PRESSURE BALANCED SUCKER ROD PUMP WITH ENGINEERED HYDRODYNAMIC VALVES

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INTRODUCTION

The sucker rod pump (SRP) valve ball and seat design, in large part, has been unchanged since circa 1938. The authors questioned if there is an opportunity to improve the sucker rod pump for today's challenging, high decline rate, gassy-sluggy-foamy and solids ladened producing horizontal wells.

Today's deep, gassy-sluggy, foamy, solids ladened, horizontal wells commonly have initially high liquid rates that exceed the rate capacity of sucker rod pumping. These rates require use of higher operating expense artificial lift systems such as electrical submersible pumps (ESP's) or gas lift methods. Improving the liquid rate capacity and reliability of sucker rod pumping in such challenging environments would be highly beneficial for producers. Later in a well's producing life, the ability for a sucker rod pumping system to efficiently and reliably handle gas and solids at very low pump intake pressures (PIP's) would also be beneficial.

The sucker rod pump is one component of a complex downhole system of components for sucker rod pumping. Other components of this system include a downhole gas separator, a downhole solids separator, a tubing anchor and sucker rods. More recently, complexity has increased with recent bottomhole assembly innovations using extended dip tubes that place the downhole gas separator deep in the curve. To maximize the efficiency and performance of a sucker rod pump, all these components must act together harmoniously to effectively feed the pump on demand with liquid that has been gas and solids depleted – unfortunately, achieving this has been particularly challenging. Consequently, the sucker rod pump must still contend with gas and solids.

Further, for deep high-rate sucker pumping, an acceptable failure frequency has been particularly challenging. Larger and longer stroke length pumping units have improved the rate capacity of sucker rod pumping but have been limited by excessive pressure losses in the system prior to the pump and through the pump's standing valve. Other limitations include pump gas interference and increased failure frequency from produced solids. Lastly, pump gas interference induces erratic compressional loading on the sucker rods and has reduced system reliability.

An improved sucker rod pump was conceptualized, and design engineered for such challenging environments, with the following features:

- minimal standing/travelling valve pressure loss at high pump rates and pump plunger velocities,
- solids tolerant at high concentrations of solids (for example, from concentrated solids slugging events),
- scale formation and adhesion resistant,
- can operate efficiently at all inclinations up to 90 degrees, and
- pressure balances the pump's travelling valve prior to commencement of the pump's downstroke to minimize sucker rod compressional loading events and to avoid efficiency losses due to pump gas interference.

The innovative Vortex Barbell System^{TM} pump valves have demonstrated a step change in performance for high inclination pumping conditions. This unique valve design revealed a transformational opportunity to improve the sucker rod pump valves.

A sucker rod pump has been limited by use of machined componentry and a ball on seat valve design. Three-Dimensional (3D) metal printing or additive manufacturing has gained significant attention in recent years. The ability to now print hard and tough metals has offed the opportunity to engineer and manufacture reliable sucker rod pump valves with very low-pressure losses, minimal flow turbulence and improved solids handling – we are no longer design limited by the ball and seat design from circa 1938. Further, improvements in surface treatments and coatings for metals has also improved wear performance in valves. A new complex shaped hydrodynamically engineered rod pump valve was developed for minimization of pressure loss and high solids tolerance.

A pressure balanced pump, has offered advantages for reducing the negative impacts of pump gas interference and compressional rod loading events. But this pump design can be limited by solids and can require precise pump space-outs. It was hypothesized that instead of tapered top barrel section, a rifled channeled top barrel section would resolve existing limitations. A rifled channel offered much greater solids tolerance and avoided the need for precise pump space-outs.

Flow loop testing and early field trials have indicated promise for improvement.

SUCKER ROD PUMPING LIMITATIONS

Figure 1[i] shows a typical sucker rod pump ball and seat, which in large part, has been unchanged since circa 1938[ii].

A typical sucker rod pump uses two ball and seat valves to form the pump's volumetric displacement chamber – a fixed/stationary standing valve and a reciprocating travelling valve. A positive pressure is required during a pump's upstroke for the pump to fill its chamber with fluid. This pressure is commonly referred to as the Pump Intake Pressure (PIP) in the wellbore annulus adjacent to the pump.

Figure 2[iii] illustrates bottomhole assembly (BHA) components for a typical sucker rod pumping system. Figure 3 shows a more recent horizontal well innovation BHA that uses an “extended dip tube” for placing the gas separator deeper into the curve at approximately 45 degrees inclination, while the pump remains in the vertical well section. A gas separator placed at an inclination improves its separation efficiency. Saponja, Kubacak and Nagoo[iv] revealed this separator efficiency improvement with separator placement at inclinations.

For production maximization, it is desirable to have PIP as low as possible. The amount of pressure differential formed between the pump’s chamber (during the pump’s upstroke) and the PIP is a function of fluid flow pressure losses from the wellbore through all the BHA components and into the pump’s chamber. The sum of all the pressure losses through the gas separator, through the piping between the gas separator and the pump (including an extended dip tube, if used) and through the pump’s standing valve is a major factor that limits the rate capacity of a sucker rod pumping system. In other words, higher system pressure losses require higher PIP’s.

A sucker rod pump can also be limited by efficiency losses from:

- free gas, solution gas liberation due to pressure losses and entrained gas (foaming tendencies) entering the pump, which reduces the effective pump stroke length for displacing fluid, and
- lack of pump compression ratio, which reduces the amount for fluid that can be displaced from a pump.

A sucker rod pumping system can experience reduced reliability or higher failure frequency from:

- pump induced rod buckling during downstroke,
- scale formation and adhesion,
- plugging and/or seizing (i.e., stuck plunger) from solids, and
- damage from solids laden produced fluids.

AN IDEAL SUCKER ROD PUMP FOR HORIZONTAL WELLS

A sucker rod pump is an essential component for rod pumping, but it has been limited by use of machined componentry and a ball on seat valve design. Today’s deep, gassy-sluggy, foamy, solids ladened, horizontal wells commonly have high initial liquid rates that exceed the rate capacity for sucker rod pumping. There rates can require use of higher operating expense artificial lift systems such as electrical submersible pumps (ESP’s) or gas lift methods. Improving the liquid rate capacity and reliability of sucker rod pumping in such challenging environments would be highly beneficial for producers. Later in a well’s producing life, the ability for a sucker rod pumping system to allow higher rate pumping at a very low pump intake pressures, while efficiently and reliably handling gas and solids would also be beneficial.

It was conceived that an ideal sucker rod pump should embody the following features:

- has near-zero pressure loss and minimal flow turbulence through the standing and travelling valves, such that the pump easily fills at very low PIP’s, allowing lower PIP’s, less pump gas interference, less fluid pound risk, and less scaling risk,

- has very high valve efficacy at any inclination,
- improves the performance of extended dip tube systems for horizontal wells,
- has high reliability with no valve insert "ball" chatter and its valve insert has low weight for minimizing seat impact load during closing,
- it automatically pressure balances the pump chamber and travelling valve close to or at the top of the upstroke, mitigating foamy fluid pump gas interference and pump induced rod buckling,
- has high corrosion tolerance, and
- it efficiently conveys solids through the pump with minimal erosion damage and avoids valve insert solids sticking risks, as well as avoids solids related stuck pumps risks.

SUCKER ROD PUMP VALVE FLOW TURBULENCE AND PRESSURE LOSS

A ball and seat type valve design will always experience flow turbulence and will have a pressure loss. The valve's insert or object (i.e., ball) is directly in the flow path and forces fluids to flow around it, which causes pressure loss and flow turbulence.

Pressure loss through sucker rod pump valves can undesirably cause:

- increased pump gas interference with liberation of solution gas from the oil and an earlier onset of pump fluid pound,
- higher PIP's, as the greater the pressure loss through a standing valve, the higher the PIP required to fill the pump for given pump rate,
- valve damaging cavitation and vibration (ball chatter),
- increased valve wear,
- increased scale formation and adhesion,
- increased peak polish rod loads due to less pump chamber pressure and,
- increased and more erratic sucker rod compressional loads (i.e., lowermost rods under a compressional load push down the pump's plunger)

Mahoney[v] explains that pump gas interference occurs when gas is pulled into the pump's chamber at a lower pressure than PIP and gas expands due to this lower pump chamber pressure – the consequence is less liquid enters the pump, which reduces pump efficiency. McCoy etal[vi] pointed out that the origins of the gas inside the pump barrel include free gas that may be present at the pump intake and/or from gas that evolves (i.e., liberation of dissolved solution gas in the oil) from the liquid due to the pressure drop caused by flow through the pump's intake system (i.e., from the wellbore through the separator, through the pump intake dip tube and through the pump's standing valve).

