PROGRESS REPORT #4 ON "FLUID SLIPPAGE IN DOWN-HOLE ROD-DRAWN OIL WELL PUMPS"

John Patterson, ConocoPhillips Company
Kyle Chambliss, Oxy Permian
Lynn Rowlan, Echometer
Jim Curfew, Oxy Permian

ABSTRACT AND SCOPE

This paper will present the results of the last fluid slippage in down-hole-rod-drawn pump fluid slippage testing recently completed at the Texas Tech University test well, Red Raider #1. This is # 4 in the series which began with laboratory testing presented in 1998 [1] and field testing presented in 1999 [2] and 2000 [3]. A portion of the research was also presented at the SWPSC in 2006 [4]. This paper will present an update to the empirical equation which will estimate down-hole fluid slippage over a range of pump sizes, clearances and pump speeds (SPM).

The objective of this latest phase of the test project was to develop a slippage equation that included both pump clearance and plunger velocity terms. It should be noted that no previous slippage equations have included a plunger velocity term. By including a velocity term in the form of pumping speed in Strokes per Minute (SPM) the rod pump system designer is given a tool to more accurately predict the fluid volumes from a beam pumped system. The new slippage equation will allow the designer to build a hydraulically efficient beam pumping system that is less prone to failures caused by compression on the down stroke due to pump drag. Historically the designer has been faced with increasing pump clearances to reduce pump drag, but he was relatively uncertain of the hydraulic efficiency with the looser fit pumps. A more accurate equation allows the designer to obtain a better balance between a looser pump fit to reduce pump drag and a tighter fit to improve efficiency. In addition, this equation has been qualified against different pump diameters improving the confidence of the equation in a wider range of operating conditions.

BACKGROUND & DISCUSSION

As stated in Progress Report #3 [3], "Most fluid slippage equations have overstated the slippage of down-hole, rod-drawn positive displacement pumps with metal plungers. The historical equations dating back to the 1940's predict about twice the observed slippage for clearances equal to or less than .006" (six thousandths of an inch) depending on the historical equation. For clearances larger than .006 inch these historical equations can overestimate the slippage by a factor greater than three". For clarification, it should be noted that this statement is referring to historical industry equations dating back into the 1940's and not to the Arco-HF 2000 equation.

SUMMARY OF RESULTS

Based on recently completed testing at the TTU Test Well, the following empirical equation has been developed which incorporates the previous field testing work done on one pump size (1.75") with clearances from 0.00387" up to 0.0205" along with the recent results which were done on 2 pump sizes (1.5" and 2.0") using pumping speeds from less than 1 SPM to 16.6 SPM. As noted above, this is the first time that any slippage equations have included a plunger velocity term. The results from the recent TTU tests were comparable to previous field tests; therefore, the following empirical equation is presented:

$$Slippage = \left[(0.14 \cdot SPM) + 1\right] 453 \frac{DPC^{1.52}}{L\mu} \qquad \text{(Equation 1)}$$

INTRODUCTION TO PUMP SLIPPAGE

Pump slippage is the liquid that slips between the plunger outside diameter and the pump barrel inside diameter into the pump chamber between the standing valve and traveling valve when the traveling ball is on seat. Slippage into the pump barrel on the upstroke results is reduced pump displacement. Many pump slippage formulas have been developed over the years. Most of these formulas over predict the pump slippage and a more accurate formula is needed. Slippage calculations are important to determine the amount of fluid required to lubricate a plunger on a plunger pump. A slippage formula is also required to calculate the pump efficiency and plunger pump production in order to maximize production.

PLUNGER PUMP OPERATION

A plunger pump consists of a barrel, plunger, traveling valve and standing valve. The outside diameter of the plunger is less than the inside diameter of the barrel. This difference in diameter is called pump clearance and is usually expressed in thousands of an inch. The traveling valve is connected to the plunger to form an assembly, and the plunger assembly is attached to the sucker rods. When the traveling valve is closed the ball is on the seat and the closed valve acts as a check holding the fluid in the tubing. When the traveling valve is open the fluid in the pump barrel can be displaced into the tubing. The standing valve when opens allows fluid to flow into the pump barrel below the plunger. When the standing valve is closed, it acts as a check valve holding fluid in the tubing. Figure 1 is a pump card representing the load the pump applies to the rod string. On the pump card the standing valve is closed from C-D, D-A, and A-B; and the standing valve is only open from B-C. On the pump card the traveling valve is closed from A-B, B-C, and C-D and the traveling valve is only open from D-A. Pump slippage can only occur when the traveling ball is on seat.

Before the beginning of the upstroke the pressure from the tubing fluid is applied to the closed standing valve and the traveling valve is open as fluid is displaced from inside the pump into the tubing (D-A). At the start of the upstroke, A, the traveling valve and standing valve are both closed and the pressures above and below the plunger are equal. During the upstroke (A-B-C-D) the fluid load applied to the rod string is due to differential pressure acting on the plunger and is equal to the pump discharge pressure minus the pump intake pressure times the area of the pump plunger. The fluid load is gradually transferred from the tubing (A-B) as the rods stretch to pick up the fluid load. The standing valve begins to open at B when the pressure in the pump drops below the pump intake pressure, allowing fluid to enter the pump. From point B to C, the rods carry the fluid load while well fluids are drawn into the pump. At C, the standing valve closes, and the traveling valve remains closed until the pressure inside the pump is slightly greater than the pump discharge pressure. From C to D, gas in the pump (if present) is compressed as the plunger moves down to increase pressure on the fluid from the intake pressure to the static pressure in the tubing. As the fluid in the pump barrel is compressed, then the fluid load is gradually transferred from the rods to the tubing. At D, the pump barrel pressure equals the static tubing pressure, and the traveling valve opens. From D to A, the fluid in the pump is displaced through the traveling valve into the tubing as the closed standing valve holds the fluid load on the tubing.

IMPORTANCE OF PROPER PUMP SLIPPAGE

Proper pump slippage is a balance between pump lubrication to extend the life of the pump and pump volumetric efficiency. Pump volumetric efficiency is defined as the actual fluid displaced per stroke divided by the fluid displacement of the pump. On the upstroke when the traveling ball in on the seat, the pressure difference acting across the plunger forces fluid to leak from the tubing (high pressure) through the clearances between the barrel and plunger back into the pump (low pressure). Fluid leakage between the pump barrel and pump plunger is called pump slippage. Slippage is one of the factors that affect pump lubrication and efficiency. Proper pump lubrication is necessary to extend the life of the pump.

Another factor that affects the efficiency is a sticking plunger, which shortens the actual pump stroke. A plunger can stick if the clearance between the pump barrel and pump plunger is too small. The clearance must be large enough to allow an appropriate amount of slippage to sufficiently lubricate the plunger and barrel. Sand and other particles need to pass between the barrel and plunger, which could otherwise jam the pump plunger in the pump barrel. The typical amount of slippage for lubrication is 2 to 5% of the total production.

Plunger sticking or galling of metal can shorten the life of the rods and the pump. Galling of the metal is when the pump plunger and pump barrel rub together with no lubrication causing the metal to wear and become rough. In order to predict pump performance and design for proper lubrication, pump slippage needs to be accurately modeled. Pump slippage can be mathematically modeled and experimentally measurements can be done to aid in the development of the model.

