve installed to increase back pressure. Some feel that this helps the well “pump.” It is not the opinion of these authors that an increase in back pressure helps the well pump, since the tubing head pressure does not have an effect on how much free gas and/or liquids enter the barrel from the reservoir. However, back pressure does help to compress free gas and keep gas in solution inside the tubing, thus reducing the amount of free gas available to cause gas “heading” and poor stuffing box lubrication. As with most techniques for solving problems in the oilfield, in moderation, increasing back pressure can be a useful fix. However, too much back pressure can cause a need to over tighten the stuffing box, thus creating a possibility for wearing out the packing and causing a leak at the well head. Also, back pressure has a tendency to increase equipment loads and power consumption.

Another common problem that can be examined by inspection of the surface and downhole cards along with observing conditions when in the field is when the pump is hitting or tagging on the bottom of the downstroke or the top of the upstroke. If the tag is hard enough, there is a pulse that is discernable by grasping the polished rod during the stroke. The tag will be evident at either the top or the bottom of the pump stroke, depending on where the pump is hitting. At the surface the tag may also occur in the end of either the upstroke or downstroke, respectively. Say the pump is hitting on the downstroke. The tag can sometimes occur a short distance after the beginning of the upstroke on the surface card. See Figure 6. This makes perfect sense since there is a slight delay for the tag to reach the surface which, is caused by the travel time for the wave to travel from the pump to the surface. This phenomenon is more pronounced with fiberglass rod designs because the dynamic waves travel slower in the less dense fiberglass rods.

When studying the pump card, take note of any gas interference. Gas interference occurs in high pump intake pressure situations when the pump is filling with a considerable amount of free gas that is not separated before entering the pump. If the volumetric efficiency can be improved or the total pump capacity can be increased, then the well may produce more fluid. There are three main ways to accomplish this. First, if there is adequate sump, lower the pump as far as possible, preferably below all perforations. Add a 1¼” dip tube to aid in separation by increasing the annular area above the pump intake, thereby decreasing the rate at which the fluid level falls downward to the pump suction. This separation method is sometimes called a modified natural separator. This method yields the most favorable results. However, as with most solutions in the oilfield, there are instances when this method is not feasible, such as the well produces frac sand or other solids that foul or destroy the pump or horizontal completions. The second method is to install a packer type separator with a TAC below to set the packer. This creates an artificial sump and uses the annular area of the casing and tubing as its separation zone. Third and most common is the poor boy separator. It is installed just below the seating nipple. This usually consists of one joint of tubing (mud anchor), either the same size as the tubing string, or sometimes a size larger with slots in the upper portion of the joint to allow fluids to enter and free gas to escape. Connected to the pump intake is a dip tube (gas anchor) which extends well past the slots in the joint down towards the bottom of the mud anchor. The dip tube should have the smallest practical diameter, while the mud anchor should have the largest practical diameter for the casing size. Large mud anchors may cause the operator to cringe because of possible hang-ups and fishing problems. For example, some operators will run a 3½” mud anchor in 4½” casing, while others will not. Also, it is not uncommon for a well to have a poor boy gas separator installed where the intake of the dip tube is barely below the slots on the mud anchor joint. This renders the gas separator completely ineffective. Keep an eye out for things like this which would never have a chance of working. In addition, care should be taken when installing the gas separation equipment because broken downhole equipment will render the separator ineffective. For instance, a fiberglass mud anchor may shatter if it is lower too quickly into the liquid downhole. Another potential problem that can occur when lowering the diptube is that it can get caught up in the upset of the seating nipple and break off. Both of these examples would cause the gas separator to be useless.

There could be an emulsion problem where the gas simply does not escape from the liquid in time to be separated before entering the pump. At best, gas bubbles in liquid rise at a rate of 0.4 ft/sec. One other observation that can be made that may have some bearing on whether or not the separator will have a chance is to take a bleeder sample at the surface to see how fast the oil, water, and gas separate at surface conditions. To the authors’ knowledge, there has not been any correlation to the speed of gas liberation at the surface to the effectiveness of a gas separator down hole. However, this may serve as part of the well evaluation for considering a gas separation scheme in hopes of increasing oil production and increasing the pump’s volumetric efficiency.