Salazar's[vii] useful research discovered that gas and liquid two phase flow (compressible) into a sucker rod pump chamber resulted in a 50% increase in pump chamber's pressure during the pump's downstroke stroke as compared to liquid (incompressible) only entering a pump's chamber. Consequently, any presence of gas entering a pump will undesirably increase the pressure required to open the travelling valve on the downstroke and therefore undesirably increases compressional loads on the sucker rods.

Pressure loss through a standing valve was researched by Cutler etal[viii]. Their flow loop used water at flow rates up to 50 gallons per minute (GPM) or 1,700 barrels per day. They tested various standard standing valve types including two closed cage valves with different size balls and seats, an insert guided valve, and two non-standard standing valves designed for high efficiency (i.e., low pressure loss). They found that using a smaller ball and larger seat internal diameter (ID) reduced the pressure drop in closed cage designs and approximated the same pressure drop as an open cage design, with pressure losses between five (5) psi and eight (8) psi at 50 GPM water rate. Testing of the high efficiency standing valves showed dramatically improved pressure loss performance compared to the standard valves, with the pressure drop staying below one (1) psi over the full flow range. They also tested 100% crude oils with up to 100 centipoise (cP) viscosity and found that at higher flow rates the pressure loss for high viscosity fluids becomes more pronounced but only changed the pressure loss by six (6) percent. They then modelled higher viscosities and found for 350 cP at 50 GPM the pressure loss substantially increased to approximately 50 psi.

An interesting discovery during their testing was the impact to pressure loss effect from valve ball chatter (i.e., valve insert chatter). Ball chatter increased flow turbulence and dramatically increased the pressure loss through the standing valve by five (5) times. They concluded that sucker rod pump valves should be engineered to avoid ball chatter for minimizing pressure loss.

Coyes[ix] proved that a high efficiency insert cage valve design using vortex flows decreased pressure loss by 40% over a bar bottom style insert cage and considerably reduced ball chatter (i.e., had less turbulence). Pressure losses of approximately 4 psi were recorded at 50 gpm using water and somewhat less when air was mixed with the water.

APPARENT VISCOSITY OF CRUDE OIL AND WATER MIXTURES AND THE IMPACT PRESSURE LOSS

The previously cited research discussed laboratory determined pressure losses through sucker rod pump valves that was conducted with either oil or water alone (mutual exclusively), and not as a mixture of oil and water. Most wells commonly produce fluids that are a mixture of oil and water. Therefore, understanding fluid dynamics and mechanics of oil and water mixtures would be more representative of downhole sucker rod pumping conditions.

Oil and water mixtures can have high apparent viscosities, which increases fluid flow frictional pressure loss. Apparent viscosity is defined as the shear stress applied to a fluid divided by the shear rate. Lv[x] showed that oil and water mixtures have varying apparent viscosities and can reach a relatively high apparent viscosity of 350 cP for light oils at 50% to 60% percent water to oil ratio. Figure 4 shows the results of their research in terms of pressure loss as a function of the oil and water mixture ratios, showing a peak of pressure loss approximately ten (10) times an oil or a water phase alone. Figure 5 from their research shows oil and water dispersion flow patterns which influences the magnitude of mixture apparent viscosity – the smaller the oil dispersed droplets the greater the apparent viscosity of the mixture.

Zhang [xi] concluded that unstable oil-water dispersed mixtures always show the characteristics of a non-Newtonian fluid. The apparent viscosity of oil and water emulsions (stable and unstable) is affected by several factors including viscosity of continuous phase, volume fraction of dispersed phase, droplet distribution of dispersed phase, viscosity of dispersed phase, temperature, shear rate, density of continuous phase, and the density of dispersed phase.

Figure 6 shows a field recorded example for how high apparent viscosity from a produced oil and water mixture can affect the peak polish rod load while rod pumping. The peak polish rod load increased by approximately ten (10) percent when the oil and water mixture entered the pump after producing water-based load fluid.

Apparent viscosity directly impacts the Reynolds number. Figure 7 from White[xii] illustrates laminar and turbulent flow conditions as a function of Reynolds number. It also shows the Reynolds number equation, and that Reynolds number is inversely proportional to apparent viscosity. If apparent viscosity increases, Reynolds number decreases and flow becomes more laminar. If apparent viscosity decreases, Reynolds number increases and flow becomes more turbulent. Consequently, designing a new valve to avoid turbulent flow conditions needs to consider that turbulent flows may not actually exist at the anticipated fluid rates with a high apparent viscosity oil and water mixtures.

It is concluded that flow loop laboratory testing with water or oil alone (i.e., individually and not as a mixture) for assessing pressure loss and flow turbulence, can be misleading. Test results will likely not represent typical downhole producing conditions. Any such tests should be adjusted or compensated for the apparent viscosities anticipated with oil and water mixtures or test with oil and water mixtures. For example, pressure loss can be ten (10) times higher than what was found in a water only flow loop test, so a pressure loss should be multiplied by ten (10).

TOTAL PRESSURE LOSS IN A CYCLICAL FLUID FLOW SYSTEM

Within a sucker rod pumping system, pressure loss is not only a function of fluid flow velocity friction. There is an additional pressure loss when fluid is accelerated. Fluids are accelerated from a static condition to a dynamic “moving” condition (i.e., a high fluid acceleration rate) very rapidly during a sucker rod pump’s upstroke. For when there is fluid acceleration in a piping system and if the fluid velocity increases, kinetic energy increases. This energy must come from somewhere and it comes at the expense of pressure in the form of a pressure drop or loss.

For cyclical flow sucker rod pumping system, pressure loss is a function of two factors that must be added together to assess the total pressure loss of the system:

1. Frictional Pressure Loss – due to viscous fluid flow resistance in a pipe or around an object and can be calculated by the Darcy Weisbach[xiii] equation.
2. Acceleration Pressure Loss – due to the force needed to increase a fluid’s velocity over a given time period.

In Figure 8, Crane[xiv] provides an industry accepted equation for fluid acceleration pressure loss, which shows the higher the acceleration rate, the greater the acceleration pressure loss. Rowland[xv] explained in Figure 9 that sucker rod pump plunger velocities during the upstroke stroke are highly variable. He showed how a pump plunger's upstroke rapidly accelerates, starting from zero (0) inches per second to eighty (80) inches per second in just one second.

A sucker rod pump can have upstroke peak plunger velocities and instantaneous pump intake/discharge liquid rates that are four (4) times the average. Figure 10 from Guzman's[xvi] research showed that a SRP's plunger velocity profile during the pump's upstroke is highly variable and not constant. Corresponding, the liquid rate into and out of the pump is therefore also highly variable and not constant. For example, the pump jack at surface can be one third the way up on its upstroke before the downhole pump's plunger starts moving (due to sucker rod string stretch and sucker rod friction). Then the plunger's velocity "slingshots" under extremely high acceleration to a peak plunger velocity. It is fundamental to understand that the plunger's velocity profile and intake liquid rate can vary from zero (0) to over four (4) times the average each pump upstroke. This importantly points out, for example, that for a well with an SRP producing an average 200 bbls per day liquid, an instantaneous peak liquid rate entering the pump can be 800 barrels per day (each pump stroke). For example, at six (6) strokes per minute, the pump's liquid intake rate during an upstroke increases for zero (0) to 800 barrels per day and then decreases back to zero (0) in five (5) seconds. To this end, the technical engineering considerations and challenges for rod pumping downhole BHA design and pump valve design are that the SRP intake liquid rates have high acceleration rates and vary over an extensive rate range each pump stoke, where both frictional and acceleration pressure losses must be considered.

Figures 11 and 12 show ChatGPT4o's example calculations of total pressure loss (friction plus acceleration) for a 370 foot long 2-3/8" EUE tubing section of an extended dip tube system. The figures compare 1 cP viscosity water versus 350 cP apparent viscosity crude oil and water mixture at a peak fluid rate of 800 barrels per day. The calculated total pressure loss was 15.9 psi (with frictional at 0.96 psi and acceleration at 14.94 psi) for water only and 49.2 psi (with frictional at 35.7 psi and acceleration at 13.5 psi) for a crude oil and water mixture.

Assessing the total system pressure loss for sucker rod pumping BHA's, it can be seen in Figure 13 that an oil and water mixture can have considerable total pressure loss, especially with an extend dip tube BHA. Relatively high PIP's would therefore be required for filling a pump at the peak upstroke plunger velocity. For example, using a 350 foot long 2-3/8 inch tubing extended dip tube system, with an average pump rate of 400 barrels per day (peak plunger velocity rate of 1,600 bbls/day) would require a PIP greater than 180 psi to fill the pump. This relatively high pressure loss through an extended dip tube system will risk liberation of a meaningful amount of solution gas from the oil, which will lower the pump fillage on a continuous basis (i.e., a level of pump gas interference will always be present during pumping).