OVERVIEW OF SLIPPAGE FIELD TEST

Texas Tech University, along with about a dozen companies, both operators and service companies, developed and funded a slippage field test which was performed at the Texas Tech test well facility, Red Raider # 1.

HOW SLIPPAGE IS CALCULATED

Slippage is the difference between what the pump should produce based on the effective plunger travel (D to A on Figure # 1) assuming there is no leakage through the traveling valve and what the pump actually produces as measured at the surface. Wave equation programs are used to estimate the effective plunger travel and pump fillage. The effective plunger travel, from the downhole pump card calculated using the wave equation, gives an estimate of what the well should produce if there were no slippage. Production is measured at the surface while surface dynamometer data is acquired to calculate the load and position at the pump.

SLIPPAGE THEORY

It has long been believed that slippage is unaffected by pumping rate, so all of the historical formulas do not consider pumping rate as a variable. [5] The slippage consists of two elements, dynamic slippage and static slippage. [6] The static slippage is slippage due to a pressure difference across the plunger. The movement of the plunger causes dynamic slippage and the slippage theoretically increases as a function of increasing plunger velocity. Dynamic slippage occurs on both the up stroke and down stroke while the static slippage only occurs in the up stroke. On the up stroke the dynamic slippage contributes to the total slippage, while on the down stroke dynamic slippage just lowers the amount of fluid that has to pass through the traveling valve. Of the two elements that make up slippage, static and dynamic, static slippage generally is the larger of the two elements. Another thing to consider is as pumping rate increases, dynamic effects in the tubing change due to friction increase. At higher speeds, a slight increase in pressure across the plunger is expected, which would cause a slight increase in slippage with speed.

If a pump does not have 100 % fillage, there are a few things to consider. Let's say the pump is 50 % full when the plunger reaches the top of the stroke. When the plunger starts to move downward the traveling valve will not open until the pressure below the traveling valve is greater than the pressure above it. Until the traveling valve opens slippage is still occurring around the plunger. This slippage adds to the fillage of the barrel. If the pump was 50 % full at the top of the stroke, it may be 55 % full, relative to the top of the stroke, once the traveling valve opens due to fluid slipping around the plunger and filling the barrel. On this same pump that was 50 % full at the top of the stroke; some of that fillage was due to slippage on the up stroke. If there was no slippage on the up stroke, the fillage would be smaller. If there was more slippage on the upstroke, the fillage would be higher. The liquid production rate measured at the surface would remain the same, as the pump fillage downhole increased due to slippage. Basically, if the pump does not have 100 % fillage, slippage helps fill the pump, and the pump efficiency would be reduced as slippage from the tubing back into the barrel fill the pump but does not increase production at the surface.

Each time the traveling valve closes there will be a small amount of leakage through the valve as it closes. During field tests to measure slippage, this leakage is not considered because it is believed to be small. It also occurs in all rod pumped wells. Modeling the slippage with this leakage included would more accurately model the real life slippage in a well. The slippage through the traveling valve does not contribute to the slippage that lubricates the plunger and barrel.

EFFECTIVE PLUNGER STROKE AND THE WAVE EQUATION

Effective plunger travel is not equal to the surface stroke of the pumping unit. The rod string has an elastic nature, which affects the plunger travel. There are five factors that affect the plunger travel. [7]

1. Rod and tubing stretch due to fluid loads changing from traveling valve to standing valve during the pumping cycle
2. Plunger overtravel caused by the dynamic nature of the rod string
3. Rod Vibration
4. Subsurface friction effects
5. Gas in the pump barrel

To accurately calculate plunger travel in the slippage test, a few things were done to minimize or remove as many of the above factors as possible. All new equipment was used and the well bore was almost perfectly straight, this reduced vibration and minimized subsurface friction. The inlet pressure to the pump was kept high enough that no air would enter the pump; this removed any gas interference in the pump and reduced pump vibration.

To calculate the effective plunger stroke, a wave equation program is used. These programs use the wave equation along with boundary conditions that describe various aspects of the pumping system. [8] Different companies have different programs, based on the same basic principals but with differences in the way the equations are solved or the data collected. Two different programs which will be discussed later were used during the slippage tests.

RED RAIDER # 1

The Red Raider # 1 is located on the East side of Lubbock, Texas, about 10 minutes from Texas Tech University. The well is 4006 ft deep with 9 $^{5}/_{8}$ in casing and a Cameron dual completion tree. The well is equipped with a Lufkin C456D-305-144 pumping unit and an ABB variable speed drive.

BASIC EXPERIMENTAL SETUP

The basic experimental setup is shown in Figure 2. This basic setup was the same for all tests. The fluid used in the test was fresh water with a few parts per million of corrosion inhibitor. The rod pump pumped fluid through the system. The fluid flow was measured by a mass flow meter, which will be discussed further in the next section. Before the fluid entered the mass flow meter, it passed through a filter to remove any particles that may have interfered with the mass flow meter. Mass flow meters will only accurately measure a single fluid, in our case water. To keep gas out of the system, a backpressure valve was down stream from the mass flow meter. A backpressure of 12 to 15 psi was kept on the flow line to insure the line was full of water. As long as gas was not pumped through the plunger pump, no gas could enter the mass flow meter. A surge tank between the filter and the mass flow meter was also used. This increased the accuracy of the flow meter to 0.2 %.

After the water flowed through the backpressure valve, it entered the wellhead and fell down the casing annulus back to the bottom of the well completing the loop. The inlet pressure of the pump was due to the hydrostatic column of water in the casing annulus. When the pumping unit was shut down, the water collected in the casing. When the unit was started, the fluid level in the casing had to come to a steady state so that the inlet pressure of the pump was constant throughout a test; this typically took about 25 minutes. This inlet pressure could be adjusted if necessary by opening the valve to the reserve tank and either pumping water to the reserve tank to lower the level, or increasing the level by turning the pumping unit off, lowering the casing pressure to zero and letting fluid flow out of the reserve tank and into the well bore using gravity.

The pump inlet pressure, pump outlet pressure and pump inlet temperature are measured with a Wood Group down hole instrument package. Tubing pressure and casing pressure were measured using various transducers. There were three different ports to screw in pressure transducers or gages on both the tubing and casing. This allowed for multiple transducers to be installed so that all of the RTUs did not have to rely on a single transducer.

DATA ACQUISITION AND PUMPING UNIT CONTROL

Data could be acquired from four different units during slippage tests. The four units were the Micro Motion mass flow meter, the Wood Group RTU, the ION RTU and the Lufkin SAM controller.

The pumping unit could be controlled by an ABB variable speed drive or the variable speed drive could be by-passed and an auto-off-hand switch on the main power panel could control the unit.