The gas separation schemes are used to prevent free gas from entering the pump. Pumps experiencing gas interference will also exhibit a gas expansion problem if the gas compression ratio is low, which is normally caused by spacing the pump too high. This is shown by a slower than usual opening of the standing valve at the beginning of the upstroke, which creates a rounding effect on the upper left corner of the pump card. See Figure 5. It should be noted that gas expansion cannot occur without gas compression. A remedy for poor spacing is to lower the pump until it tags then raise the pump slightly to avoid tagging. The pump card should then show no significant gas expansion and more fluid (liquid and gas) should enter the pump during the stroke.

In Figures 5 and 6 is a C228-246-86 in the long hole with a 1½" pump running clockwise at 9.9 SPM. The seating nipple is at 8308' which is above the perforations, the TAC is set at 8312', and the pump intake pressure from a fluid shot was measured at about 694 psi. From first glance of Figure 5, we see that this well is experiencing both gas interference and gas expansion problems. The well has reached a stable state, as it has been pumped this way for over a year's time. The run time is 24 hours and the fluid level is never pumped down. From the fluid shot, we have a liquid/gas gradient of 0.085 psi/ft with 6896' of pump submergence, a surface casing pressure of 100 psi, and a gas column pressure from surface to fluid interface of 108 psi. Using Gilbert's modified S-curve, this corresponds to a gas free liquid level of 1636' above the pump. This indicates that there is a large amount of gas in the tubing casing annulus and an improved gas separation scheme might be beneficial. A clue that the gas may separate in a short time if the fluid velocity could be reduced is observed in Figure 6. The well was shut down for a few minutes during which time the gas separated. When started up, it can be noted that the stroke length has increased, there is no gas compression or gas expansion, and in fact there is a tag at the bottom of the stroke. This was observed in the field by the unit rattling from the shock waves sent up the rod string. It can be seen in the surface card during the beginning of the upstroke. There is a very obvious loss of load (tag) during the first part of the upstroke. This tag is also observable by a tail on the bottom left of the pump card at the end of the downstroke.

If there is no possibility of improving the gas separation method, then consideration should be given to increasing the pump capacity. Wells with gas interference do not have large fluid loads due to the light gradient in the tubing and high pump intake pressure. A 1¾" pump in this well would indeed pump more gas, but hopefully more oil would come with it. Further inspection of gearbox torque on this unit yielded the conclusion that larger surface equipment would be needed in order to increase pump size.

To conclude, we will discuss one of the most common and unfortunately the most detrimental pump cards- the fluid pound. A fluid pound can be damaging to the entire pumping unit system. Also, pumping system efficiency drops as pump fillage drops. This means that the harder the pump pounds fluid, the more the power bill increases. When the pump hits the fluid on the downstroke, there is a tremendously high tendency to buckle the rods and slap the tubing causing shiny spots which are most susceptible to corrosion-erosion, rattle the surface unit, and put unnecessary stress on the gearbox and structural bearings. The worst case is where the pump velocity is at its fastest and hits the fluid. This is usually around mid-downstroke. A slight fluid pound is usually what is sought after in the field. This condition is sometimes thought of as ideal since the operator is obtaining the maximum amount of production from the well and the pound is occurring early enough into the downstroke that the pump velocity has not increased to a speed where severe damage would result. However, allowing a hard fluid pound to persist is not advisable. Some remedies include slowing the unit down (normally best), shortening the stroke, installing a smaller pump, or installing a time clock or pump off controller. Use of various predictive programs can give the operator a variety of solutions. The most economical and effective solution can then be determined by modeling the different possibilities.

MECHANICAL CONDITION OF THE PUMP

The mechanical condition of the pump can best be investigated by performing valve checks. From the valve check routine, the amount of traveling valve leakage and/or slippage past the plunger or the amount of leakage past the standing valve can be quantified with good accuracy. There are differences of opinion concerning whether or not leakage/slippage past both the traveling valve and standing valve is observable. To this point, there are no known real data which support observing both valves leaking during a valve check routine. However, it is highly likely that both valves leak at least some. It is usually assumed that only the valve which leaks the worst will be detected.

When performing the valve check, one must bring the unit to a gentle stop to minimize dynamic effects. This is most easily done close to the ends of the stroke, where due to the design of pumping units, the polished rod velocity is already slowing down. Old school thought has been to stop the unit at various points during the upstroke. In fact, the authors agree that this technique can be beneficial in indicating whether or not there is a certain point in the barrel where leakage is particularly bad. Bringing the pumping unit to a halt in the middle of the upstroke requires even greater care in stopping gently. In this case, the operator is working against the unit geometry because at points other than the ends of the stroke the polished rod is moving at a considerably higher rate. Abrupt or sudden stops will result in valve checks that are invalid.