RESEARCH FOR REDUCING SUCKER ROD PUMP VALVE PRESSURE LOSS AND TURBULENCE

A sucker rod pump valve's features that affect valve pressure loss and turbulence are as follows:

1. shape and cross-sectional area of the valve's guide cage conduit,
2. shape and cross-sectional area of the valve's closing "object" and its surface area, and
3. surface finish of the wetted areas.

Pressure loss and flow turbulence are closely related. Their relationship is that turbulence causes chaotic, swirling fluid motion that increases flow resistance, consuming energy and disrupting smooth flow, which leads to greater pressure loss through a flow conduit. Pressure loss is a symptom of energy dissipation. When flow is turbulent, some of the fluid's energy gets converted into heat and eddies rather than moving fluid forward. This energy loss manifests as a pressure loss from a flow conduit's inlet to outlet.

Flow conduit dimensional changes (i.e., cross-sectional flow conduit area contractions and expansions) and/or directional changes (i.e., bends, elbows, turns) cause increased pressure loss and turbulence. Turbulence is often caused by sharp turns, narrow gaps, abrupt expansions or contractions (for example, at a valve seat) and when flow goes around an object (for example, around a ball).

The surface finish of the wetted area (i.e., the internal surfaces in contact with the fluid) significantly impacts pressure loss and turbulence. When fluid moves along a surface wall, it interacts directly with the wall through the fluid's boundary layer – a thin layer of fluid adjacent to the wall slowed down by wall's friction. The condition of the wall affects this boundary layer dramatically, where a rougher surface having more friction than a smooth surface. A smoother wetted surface leads to less turbulence and lower pressure loss because it reduces friction between the fluid and the wall, preserving energy and maintaining flow efficiency.

When it comes to pressure loss and turbulence, a valve's closing object's shape and size matters, and the shape of the valve guide cage's flow path conduit. Shape and surface area of the valve's closing "object" insert affects the magnitude of hydrodynamic flow drag. Hydrodynamic flow drag is the force on an object that restricts the velocity of the fluid past the object. In other words, it is the resistance a fluid experiences as it flows past an object. Pressure loss is a direct consequence of the work done by the fluid to overcome hydrodynamic flow drag. In valves, the valve's insert closing object and the guide cage conduit, hydrodynamic flow drag is what causes the pressure loss. The are two components associated hydrodynamic flow drag over an object:

- Form Drag – flow drag due to the cross-sectional size and shape of an object in a flow conduit and the amount of turbulence it generates (also referred to as pressure drag).
- Skin Friction Drag – flow drag due to the friction of the flow along an object's surface area.

Figure 14[xvii] illustrates and describes the relationship between form drag and skin friction drag around various three-dimensional objects. Increased turbulence downstream of an object results in more form drag (i.e., more pressure loss). Ma's[xviii] research in Figure 15 showed that the shape of valve's closing object affects the level of flow turbulence. For example, a ball shape has less turbulence than a cone shape, whereas a cone shape has less turbulence than disc shape.

In summary, to lower pressure loss through a valve its hydrodynamic drag must be reduced. Design considerations for reducing valve hydrodynamic flow drag are as follows:

1. streamlining the closing object's shape and guide cage,
2. applying surface treatments,
3. reducing velocities,
4. improving the flow dynamics

Streamlining is designing an object's shape to minimize turbulence and pressure differences. To assess the level of an object's drag, a drag coefficient (Cd) is applied. The drag coefficient is a dimensionless number that quantifies how much drag force an object experiences as fluid flows around it. Figure 16[xix] details drag coefficients for various object shapes. The lower the Cd the more streamlined the object. A round ball is not an overly streamlined shape with a Cd of 0.5, whereas a teardrop shape reduces fluid flow drag by ten (10) times with a Cd of 0.05. High drag with a round ball is because it bluntly disrupts flow and causes turbulence. Figure 17[xx] illustrates how an object's shape impacts the level of turbulence. To minimize pressure loss in a ball and seat style valve, changing the shape of the ball to a tear drop shape could considerably reduce pressure loss. For streamlining of a guide cage, Cutler etal[xxi] in Figure 18 illustrated and demonstrated the pressure loss reduction benefit of smooth and efficient flow conduit (i.e., smooth cross-sectional area transitions) in a travelling valve. Beveling the valve seat and the flow conduit downstream of valve object's stop reduced pressure loss though the valve by more than 50%.

Surface treatments on an object can beneficially reduce hydrodynamic flow drag (skin friction drag component). Rough surfaces generally increase the drag coefficient because they cause earlier turbulence (detached flow) and more energy loss. Polished or smooth surfaces reduce drag, especially on streamlined shapes, by allowing smoother, more attached flow and therefore less pressure loss. Applying a smooth coating such as Diamond Like Carbon (DLC) or applying a surface treatment like Quench Polish Quench (QPQ) with low friction coefficients can reduce pressure loss. Using surface texture features such as fins or vortex generators to reduce turbulence can lower pressure loss. Fua[xxii] in Figure 19 showed that a shark skin's riblets can reduce flow drag by 20%. Chear's[xxiii] research in Figure 20 demonstrated that dimpling of a round object's surface can reduce a flow drag by up to 50% (as in the case of the design of a golf ball).

Reducing velocities across an object can reduce fluid flow drag. Engineering for less fluid velocity can be achieved by increasing the valve guide conduit's cross-sectional area or reducing the cross-sectional area of a valve's closing object (i.e., using a smaller ball such as an "alternative" ball). Figure 21 shows a full flow cage and a cage with an alternate ball. An alternate smaller ball can increase the cross-sectional flow area by 267 percent but takes on ball chatter risk with high valve guide cage to ball clearance. Figure 21 also shows a full flow cage with an API ball and an insert guided cage with an alternate ball.

Improving the flow dynamics through a valve can reduce fluid flow drag and pressure loss. Incorpoating flow guiding features such a longitudinal grooves or fins along the surface of a closing object to stabilize flow and reduce turbulence as fluid passes by. These fins guide flow and prevent turbulence. Coyes[xxiv] in Figure 22 discovered that inducing torsional or vortex flow through a sucker rod pump guide cage beneficially reduces flow turbulence, lowers pressure loss and improves valve reliability. Sucker rod pump valve balls have evolved to where there is significant clearance between the guide cage and the ball for avoiding solids seizing risks. Such clearance increases the risk for damaging ball chatter and increased pressure loss. Vortex flow uniforms the flow velocity profile, reduces turbulence, reduces flow separation, and most importantly self-centers the ball to avoid damaging and pressure loss increasing ball chatter. In addition, self centering of the ball using vortex flows improves valve closing efficacy with helping the ball reliably land on the seat to maintain sealing integrity.

ENGINEERED HYDRODYNAMIC SUCKER ROD PUMP VALVES FOR MINIMIZATION OF PRESSURE LOSS AND HIGH SOLIDS TOLERANCE

Legacy sucker rod pump valve prior art includes full flow cages and insert guide cages, which use a ball and seat. Design evolutions have focused on the valve guide for pressure loss reduction. The ball itself and its shape was fundamentally limiting further reductions in pressure loss.

QSO Inc.[xxv] empirically found for plunger lift bumper spring applications that a barbell shaped valve insert consistently achieved superior closing efficacy over a ball, particularly at high inclinations (i.e., greater than 45 degrees). Figure 23 shows a technology break through and evolution for engineering hydrodynamic sucker rod pump valve inserts. Q2 Artificial Lift Services (Q2 ALS) and QSO Inc. then adapted and design engineered (and patented[xxvi]) a barbell shaped valve insert with a dual Vortex flow insert guide cage for high inclination sucker rod pumping (trade named the HVS™). The barbell's metallurgy is tungsten carbide and weighs significantly more than a regular valve ball, yet its closure efficacy was found to be superior to that of a ball.

It was studied for why improved valve closure efficacy was occurring. Flow loop testing with nano-bubbles confirmed an improvement in valve efficacy but that the barbell had higher pressure loss as compared to a ball – it has higher hydrodynamic drag. See Figure 24 of a barbell valve insert in the flow loop. The results revealed that hydrodynamic drag forces were primarily responsible for valve opening and closing, whereas valve insert weight and inclination had a minor influence. The barbell shaped valve insert beneficially uses hydrodynamic flow drag for opening and closing. The barbell's design contains two opposing teardrop shapes. Flow one direction around a tear dropped shaped object has a low drag coefficient of 0.05, whereas flow in the reverse or opposite direction around that tear drop shaped object has a high drag coefficient (similar to a round ball) of 0.5, ten (10) times higher. A barbell shape with two opposing tear drops has more hydrodynamic flow drag than a ball since it has more surface area (i.e., has greater skin friction drag). It was concluded that the unique barbell shaped valve insert benefits from hydrodynamic flow drag and therefore outperforms a ball shape for valve opening and closing.