Mass Flow Meter

The mass flow meter that was installed on the loop was a Micro Motion F-100 Coriolis Effect mass meter. The Micro Motion meter did not have any data logging capability, so to log the data Prolink software installed on a laptop was used. Before the pumping unit was turned on, the laptop with the Prolink software was connected to the Micro Motion meter. Data logging was then started and the pumping unit turned on. That way the data being logged could be monitored to make sure the flow through the meter reached steady state. The Prolink software recorded date, time, elapsed time, density, mass flow rate, total mass flow rate, temperature, volume flow rate, total volume and drive gain. The software sampled the meter about twice a second. Drive gain recorded from the meter was basically how hard the meter was working; adjusting for substances such as any particles or gas that flowed through the meter. Generally the drive gain was less than 3 %. Anytime this gain went higher, it usually meant that the pump was pumped off and pumping some gas through the meter. When this happened, more water was added to the well bore to increase the bottom hole pressure and the drive gain went back to around 3 %.

Wood Group Equipment

The Wood Group equipment consisted of a down hole Wood Group instrument package called a SmartLift Sensor, two ported subs that were connected to the SmartLift Sensor by stainless steel capillary tubes, an RTU that monitored and data logged the monitored data and about 4000 ft of ¼ in tubing encased wire that connected the SmartLift Sensor to the surface RTU. The SmartLift Sensor measured pump intake pressure, pump intake temperature, vibration, sensor current leakage and pump discharge pressure. Wood Group also had a set of pressure transducers at the surface, one to measure tubing pressure and one to measure casing pressure.

ION Unit

The ION 7600 unit is a power measurement unit that can measure and data log each of the three power legs or phases separately. It measures voltage on each leg, current on each leg, frequency, power received and power delivered as well as other power values. The unit was also capable of logging data from other RTUs. It recorded data from the Wood Group RTU, the Micro Motion flow meter and the ABB variable speed drive.

Lufkin SAM Controller

The Lufkin SAM controller is a dynamometer and pump-off controller. It can record surface cards and pump cards for use during diagnostics. It can also be used as a pump-off controller, but during our pump slippage test, this function was not used. The unit is attached to a load cell on the unit that measures the polished rod load. It also had a string gage that could be connected to the pumping unit to accurately measure polished rod position and a tubing pressure transducer to measure tubing pressure. The unit uses this data to calculate surface cards and pump cards.

ABB Variable Speed Drive

The ABB Variable Speed Controller is a 120 hp drive capable of output from 0 to 300 Hz. The unit has many features that can be used to control the speed of the pumping unit. In the slippage tests, the ABB controller was used to set pump speed by selecting a certain fixed frequency on the controller. Changing the speed of the pumping unit with the variable speed drive is much faster than changing motor sheaves. Changing the pumping speed with the ABB variable speed drive was almost instantaneous, while manually changing a motor sheave required a little over an hour. In preliminary testing before the ABB was used to control speed, only four pump speeds could be tested due to having four motor sheaves (6, 8.5, 10, and 12 inches).

EXPERIMENTAL PROCEDURES

Echometer Procedure

Three Echometer Well Analyzers were used to acquire data during the 20-minute slippage tests. Surface dynamometer cards were acquired by measuring load output from a calibrated horseshoe load cell and position was determined from measured polished rod acceleration. Input motor power and current were acquired simultaneously with the dynamometer data. The analyzer calculated and recorded dynamometer cards for every stroke during the entire 20-minute slippage test. At a rate of 30 samples per second the instantaneous flow rate data was acquired from the Micro Motion flow meter and at the same time motor input current, surface tubing and casing pressure were acquired. The elapsed time and motor input current were used to merge the datasets acquired from two different well analyzer systems. Acoustic fluid level measurements were automatically acquired at two-minute intervals to track the fluid level during the slippage test. Each fluid level was used to calculate pump intake pressure. For the slippage test nitrogen gas was used to pressurize the casing to approximately 5 to 7 psi in order to improve the reflected acoustic signals compared to the noise produced by water falling down the casing from the surface back into the well.

The procedure for acquiring the Echometer data was to connect the Texas Tech laptop to the Micro Motion meter and begin acquiring data before the pumping unit was turned on. After the Echometer equipment had been connected, the pumping unit was started. Once steady state was reached in approximately 20-30 minutes, then the well analyzers began acquiring data for about twenty minutes.

After the slippage test was complete, the Texas Tech laptop hooked up to the Micro Motion unit was disconnected after the Micro Motion log had been saved. The data from the ION RTU was then downloaded and saved on to the Texas Tech laptop and backed up to a flash drive along with the Micro Motion log.

The Texas Tech laptop was then reconnected to the Micro Motion meter for the next test. At this time the next test began if it was at the same pumping speed. If the next test was at a different speed the ABB variable speed drive was adjusted. Once the speed change was complete, the next test was begun repeating the procedure.

Lufkin Procedure

Lufkin dynamometer data was collected Lufkin Automation employees. As the well analyzers recorded data, a dynamometer card was collected. Once the test was completed, the data from ION RTU was then downloaded and saved on to the Texas Tech laptop and backed up to a flash drive along with the Micro Motion log. The Texas Tech laptop was then reconnected to the Micro Motion meter for the next test. At this time the ABB variable speed drive was adjusted for the next test. Once the speed change was complete, the next test was begun repeating the procedure.

AUGUST 2006 TEST SETUP

The slippage tests conducted on August 25, 2006 where performed with the same equipment as the previous test on the Red Raider # 1, but with a different pump and a 76 rod string. The pump was 1.5" pump from Harbison Fischer with a clearance of 0.005" and a 4 ft plunger. The Data for each test is listed in Table # 1 in the appendix.

ANALYSIS

As a refresher, the reader will recall that the previous pump slippage progress reports have been conducted at either static condition or at a single pumping speed. In 2000 Progress Report #3 provided the test data for a single speed (6.7 SPM) and a single surface stroke length using a 1.75" pump and various clearances. With this test data the original ARCO-HF equation was developed as shown in equation 2. It should be noted that there was not a "velocity term" in the equation as the data was taken at a single pumping speed.

$$Slippage = 870\frac{DPC^{1.52}}{L\mu} \qquad \text{(Equation 2)}$$

The latest tests at the Texas Tech test well were conducted over a range of pumping speeds in order to evaluate the effects. In the first test at Texas Tech, a 2" pump with a 0.009" clearance was run at various speeds. By combining both sets of test data from the ARCO-Harbison Fischer tests and the 2005 Texas Tech tests, it was possible to present an empirical equation that combines the effect of both pump clearance and pumping speed.

Figure 3 presents the data from the previous test for a 1.75" pump as presented in the Progress Report #3 [3] combined with the Texas Tech 2" bore pump data. The vertical line in this figure is the field data from Progress Report # 3 plotted at a pump speed of 6.6 SPM with all data corrected to TTU Test well conditions by applying the appropriate ratio factors for differential pressure, plunger length and viscosity. It should be noted that field test data taken in 2000 with the 1.75" pump had a measured slippage of 80.1 BPD but adjusts to 64.9 BPD for the 2" pump at the TTU conditions. The actual 2" pump with a 0.009" clearance tested in the test well with an 88 rod string at a speed 6.9 SPM had a measured slippage of 62 BPD. There is close agreement in the slippage numbers between similar operating points from the 2000 data and the 2" TTU data. The ARCO-Harbison Fischer line represents the calculated slippage using Equation 2 and the TTU conditions (2" pump, 0.009" clearance, 48" plunger, 1576 psi pressure differential and 0.76 cp viscosity for water at 90 F) fits very well with the data from the TTU test at 6.6 SPM intersection point. These comparisons provided confidence in the test data between the two tests and in using Equation 2 to predict slippage at similar SPM.