When the unit is stopped correctly on the upstroke, the leakage past the traveling valve and/or the slippage past the plunger is checked by measuring the load loss (if any) at the polished rod versus time. The idea that the fluid load loss is a characteristic of traveling valve leakage and/or plunger slippage is clear. The same method is used to check leakage past the standing valve, except on this check the operator is looking for a load increase. This can be explained by the following. When beginning a standing valve check, the unit is stopped at the bottom of the stroke. The standing valve rests tightly on seat, while the traveling valve rests lightly on seat. The pressures above and below the traveling valve are equal. If the standing valve is leaking more than the traveling valve and plunger, as fluid moves past the standing valve faster than the traveling valve/plunger, the pressure below the traveling valve begins to decrease causing a pressure difference across the plunger. This in turn causes the traveling valve to seat firmly and the rods begin to take on the load of the fluid in the tubing. Any change in load versus time does signify a leak, but there is further mathematics required to quantify the amount. The other elements involved are the stiffness of the rod string, the diameter of the pump, whether or not there is a tubing anchor, and the initial fluid load.

To quantify pump leakage in BPD three methods can be used[6]. They are pump leakage from initial load loss rate, pump leakage from polished rod velocity, and pump leakage from downhole pump card and pump velocity. Pump leakage from initial load loss rate is the most widely used method. The initial load loss rate method is illustrated in Figures 7 and 8. The first point, point 1 in Figure 8, is selected at or just after the instant the unit is stopped. The second and third points, points 2 and 3 in Figure 8, are chosen along the load loss trace. A curve is fitted to these points and the maximum rate of change (slope) is calculated at the first point. This maximum rate of change could also be determined by drawing a tangent line to the load loss line at the first point.

Other useful values that are obtained from a valve check routine are the counterbalance effect, in pounds, at a certain polished rod position. In Figure 7, the vertical line crosses the position curve where the measured polished rod position was during the stroke when the counterbalance equilibrium was found and indicates whether the unit was on the upstroke or the downstroke. In Figure 8, the vertical line crosses the load curve where the measured polished load was during the stroke when the counterbalance equilibrium was found. Using the torque factors from the pumping unit geometry and the structural unbalance, the actual moment of the crank arms and counterweights can be calculated. This can then be compared to counterbalance tables in the literature or to various computer programs which can calculate counterbalance moment by knowing the crank arm, counterweights and their respective positions on the crank.

The residual friction is another useful value that is measureable from the valve check routine. Note that the traveling valve load, point A in Figure 8, is made up of the buoyant rod weight, the fluid load, and the drag friction acting downward during the upstroke. This is heavier than just the buoyant rod weight plus the fluid load. The standing valve load, point B in Figure 8, is made up of the buoyant rod weight minus the drag friction acting upward during the downstroke. This is lighter than just the buoyant rod weight. When performing this routine, multiple travelling valve and standing valve checks are recommended and using the highest travelling valve load and the lowest standing valve load will yield the most accurate results when using these values to perform the following calculations. The leaked off traveling valve load, point C in Figure 8, is the traveling valve load after the entire fluid load has leaked off. Taking the difference in the leaked off traveling valve and the standing valve, we obtain the total drag friction in the system, which is equal to the combined total of the upstroke and downstroke drag friction. A general rule of thumb is that for every 1000’ of well depth, one can expect to have 100 lbs. of drag friction in the system. Using the measured travelling valve load, standing valve load, and residual friction, one can also calculate what the fluid load should be by taking the travelling valve load and subtracting both the standing valve load and the residual friction. This offers a cross check to the fluid load that is obtained from the pump card.

CONCLUSION

It is the hope of the authors that this paper has provided a general scope of what can be accomplished by going to the well site and weighing the well. There are many diagnostics that can only be performed by getting on the pumping unit. Regardless of the fact that we are finding ourselves in a communication revolution with SCADA hosts and an automation revolution with POCs and VSDs, to fully understand a well, we have to lay our hands on it, see the unit working, and listen to the system to really know what’s going on. Any automated method will never replace a trained operator at the pumping unit. If one is wondering what action to take on a well, a way to feel confident about determining what should be done is to have a well weigher get out to the site, put his equipment on the pumping unit, and make a diagnosis. There is no substitute for hands on and getting dirty.