There were initial challenges for seat sealing reliability with the barbell's back and forth "shuttle" valve insert sealing on the same face location each closure. A round ball valve insert has infinite sealing surfaces, whereas a non-round valve insert will only have one sealing surface. Metallurgy for the barbell over the past four years has iterated, from field experience to tungsten carbide with 89 Rockwell HCC and a 10 micron sealing face finish. The seat evolved to a proprietary tapered tungsten carbide seat. Coyes etal[xxvii] discuss that over the past few years these sealing features have proven to be reliable.

The barbell valve insert unique design then has allowed new design possibilities for valve pressure loss reduction, for improving valve efficacy and solids tolerance – applying engineered hydrodynamic features. A round ball can seal anywhere on its surface area and therefore needs to be perfectly round, as it can freely three dimensional rotate on its axis. For a ball there will always be a cross-sectional area downstream of its sealing contact area that is inadvertently adding to pressure loss and flow turbulence. Conversely, the vortex barbell shuttles back and forth during closing and opening. It therefore seals on the seat in the same location each time (i.e., seals on the same sealing face area that contacts the seat). It was realized that any cross-sectional area larger than this sealing face area on the barbell and downstream of this sealing face area is unnecessarily adds to pressure loss. It was therefore hypothesized that the most of this cross-sectional area could be removed, such that it would have much less cross-sectional area and pressure loss than a ball.

Engineering valve hydrodynamic drag for improving valve performance, Figures 25 and 26 show some early iterations of engineered hydrodynamic barbell valve inserts. Features developed in this design iteration included:

1. Valve insert object shuttles axially, so only need approximately 50% of an equivalent ball's face for sealing, so remove unnecessary cross-sectional area to reduce pressure loss.
2. Used vortexed large axial channels for flow stabilization and for reducing solids risks (minimal ball guide/cage clearance ball sticking concerns).
3. Valve insert object's outer diameter was designed to eliminate the risk of valve insert chatter; the clearance to the guide cage internal diameter was reduced to 5 thousands of an inch (10 thousands total).
4. Used the smooth tapered seat for reducing turbulence in the guide cage.
5. Used the existing and proven vortex barbell guide with dual inserts.

For controlling solids risks, legacy ball and seat designs and industry practices have empirically settled on an API ball's clearance to the guide cage needing to be in the range of 15 to 20 thousands of an inch per side (30 to 40 thousands of an inch total). It is the authors' understanding that this clearance specification has been adopted for reducing the risk of ball sticking or jamming due to solids. Unfortunately, to address solids risks, this clearance trades off risk of reduced reliability – allowing lateral ball movement and therefore ball chatter, especially if flows are turbulent. With large axial flow channels carved into a barbell allowed for clearance to the guide cage to be minimized, avoid insert chatter. It was found that the insert axially flow channels control the risk of solids ball sticking. The hydrodynamic valves flow channels are massive, such that the fin clearance to the guide change has been reduced 7.5 thousands of an inch per side (15 thousands of an inch total), therefore minimizing chatter risks.

Application of three-dimensional (3D) metal printing (additive manufacturing) was then employed to avoid machining limitations and to allows rapid and limitless design iterations. Figure 27 shows transformative iterations where the valve insert shape was hydrodynamically engineered for minimal pressure loss (minimal flow drag) in the valve's opening flow direction, but it was then engineered with greater flow drag in the closing reverse flow direction. The open flow direction pressure loss target was as close to an open cage (with not valve insert inside) as possible. Certainly, the goal was to be less pressure loss than a ball. Higher engineered hydrodynamic drag in the reverse flow direction assists valve closure. Very large axial fins improved solids conveyance and avoids valve sticking / erosion from solids. Post 3D metal printing machining of the sealing face was required to achieve the API vacuum test.

Key embodiments for final new design iteration included:

1. Engineered for absolute minimum drag and pressure loss in the valve opening flow direction.
2. Engineered higher drag in the valve's closure flow direction.
3. Has high open and closure efficacy at any inclination.
4. Significantly less mass and valve insert travel distance, reducing seat impact forces and improving valve opening/closing actuation times.
5. The valve insert has massive flow channels that act as an integrated guide cage for increasing the cross-sectional area to flow. A valve guide cage's flowby cross-sectional area is limited by the outside diameter of the cage for the pump size.

Figures 28 and 29 show the more recent design embodiment and the final version to be used in field trails. The valve was traded named the HammerHead™. This design was heavily influenced by flow loop testing and subsequent streamlining optimization. A football shaped nose was adopted from submarine research[xxviii]. Figure 29 shows a Diamond Like Carbon (DLC) coated version that was 3D metal printed with 316 stainless steel. Figure 30 illustrates the valve insert inside the dual vortex barbell guide cage and bevelled seat.

FLOW LOOP TESTING

Multiple iterations of engineered hydrodynamic sucker rod pump valves inserts were flow loop tested with water. Figure 31 shows the flow loop test set up. The flow was fully instrumented with high resolution and frequency digital recording capabilities. It also was configured with a four-way valve that allow rapid flow reversal for valve closing and opening efficacy testing.

Figure 32 shows the pressure loss flow results at 1,300 barrels per day water of various 1.75 inch valve insert shapes and a scenario for pressure loss with no valve insert in the cage. The final version was able to reduce pressure loss by approximately 50 percent.

To achieve the target and goa pressure loss of a guide cage with no insert, the guide design will need to change, which is currently a subject of ongoing research.

Figure 33 shows valve closure efficacy testing results of the final design. Valve closure efficacy was found to be high.

<u>IMPROVING THE DESIGN OF A PRESSURE BALANCING PUMP</u>

William's[xxix] in Figure 34 shows the legacy pressure balancing pump with a tapered top section (upper extension) of the barrel. The taper gradually increases the internal diameter of the barrel towards the top. When a plunger enters this tapered region, fluid slippage past the plunger increases rapidly, effectively pressure balancing the travelling valve and pump's chamber prior to the top of the pump stroke. He discusses the design of the tapered barrel pressure balancing pump and related field results.

Pressure balancing of a sucker pump such that pressure is equalized across the travelling valve before the downstroke has many benefits and advantages but has also faced challenges.

Advantages:

- Avoids downstroke compressional rod loadings (higher min loads)
- Can achieve very low PIP's
- Avoids fluid pound and improves pump efficiceny by controlling gas/foam interference

Challenges and limitations:

- Pump's plunger space-out needs to be precise, which is very challenging is gassy-sluggy-foaming deep horizontal wells and when composite fiberglass sucker rods are used
- Low solids tolerance and prone to fluid erosion.

It was hypothesized that instead of using a tapered barrel top section, that use of an engineered rifled channel in this same top barrel would resolve limitations. Figure 35 shows and engineering drawing of a rifled top barrel.

In theory, the main functions and advantages of a rifled channel in a top section of a sucker rod pump barrel would be to:

1. Partially but mostly pressure balance the pump's travelling valve regardless of plunger position in the rifled barrel section, resolving the issue of precise pump space outs. It is believed that full pressure balancing, or equalization is unnecessary and that still allow the pump's plunger to build pump chamber pressure on the downstroke will manage fluid erosion risks and improve reliability
2. The rifled channel would be sized for fluid slippage to efficiently transport or convey solids in the direction of the pressure differential, minimizing the risk of pump plunger solids sticking/seizing.

At the time of this paper’s writing, a pressure balancing pump with a rifle channeled top barrel has not been field trialed.