However, as shown in Figure 3, the fluid slippage increases as the pump speed increases. It is also shown that the rod dynamics play a part in the slippage, as the net pump stroke length changes based on speed. So to provide a predictive equation, all of the 2", 0.009" clearance data was used to develop a liner equation of slippage rate versus pump speed. Equation 2 was developed using data at 6.6 SPM and as shown in Figure 3, there is close agreement between the TTU data, the 2000 data and Equation 2 at 6.6 SPM. This indicated that a SPM modifier applied to equation 1 could provide reasonable slippage values.

Using a pump speed of 6.6 SPM as the base, a slippage ratio versus pump speed ratio can be developed as the multiplier as shown in Figure 4. A slippage ratio of 1.0 and a speed ratio of 1.0 occur at 6.6 SPM. Equation 1 is reduced to Equation 2 at 6.6 SPM.

Figure 5 demonstrated the accuracy of the Equation 1 with the SPM modifier. The dots are the measured slippage data from the TTU test with the 2" pump at 0.009" clearance and the solid line is the predicted values from Equation 1. The dotted lines are error bars set at ± 10% and ± 20%. For all of the data taken, the range of error to the predicted equation is +12 % to – 20%. This is considered acceptable.

Once the slippage rate is known, the volumetric efficiency of the pump can be calculated. The measured efficiency is the surface measured flow rate divided by the pump displacement using the effective plunger stroke length.

$$\Pr oduction = 0.1166 \cdot SPM \cdot SL \cdot D^2 \qquad \text{Equation 3}$$

$$Efficiency = \frac{SurfaceRate}{PumpDisplacment} \qquad \text{Equation 4}$$

Calculated efficiency = (pump displacement – pump slippage) / pump displacement   Equation 5

Figure 6 shows the calculated pump efficiency vs. pumping speed (SPM). The efficiency dramatically decreases at low pumping speeds . Using the ABB VSD Drive to reduce pump speed, it is possible to have slow enough pump speed that all of the pump's displacement is lost to slippage. The solid line shows the calculated pump efficiency with the average effective pump stroke length of 98 inches with the dotted lines being the calculated pump efficiency at the 91.5" minimum effective stroke length and the 105.6" maximum effective stroke length.

Equation 1 was developed using data from a 1.75" pump with various clearances at the same pump speed and a 2" pump at the same clearance but at different pump speeds. After the 2" pump tests were completed, a 1.5", .005" clearance pump was run on a 76 rod string and the tests repeated at a 105.6" surface stroke length and multiple pump speeds. The results were similar to the 2" pump with slippage increasing with speed. Figure 7 shows the data from the 1.5", 0.005" clearance pump compared to the predicted results from Equation 1. The solids line is the predicted results using Equation 1 with the test conditions of the TTU test (1616 psi plunger pressure differential, 48" plunger and 0.76 cp water viscosity). The horizontal dashed line is the ARCO-HF prediction, Equation 2. The lines parallel to the calculated are error bars of ± 10% and ± 20%. The equation predicted the actual slippage with an error of 0% to a maximum of about 30% to the actual measured values. It should be noted that the error increased as the speed decrease with the 30% error at the 2.5 SPM test point. Above 6 SPM, the error range was 0% to about 12%. Over the range of the test points, Equation 1 provided reasonable results for a pump size not used in the development of the empirical equation.

Figure 8 shows the efficiency of the 1.5" .005" clearance pump. The increasing error with slower speed can also be seen in this figure.

Figures 9 and 10 show the combined data for the 2", 0.009" clearance pump and the 1.5", 0.005" clearance pump at 105.6" surface stroke length from the TTU test well. Figure 9 shows the slippage rate versus pump speed and Figure 10 shows the calculated efficiency versus pump speed for both pumps.

Overall the new equation presented as Equation 1 has predicted the slippage with reasonable accuracy over a wide range of SPM and pump diameters and clearances. There can be some additional refinement of this equation to improve prediction but additional tests are needed.

It should be recognized by evaluating Equation 1 that the input data is very critical in the results of the predicted slippage, especially the fluid viscosity and the pump pressure differential. These two variables are the operating conditions down hole and will have the greatest uncertainty in the input data. The fluid temperature and the pressure differential were measured in the TTU test but in the field these may not be available without some investigation.

OBSERVATIONS

- Data from Texas Tech Pump Slippage test is consistent with previous field tests.
- Pump slippage increases with increasing pump speed
- Pump displacement increases faster than pump slippage resulting in greater pump efficiency with increasing speed.

CONCLUSIONS & RECOMMENDATIONS

In conclusion, the following empirical equation is presented as being the best predictive tool for rod pump slippage at the present time. As more testing is done, a few refinements may be made; however, the current accuracy seems to well within plus or minus 20%, which is sufficient for most applications.

$$Slippage = [(0.14 \cdot SPM) + 1] 453 \frac{DPC^{1.52}}{L\mu} \qquad \text{(Equation 1)}$$

Based on this work and previous work, the following minimum pump clearances are recommended for a 48" Plunger with a "+1 Barrel". These clearances have become widely used in the Permian Basin for well depths up to 8000 feet.

- 1.25" pump = -3 to -4 plunger (0.004" to 0.005" total clearance)
- 1.50" pump = -4 to -5 plunger (0.005" to 0.006" total clearance)
- 1.75" pump = -5 to -6 plunger (0.006" to 0.007" total clearance)
- 2.00" pump = -6 to -7 plunger (0.007" to 0.008" total clearance)

By unanimous consent of all test participants, it is agreed that Equation 1 should henceforth be referred to as the "Patterson Equation" in honor of John C. Patterson who has spearheaded the effort since it's inception in 1996.

ADDITIONAL WORK

As additional funds and schedule allow, the following test will be conducted at the Texas Tech test well:

- 1.5" pump with 0.005" clearance run at different stroke lengths to quantify the plunger velocity effect and confirm the SPM multiplier
- 1.75" pump with 0.006" clearance to obtain a different set of data refine the empirical equation, Equation 1

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Nomenclature

Symbol		Unit
D	Plunger Diameter	in
C	Plunger – Barrel Clearance	in
L	Plunger Length	in
M	Viscosity	cp
P	Pressure Differential across Plunger	psi
SPM	Pump Speed	strokes per minute
SL	Effective Stroke Length from Dynamometer Downhole Card	in