REFERENCES

1. K.B. Nolen and S.G. Gibbs, Operating instructions for the DIAG$$ system of programs, Nabla Corporation (1994), 177—182.
2. S.G. Gibbs, Computing gearbox torque and motor loading for beam pumping units with consideration of inertia effects, SPE 5149, *J. Pet. Tech.* (September 1975), 1153—1159.
3. W.E. Gilbert, An oil-well pump dynagraph, *Drill. and Prod. Prac., API* (1936), 94—115.
4. S.G. Gibbs, Predicting the behavior of sucker-rod pumping systems, *J. Pet. Tech.* (July 1963), 769—778; *Trans. AIME* **228**.
5. S.G. Gibbs, A method of determining sucker rod pump performance, U.S. Patent 3343409 (September 26, 1967).
6. K.B. Nolen and S.G. Gibbs, Quantitative determination of rod-pump leakage with dynamometer techniques, SPE 18185, *SPE Production Engineering* (1990), 225—230.

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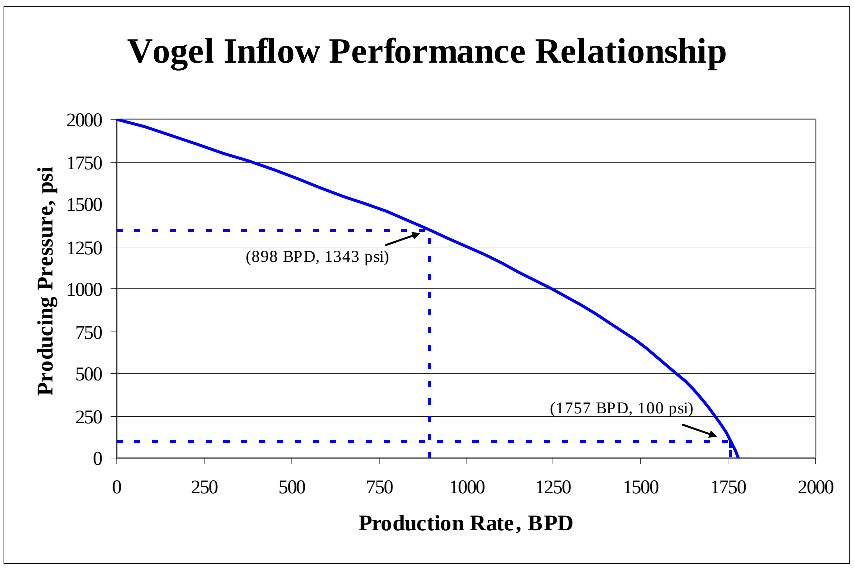
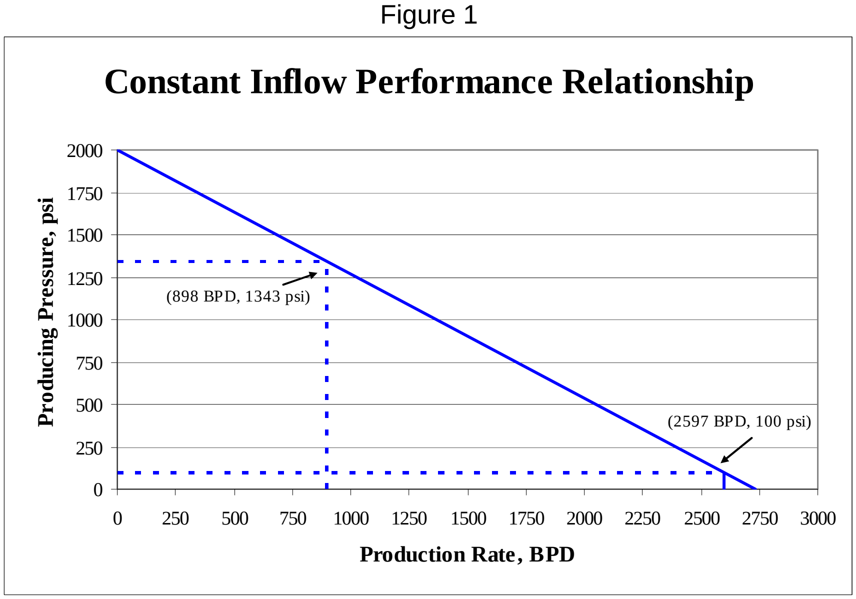