NEW SUCKER ROD PUMP APPLICATIONS

Five scenarios were studied where there would be advantages for a sucker rod pump that pressure balances, has minimal pressure loss across the standing valve and with high solids tolerance:

1. Expanding of high rate pumping limits. Less pressure loss into a pump chamber means the pump will be more efficient at high peak plunger velocities such that average pump rate can therefore increase.
2. Improved stripper type vertical well. Less pressure loss into the pump means a lower pump intake pressure is required to fill the pump and avoidance of fluid pounding for increased reliability. The opportunity for this application is increased drawdown and more production.
3. Improved solids tolerance. Excessive solids in the produced fluids that result in a high stuck pump failure frequency. Continuous solids conveyance in the rifled channel would prevent plunger sticking events.
4. Improved handling of excessively foamy produced fluids. High compressible foamy fluids would be more efficiency pumped since the pressure is balanced at the commencement of the downstroke.
5. Improved extended dip tube systems performance. Any reduction in pressure loss will benefit horizontal wells where the pump is placed at the kick off point (KOP) and the separator is placed deeper into the curve at 45 degrees inclination using an extend dip tubing configuration. See wellbore BHA illustrations in Figure 36. An extended dip tube has an inherent increased system pressure loss due to the extended dip tube (typically comprising of 370 feet of 2-3/8” EUE tubing. There will always be a level of gas entering the pump due to the system’s pressure loss, therefore pressure balancing would increase the pump’s efficiency. The opportunity for this application is improved pump fillage, increased drawdown (lower PIP), more production and improved failure frequency with solids risks.

Sucker rod pumping at very low PIP’s is challenging. Figure 37 shows a graph of gas formation volume factor. Pressure loss within a rod pump system causes both gas liberating from solution (dissolved gas in the oil) and simultaneous gas volumetric expansion. Solution gas entering the system is relatively linear, while gas expansion has a more exponential increase with loss in pressure. PIP’s below 200 psi will have to contend with very high gas formation volume factors, as gas expansion becomes highly exponential below 200 psi. Any pressure loss in the rod pumping system will result in a large gas volume fraction in the pump’s chamber (i.e., excessive pump gas interference).

As absolute operating pressure decreases, so does system efficiency related to pressure drop. System inefficiencies are magnified in extended dip tube BHA designs. Figure 38 shows a modelled scenario of an extended dip tube system adding to pressure losses prior to the pump’s chamber in the order of 130 psi to 170 psi. The amount of gas liberated from the oil and the amount of volumetric gas expansion from the added pressure loss will be observed as an increase in pump gas inference (i.e., reduced pump fillage). Figure 39 shows the effective pump fillage at 250 psi PIP will be less 50 percent. It also shows that a pup placed in the vertical with an extended dip system will remain reasonably efficient at low PIP's It is therefore concluded that extended dip tube system will face a risk of under performance at low PIP's.

Figure 40 shows that at lower PIP's, a reduction in pressure drop across the sucker rod pump's standing valve can improve system efficiency by an absolute 5% in vertical inclination placed pumps. For extended diptube designs, reducing pressure drop across the standing valve can result in greater than an absolute 10% improvement in system efficiency.

CASE STUDIES

Case Study 1 – Figures 41 and 42 show field recorded sucker rod pumping parameters. Figure 41 shows an extended dip tube BHA configuration in a horizontal well with the pump placed in the vertical section of the well above the curve. It has continuous pump gas interference with pump fillage averaging 80 percent.

Figure 42 shows a well with the pump and gas separator placed at 85 degrees inclination using barbell standing and travelling valves. With low pressure loss into the pump and with high gas separation efficiency with a separator placed at high inclination, pump gas interference is low with pump fillage averaging over 95 percent.

Field implementations have commenced effective April 2025 for the engineered hydrodynamic valves. Production performance results and statistics are being compiled will be updated in future publications.

CONCLUSION

A patent pending hydrodynamic sucker rod pump valve has been develop for minimization of pressure loss and with high solids tolerance. Reduced pressure loss into sucker pump will benefit sucker rod pumping in low PIP conditions and will allow for increased sucker pumping rate capacity.

A patent pending rifled top section for a sucker rod pump barrel has been developed to allow beneficial pressure balancing of a pump's chamber prior to the downstroke and for greater solids tolerance.

FIGURES

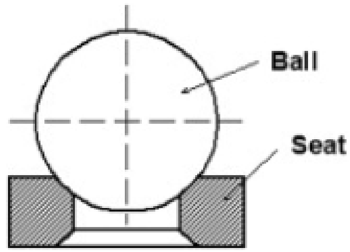


FIGURE 1 – SUCKER ROD PUMP API BALL AND SEAT VALVE, CIRCA 1938

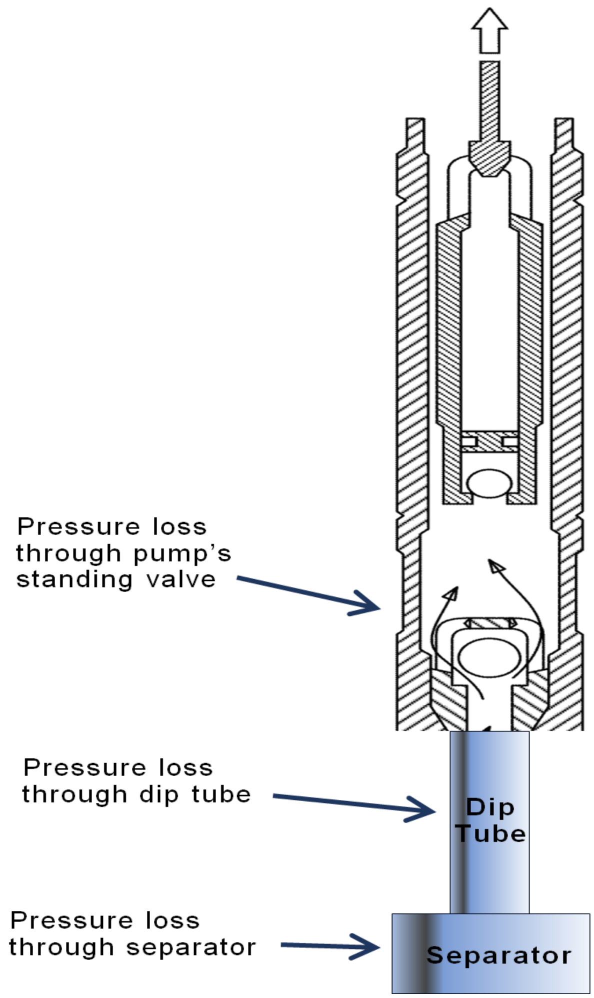
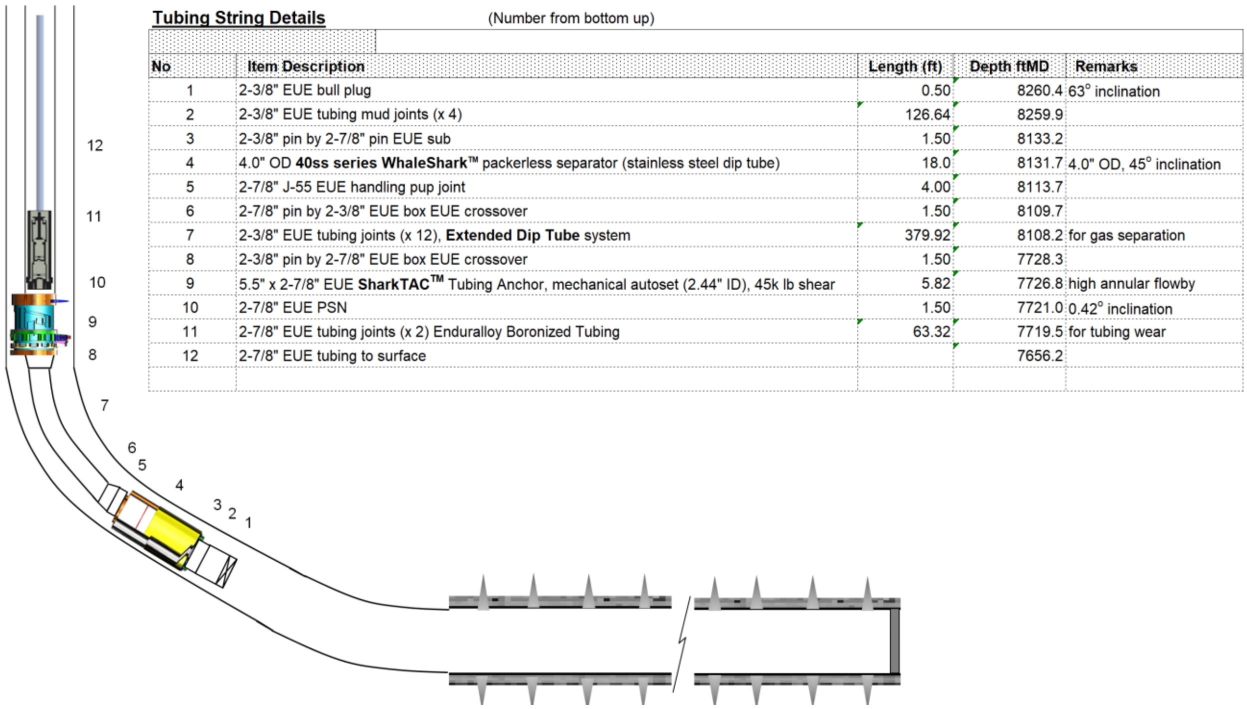