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Table 1
Slippage Data

Test #	Date	Rod String	Stroke Length (in)	Control Method [3]	Frequency (Hz) [4]	Pump Speed (spm)	Pump Intake Pressure Woodgroup (psig)	Pump Intake Pressure Echometer (psig)	Echometer Inferred Production (bpd)	Lufkin Inferred Production (bpd)	Surface Production (bpd)	Echometer Slippage (bpd)	Lufkin Slippage (bpd)	Average Slippage (bpd)	Pump Efficincy (%)
1-01	7/8/05	76 [1]	105.6	ABB (12")	60	9.73	gauge down	161.5	427.72	424.8	367.1	60.6	57.7	59.1	86.1
1-02	7/8/05	76 [1]	105.6	12" sheave	60	9.74	gauge down	152.6	428.11	425.6	368.0	60.1	57.6	58.8	86.2
1-03	7/8/05	76 [1]	105.6	ABB (12")	51	8.25	gauge down	165.5	357.49	350.2	301.3	56.2	48.9	52.6	85.1
1-04	7/8/05	76 [1]	105.6	ABB (12")	43	6.93	167	167.7	297.36	292.6	242.4	55.0	50.2	52.6	82.2
1-05	7/8/05	76 [1]	105.6	ABB (12")	31.5	5.03	165.7	172	214.7	214.0	163.5	51.2	50.5	50.9	76.3
1-06	7/8/05	76 [1]	105.6	ABB (12")	na	1.82	183.2	182.7	81.5	81.0	41.6	39.9	39.4	39.6	51.2
2-01	7/28/05	88	105.6	ABB (12")	.8 spm	0.80	175	178.1	39.2	na	5.6	33.6	na	na	14.2
2-02	7/28/05	88	105.6	ABB (12")	.7 spm	0.70	178	178.1	34.4	na	4.4	30.0	na	na	12.8
2-03	7/28/05	88	105.6	ABB (12")	.6 spm	0.60	179	178.1	29.55	na	0.0	29.6	na	na	0.0
2-05	7/28/05	88	105.6	12" sheave	60	9.72	150	165.4	444.6	437.8	377.9	66.7	59.9	63.3	85.6
2-06	7/28/05	88	105.6	ABB (12")	60	9.71	150	151.7	444.6	440.0	378.2	66.4	61.8	64.1	85.5
2-07	7/28/05	88	105.6	ABB (12")	51	8.22	153	149.9	371.6	370.0	308.6	63.0	61.4	62.2	83.2
2-08	7/28/05	88	105.6	ABB (12")	43	6.90	156	154.6	313.4	312.6	250.9	62.5	61.7	62.1	80.2
2-09	7/28/05	88	105.6	ABB (12")	31.5	5.01	156	163	224	223.6	170.2	53.8	53.4	53.6	76.0
3-01	7/5/05	76 [1]	105.6	16" Sheave	60	12.97	gauge down	na	na	591.1	496.4	na	94.7	na	84.0
4-01	7/14/05	FG [2]	87.5	16" Sheave	60	12.95	145	na	na	641.7	565.8	na	75.9	na	88.2
5-01	7/14/05	FG [2]	87.5	ABB (16")	72.5	15.47	138	na	na	868.3	777.2	na	91.1	na	89.5
6-01	7/26/05	88	105.6	16" Sheave	60	12.92	146	na	na	625.7	540.1	na	85.6	na	86.3
6-01	8/25/06	76 [5]	105.6	ABB (12")	31.5	5.1	141	na	na	130	109.5	na	20	na	84.2
6-02	8/25/06	76 [5]	105.6	ABB (12")	43	7.1	135	na	na	184	162.7	na	21	na	88.4
6-03	8/25/06	76 [5]	105.6	ABB (12")	51	8.3	138	na	na	217	189.4	na	28	na	87.3
6-04	8/25/06	76 [5]	105.6	ABB (12")	60	9.7	132	na	na	255	229.6	na	25	na	90.0
6-05	8/25/06	76 [5]	105.6	ABB (12")	60	9.7	130	139.3	254.2	na	230.1	24.1	na	na	90.5
6-06	8/25/06	76 [5]	105.6	12" sheave	60	9.7	130	122.4	254.7	na	232.1	22.6	na	na	91.1
6-07	8/25/06	76 [5]	105.6	ABB (12")	51	8.3	134	122.4	207.9	na	185.1	22.9	na	na	89.0
6-08	8/25/06	76 [5]	105.6	ABB (12")	43	7.1	135	116.8	180.4	na	159.1	21.4	na	na	88.2
6-09	8/25/06	76 [5]	105.6	ABB (12")	31.5	5.1	135.7	132.6	127.0	na	107.6	19.4	na	na	84.7
6-10	8/25/06	76 [5]	105.6	ABB (12")	na	2.5	141	142.2	62.5	na	45.5	16.9	na	na	72.9

A 2.00 in pump with a 0.009 in clearance and 4 ft plunger was used for tests 1 thru 5
A 1.50 in pump with a 0.005 in clearance and 4 ft plunger was used for test 6

1. 76 string has 1468 ft of 7/8, 2000 ft of 3/4, and 400 of 7/8 rods
2. Fiberglass string had 2818 ft of 1" fiberglass and 1050 ft of 1 5/8" sinkerbars
3. When sheave size is listed, ABB was bypassed - when ABB used, sheave size is in parentheses
4. If pump speed listed, drive was set for constant speed, not constant frequency'
5. 76 string has 1950 ft of 7/8 and 2002 ft of 3/4