Figure 1



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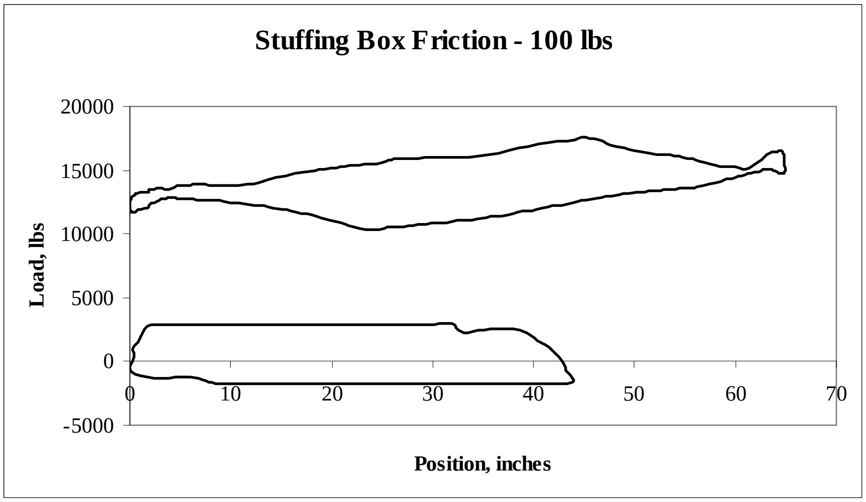

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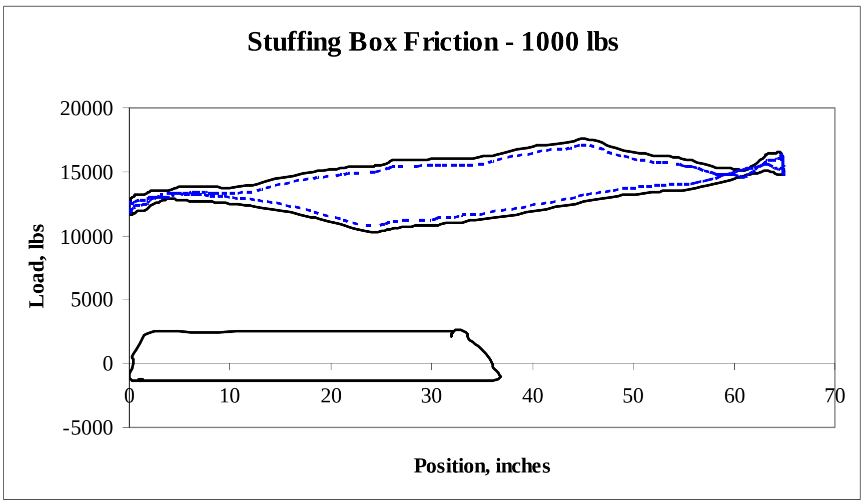