FIGURE 2 – BHA SYSTEM PRESSURE LOSSES BEFORE ROD PUMP'S CHAMBER



Tubing String Details (Number from bottom up)

No	Item Description	Length (ft)	Depth ftMD	Remarks
1	2-3/8" EUE bull plug	0.50	8260.4	63° inclination
2	2-3/8" EUE tubing mud joints (x 4)	126.64	8259.9	
3	2-3/8" pin by 2-7/8" pin EUE sub	1.50	8133.2	
4	4.0" OD **40ss series WhaleShark™** packerless separator (stainless steel dip tube)	18.0	8131.7	4.0" OD, 45° inclination
5	2-7/8" J-55 EUE handling pup joint	4.00	8113.7	
6	2-7/8" pin by 2-3/8" EUE box EUE crossover	1.50	8109.7	
7	2-3/8" EUE tubing joints (x 12), **Extended Dip Tube** system	379.92	8108.2	for gas separation
8	2-3/8" pin by 2-7/8" EUE box EUE crossover	1.50	7728.3	
9	5.5" x 2-7/8" EUE **SharkTAC™** Tubing Anchor, mechanical autoset (2.44" ID), 45k lb shear	5.82	7726.8	high annular flowby
10	2-7/8" EUE PSN	1.50	7721.0	0.42° inclination
11	2-7/8" EUE tubing joints (x 2) Enduralloy Boronized Tubing	63.32	7719.5	for tubing wear
12	2-7/8" EUE tubing to surface		7656.2	

FIGURE 3 – EXTENDED DIP TUBE BHA FOR HORIZONTAL WELLS

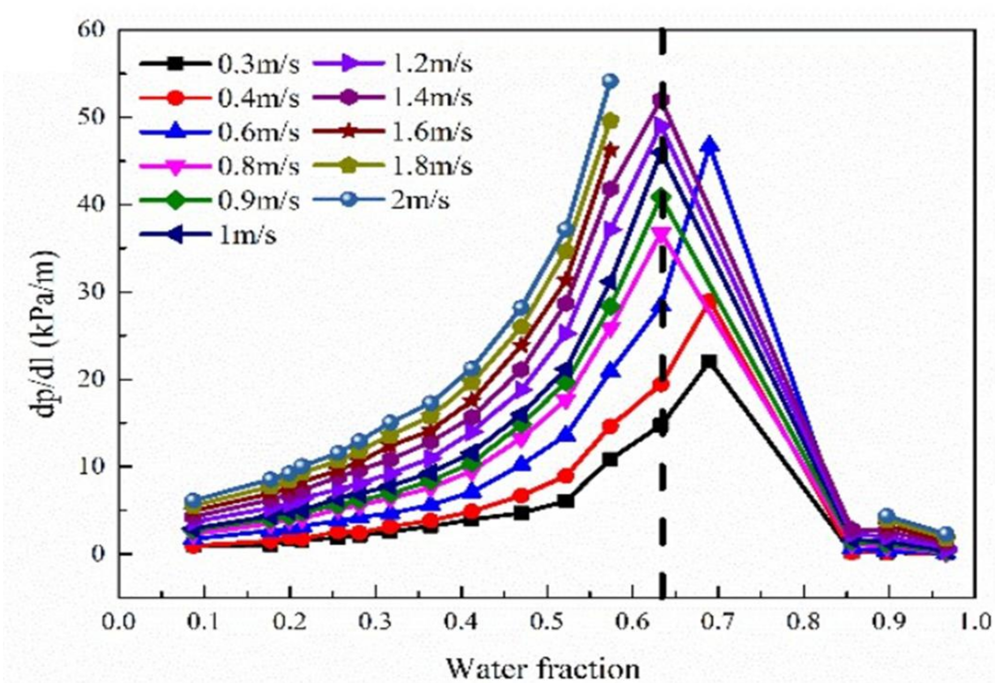


FIGURE 4 – APPARENT FLUID VISCOSITY AS A FUNCTION OF OIL AND WATER RATIO CAN INCREASE PRESSURE LOSS BY AN ORDER OF 10 TIMES OVER WATER OR OIL ALONE

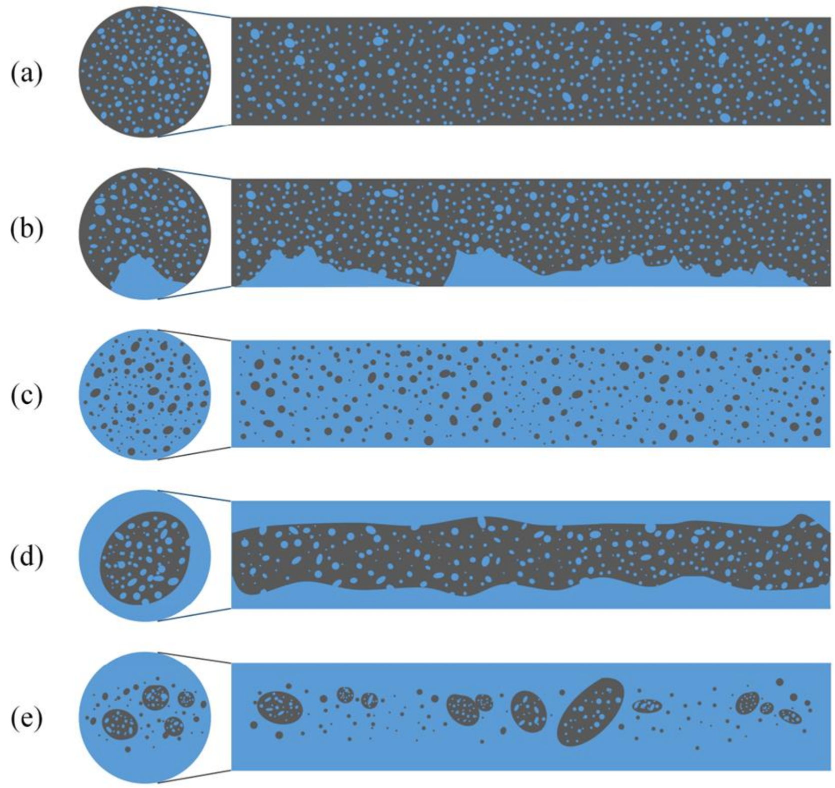


FIGURE 5 – OIL AND WATER MIXTURE FLOW PATTERNS AND DISPERSION AFFECT APPARENT VISCOSITY

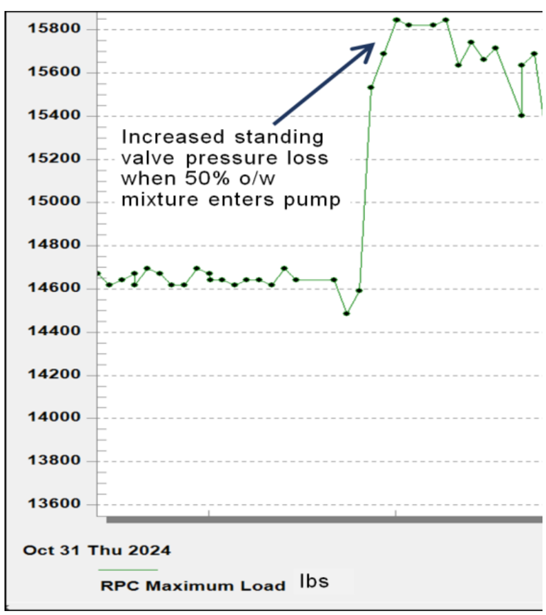
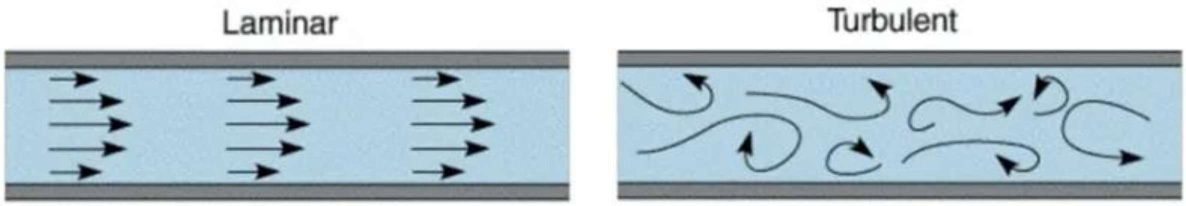


FIGURE 6 – FIELD RECORDED EXAMPLE ON INCREASE PEAK POLISH ROD LOAD FOR WHEN AN OIL AND WATER MIXTURE ENTERS A PUMP



Reynolds Number	Condition of flow
Re < 2000	Laminar
2000 < Re < 4000	Transitional
Re > 4000	Turbulent

🧪 Standard Reynolds Number Formula (Newtonian Fluids):

$$Re = \frac{\rho v D}{\mu}$$

Where:

- ρ = fluid density (kg/m³)
- v = velocity (m/s)
- D = characteristic length (e.g., pipe diameter)
- μ = **dynamic viscosity** (Pa·s)

🧬 For Non-Newtonian or Multiphase Mixtures:

We use the **apparent viscosity** μ_{app} in place of μ:

$$Re = \frac{\rho v D}{\mu_{\text{app}}}$$

FIGURE 7 – LAMINAR VERSUS TURBULENT FLOW AND THE REYNOLDS NUMBER CALCULATION

From Newton's 2nd law for fluids:

$$\Delta P = \rho \cdot a \cdot L$$

Where:

- ΔP = pressure loss
- ρ = fluid density
- a = acceleration
- L = length of pipe where acceleration occurs

So even **without friction**, just **accelerating fluid** in a pipe causes a pressure drop due to the energy needed to increase the velocity.