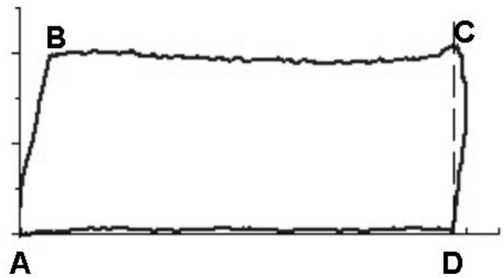

Figure 1- Pump Card

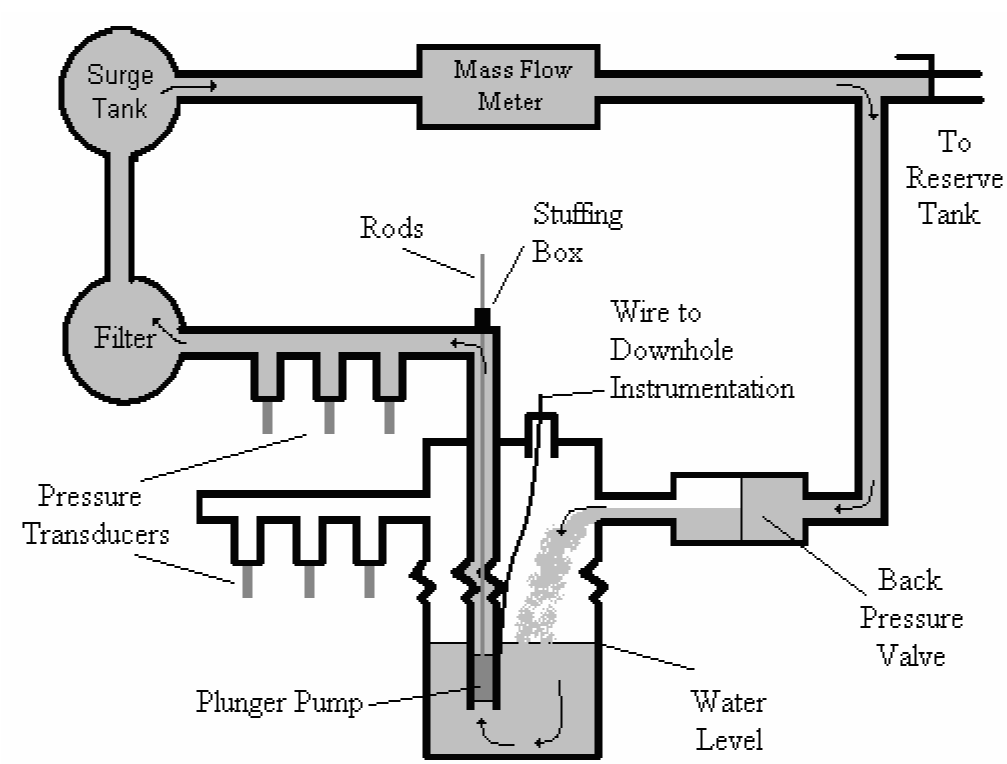

Figure 2 - Flow Loop Diagram

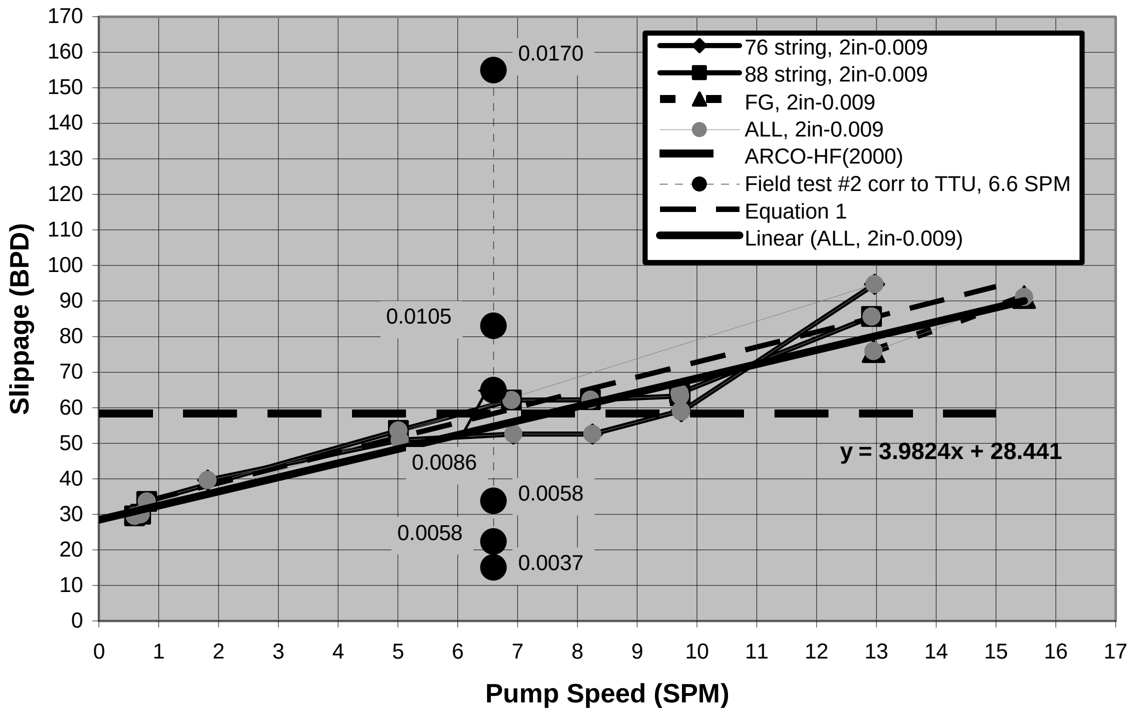

Figure 3 - Data from Previous Test

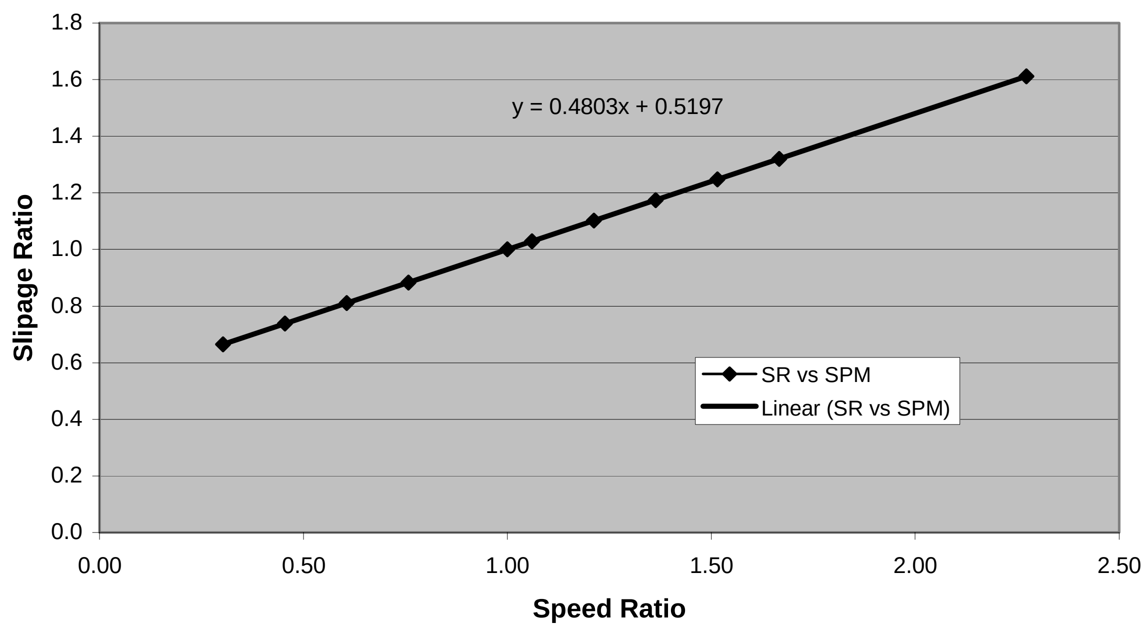

Figure 4 - Slippage Ratio vs. Speed Ratio