Figure 4

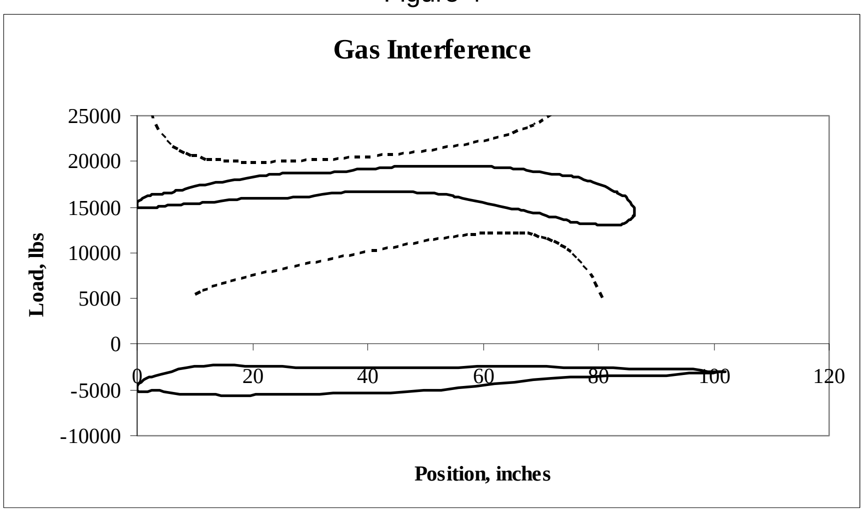

Figure 5

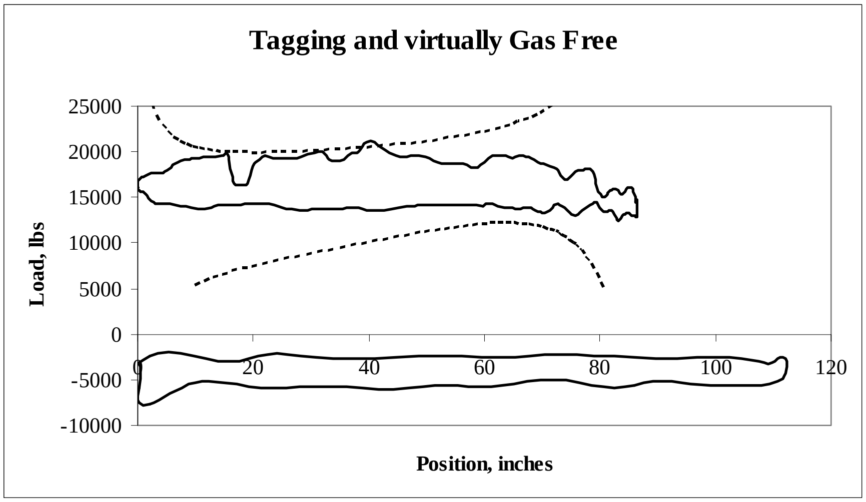

Figure 6

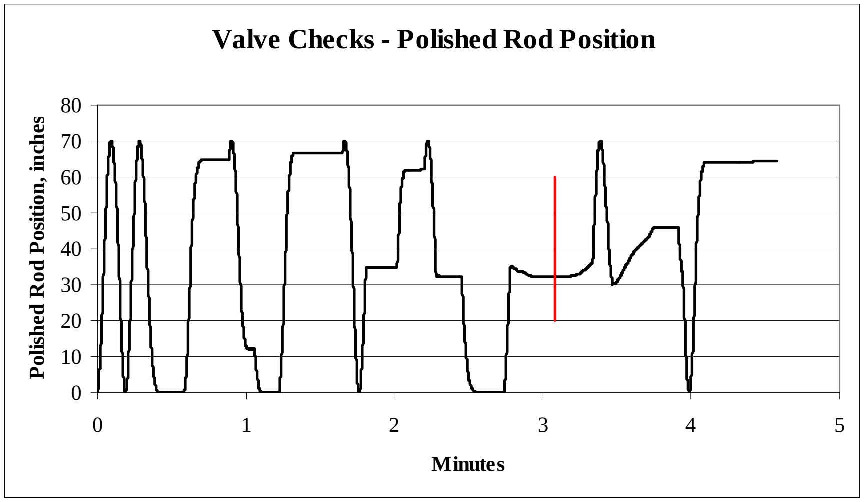

Figure 7

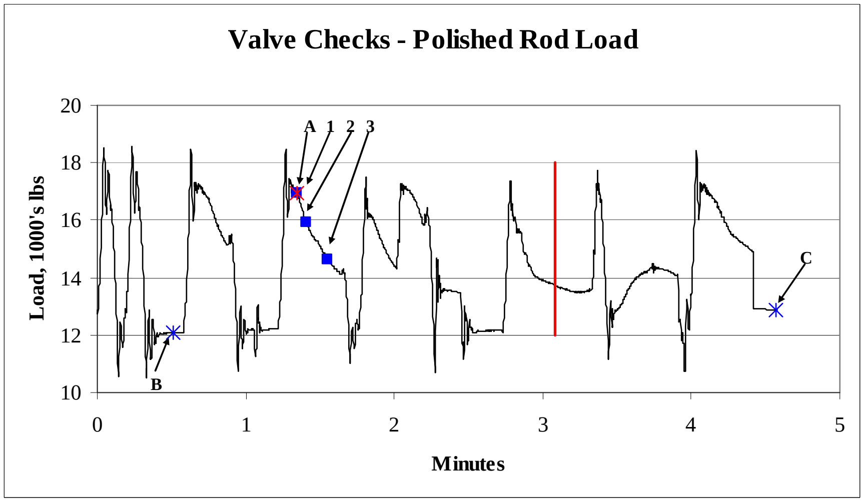

Figure 8