FIGURE 8 – EQUATION FOR FLUID ACCELERATION PRESSURE LOSS

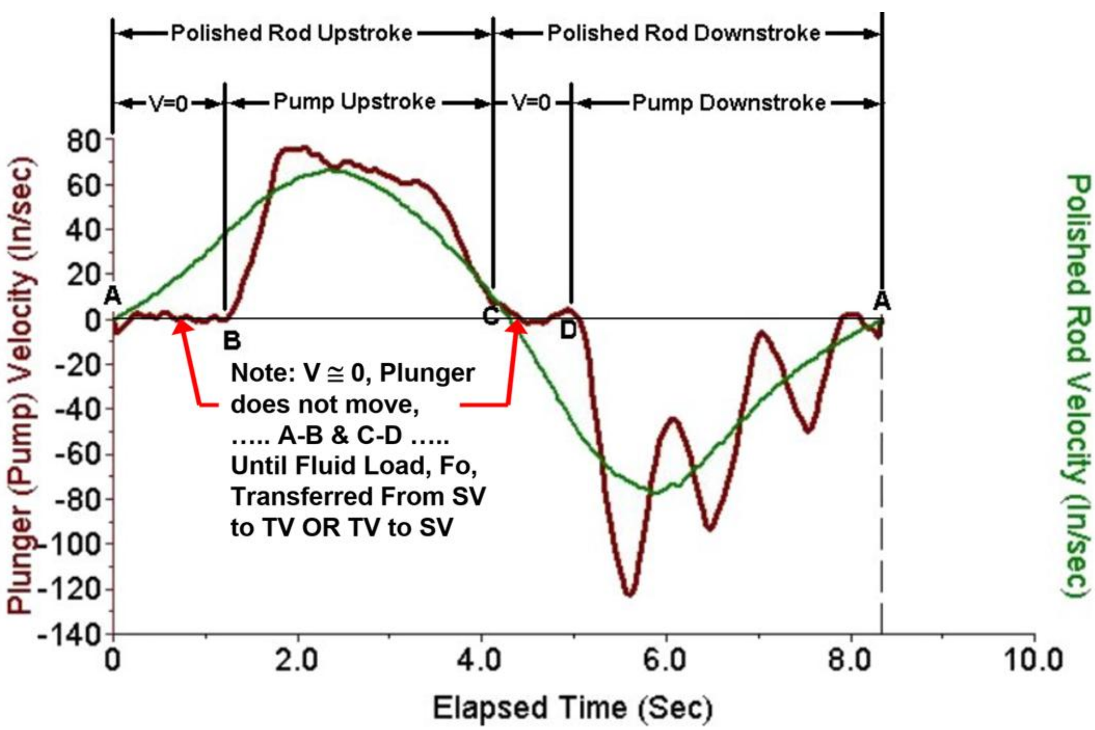


FIGURE 9 – ROWLAND EXPLAINS HOW A PUMP'S PLUNGER RAPIDLY ACCELERATES DURING A SUCKER ROD PUMP'S UPSTROKE (ZERO TO 80 INCHES PER SECOND IN ONE SECOND)

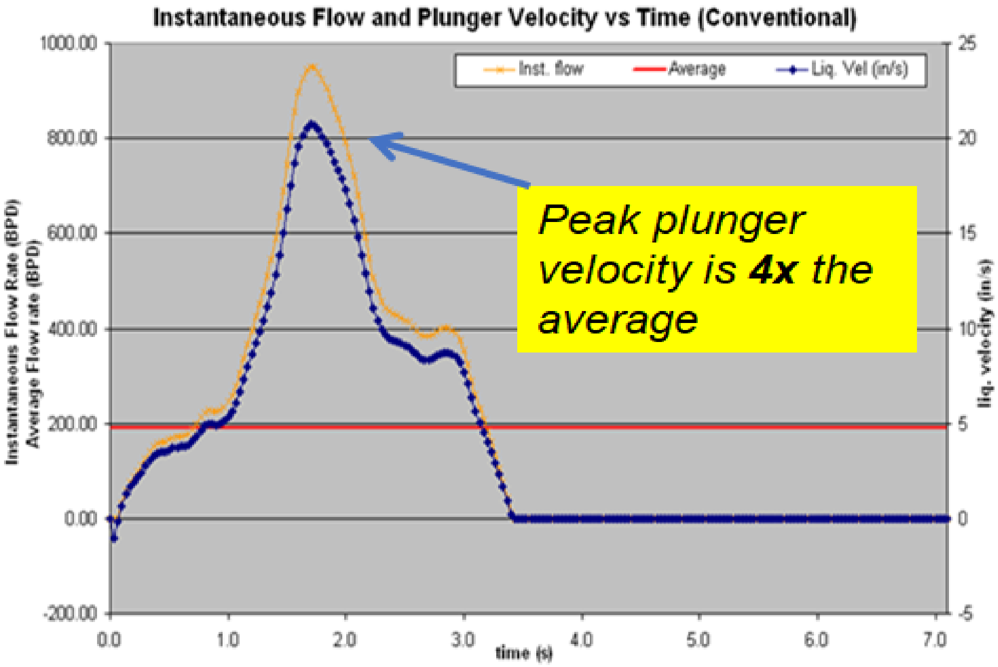


FIGURE 10 – INSTANTANEOUS LIQUID FLOW RATE INTO A PUMP DUE TO UPSTROKE PLUNGER VELOCITY

> What is the pressure required for when 1 cP water in a 370 foot long 1.9" ID pipe is accelerated from zero feet per second to 3 feet per second in 1 second

Let's calculate the **total pressure required** to accelerate **1 cP water** (i.e., close to normal water viscosity) from **0 to 3 ft/s** in **1 second**, through a **370-ft-long**, **1.9-inch ID pipe**.

We'll again consider:

1. Pressure to **accelerate the fluid**
2. **Frictional losses** (Darcy-Weisbach)

Given:

- Pipe length: $L = 370 \text{ ft}$
- Pipe ID: 1.9 in = 0.1583 ft = 0.04827 m
- Final velocity: $v = 3 \text{ ft/s}$
- Acceleration time: 1 second → $a = 3 \text{ ft/s}^2$
- Average velocity: $\frac{0+3}{2} = 1.5 \text{ ft/s}$
- Viscosity: $\mu = 1 \text{ cP} = 0.001 \text{ Pa}$\cdotps
- Density: $\rho = 62.4 \text{ lbm/ft}^3 = 1000 \text{ kg/m}^3$

✅ Step 1: **Pressure for Acceleration**

$$P_{\text{acc}} = \rho \cdot L \cdot a$$

Convert density to slugs:

$$\rho = \frac{62.4}{32.174} = 1.94 \text{ slugs/ft}^3$$

$$P_{\text{acc}} = 1.94 \cdot 370 \cdot 3 = 2151 \text{ lb/ft}^2$$

Convert to psi:

$$\frac{2151}{144} \approx \boxed{14.94 \text{ psi}}$$

✅ Step 2: **Frictional Losses**

Use average velocity = 1.5 ft/s = 0.457 m/s

Reynolds number:

$$Re = \frac{\rho v D}{\mu} = \frac{1000 \cdot 0.457 \cdot 0.04827}{0.001} \approx 22,000$$

→ Turbulent flow

Darcy friction factor (Blasius):

$$f \approx 0.316 \cdot Re^{-0.25} \approx 0.316 \cdot (22000)^{-0.25} \approx 0.027$$

Darcy-Weisbach equation:

$$\Delta P_f = f \cdot \frac{L}{D} \cdot \frac{\rho v^2}{2}$$

- $L = 112.78 \text{ m}$
- $D = 0.04827 \text{ m}$
- $v = 0.457 \text{ m/s}$

$$\Delta P_f = 0.027 \cdot \frac{112.78}{0.04827} \cdot \frac{1000 \cdot (0.457)^2}{2}$$

$$= 0.027 \cdot 2337.2 \cdot 104.5 \approx 6607 \text{ Pa} = \boxed{0.96 \text{ psi}}$$

✅ Total Pressure Required:

$$P_{\text{total}} = 14.94 + 0.96 = \boxed{15.9 \text{ psi}}$$

✅ Final Answer:

Approximately 15.9 psi is required to accelerate **1 cP water** from **0 to 3 ft/s** in **1 second** through a **370-ft-long**, **1.9-inch ID pipe**.