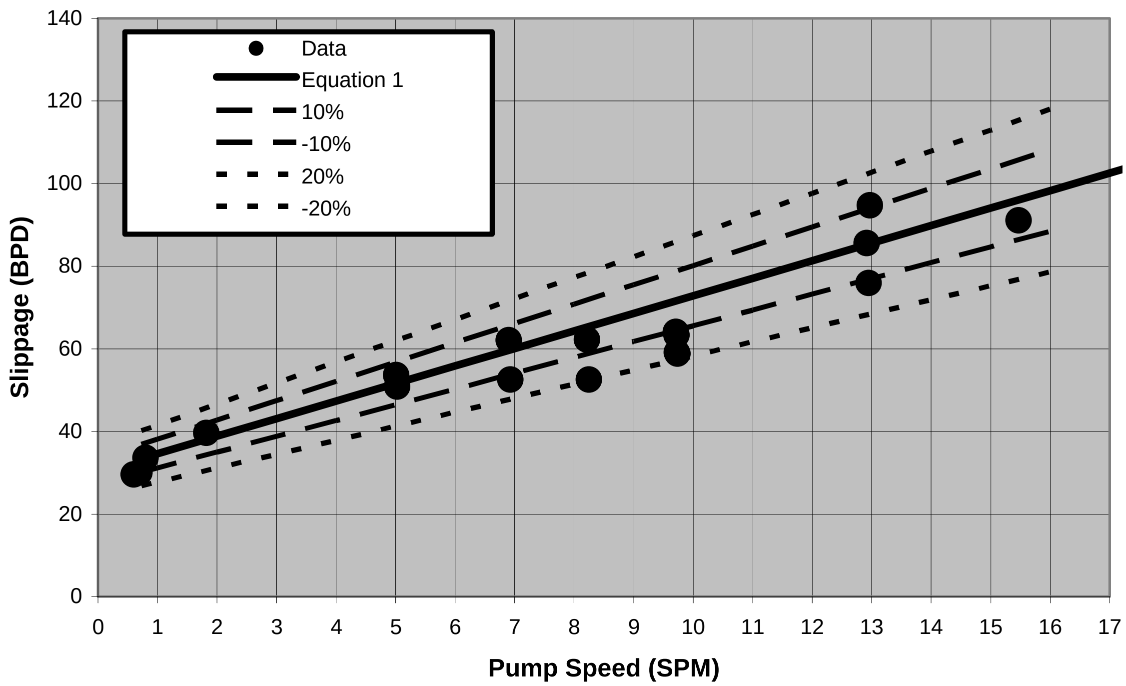

Figure 5 - Slippage vs. Pump Speed with Error Bars

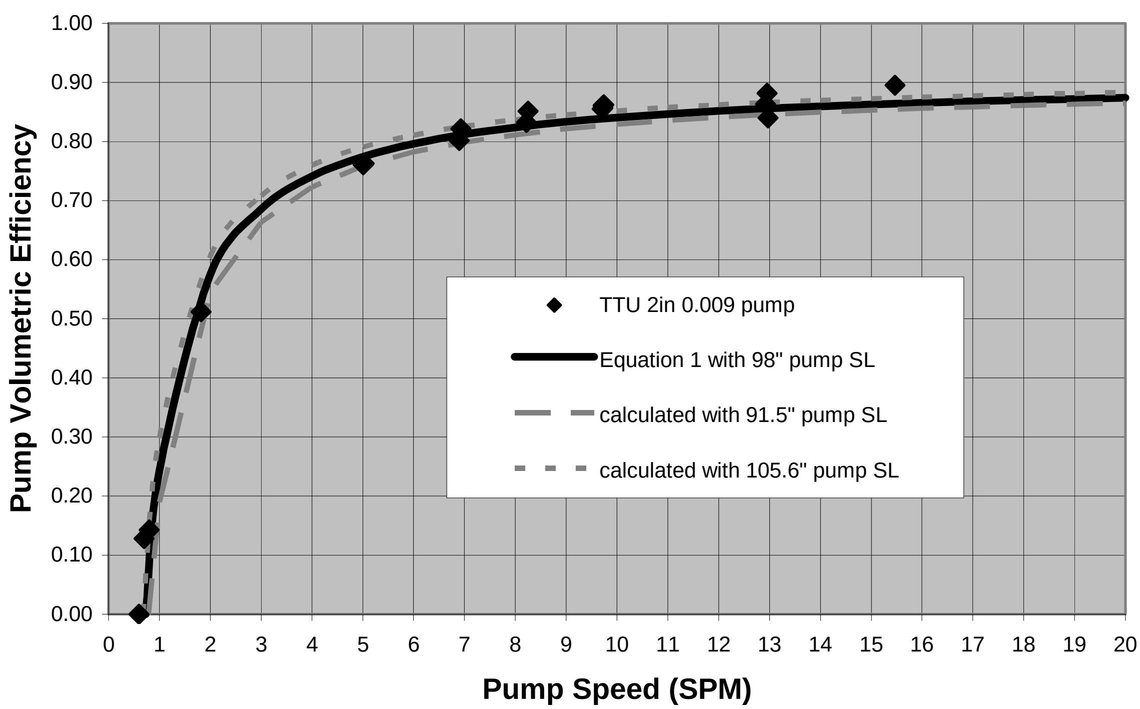

Figure 6 - Pump Efficiency vs. Pump Speed

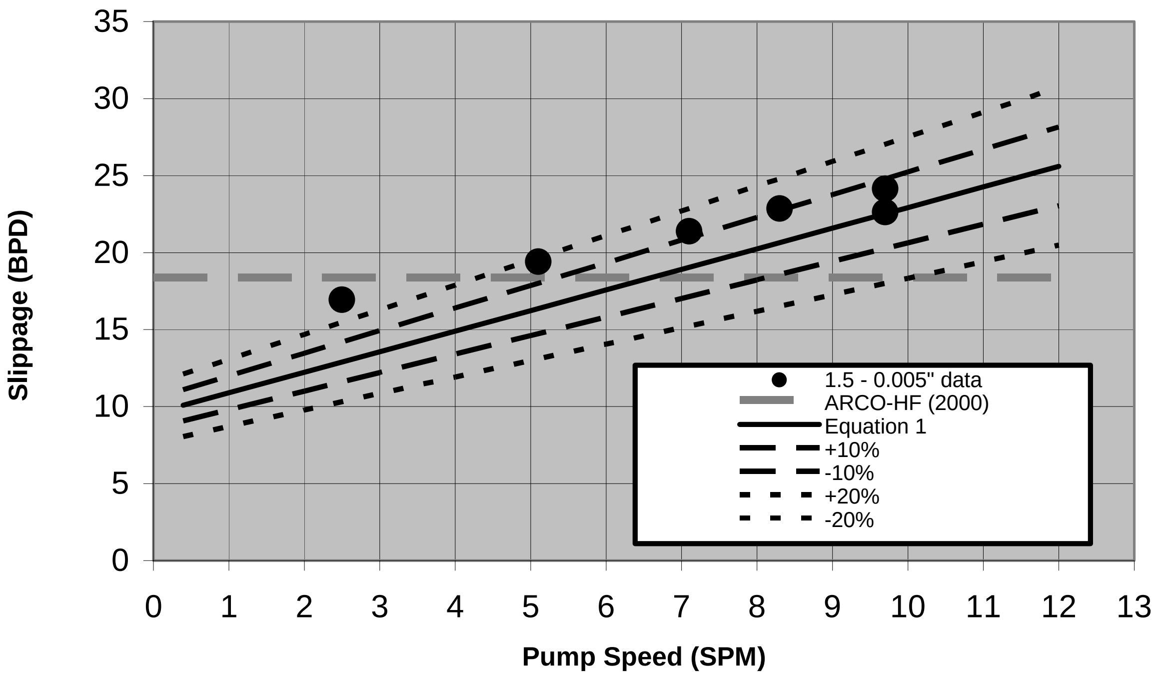

Figure 7 - 1.5” Pump Data with Error Bars

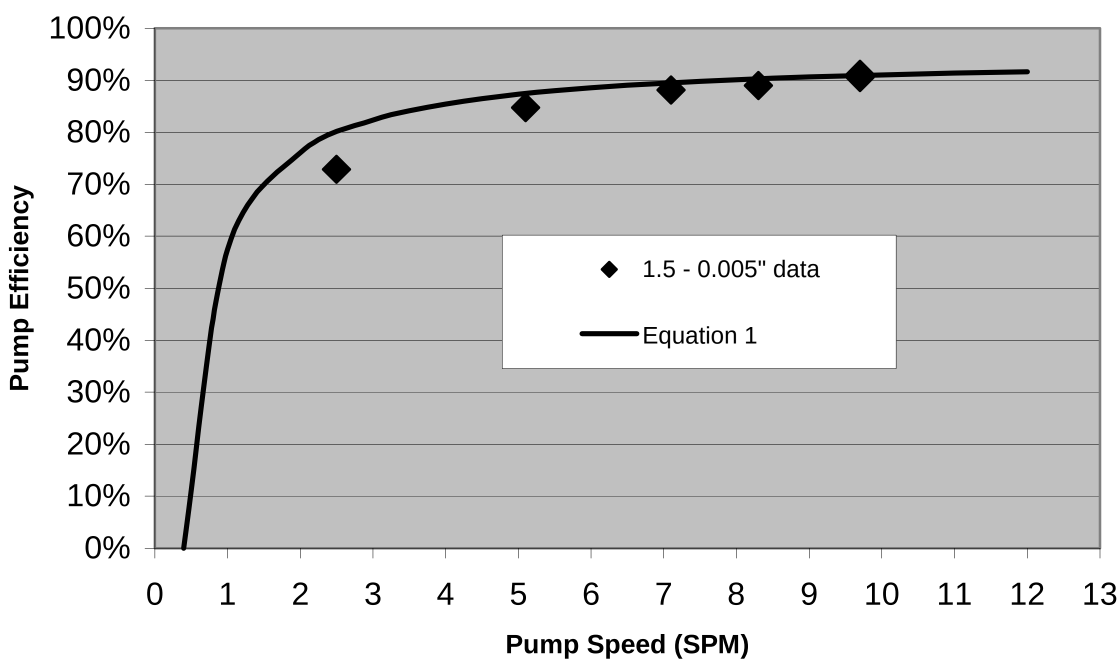

Figure 8 - 1.5” Pump Efficiency

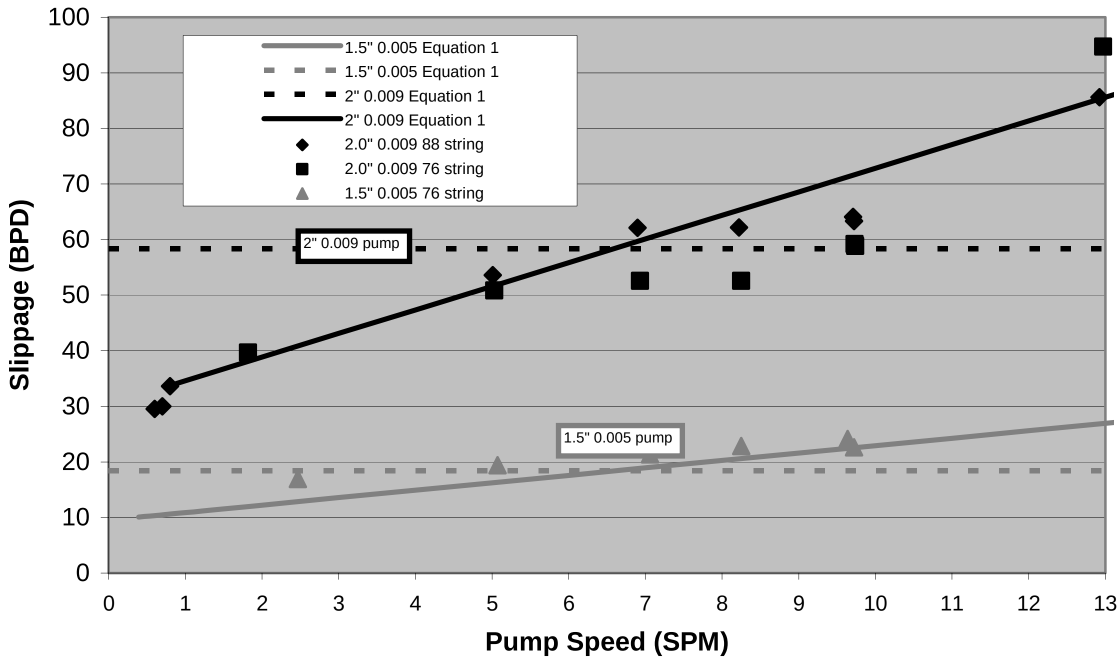

Figure 9 - Slippage Comparison of 2.0” Pump to the 1.5” Pump

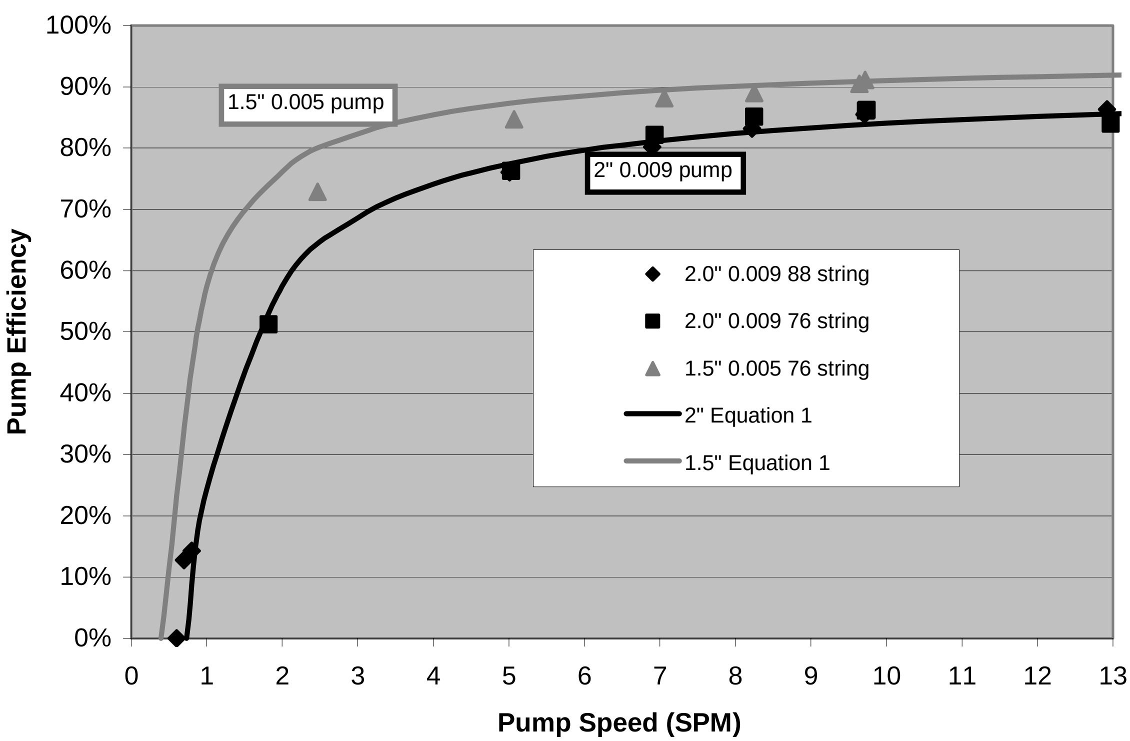

Figure 10 - Efficiency Comparison of 2.0” Pump to the 1.5” Pump