Let me know if you want the same calc for a longer ramp time or higher final velocity!

FIGURE 11 – EXAMPLE CHATGPT 4o CALCULATIONS FOR PRESSURE LOSS IN AN EXTENDED DIP TUBE WITH 1 CP WATER

✅ **Final Pressure Required:**

$$P_{\text{total}} = 13.5 \text{ (acceleration)} + 35.7 \text{ (friction)} = \boxed{49.2 \text{ psi}}$$

FIGURE 12 – EXAMPLE CHATGPT 4O CALCULATIONS FOR PRESSURE LOSS IN AN EXTENDED DIP TUBE WITH 350 CP WATER/OIL MIXTURE

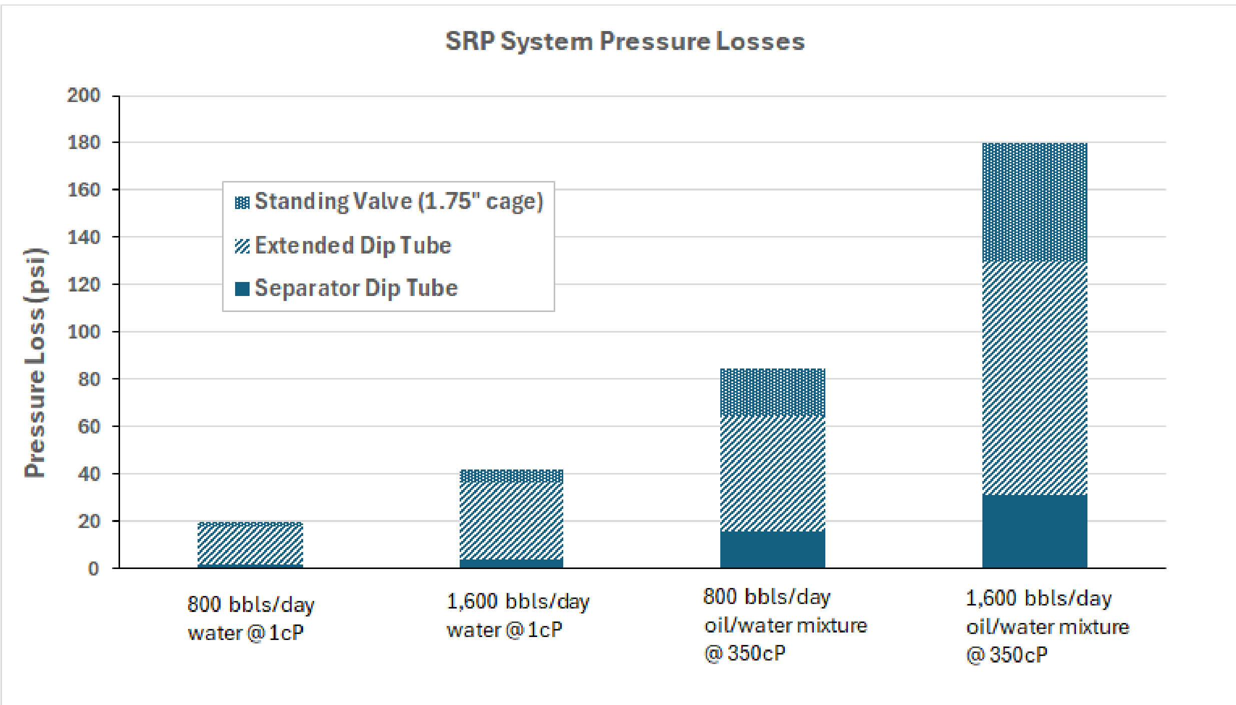


FIGURE 13 – PRESSURE LOSS PRIOR TO A ROD PUMP'S CHAMBER (DOWNSTREAM STANDING VALVE), WATER VERSUS OIL/WATER MIXTURE

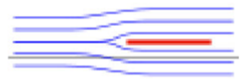
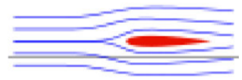
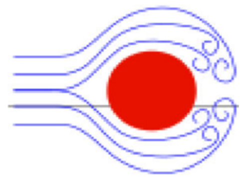
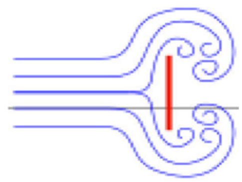
Shape and flow	Form Drag	Skin friction
	≈0%	≈100%
	≈10%	≈90%
	≈90%	≈10%
	≈100%	≈0%

FIGURE 14 – FLUID FLOW DRAG AROUND VARIOUS THREE-DIMENSIONAL OBJECT SHAPES

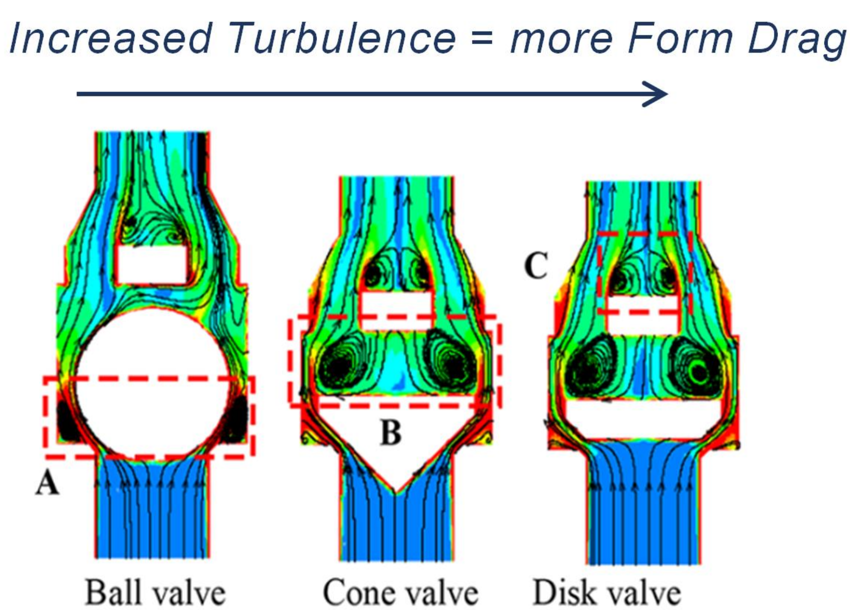


FIGURE 15 – INCREASED TURBULENCE INCREASES FORM DRAG IN VARIOUS VALVE CLOSING OBJECT SHAPES

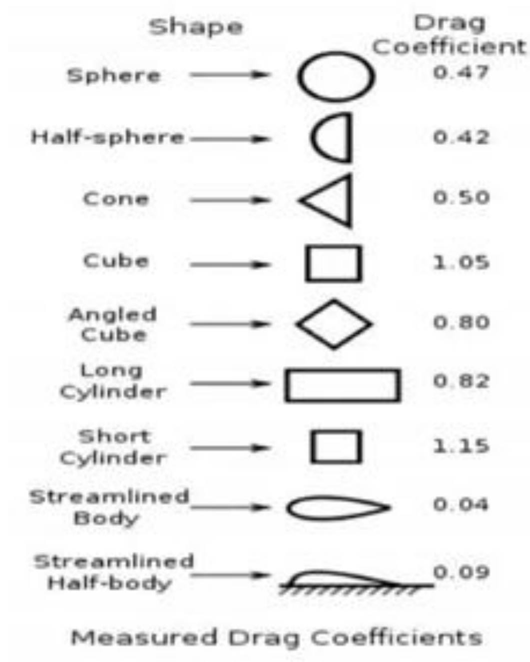


FIGURE 16 – DRAG COEFFICIENTS OF VARIOUS OBJECT SHAPES

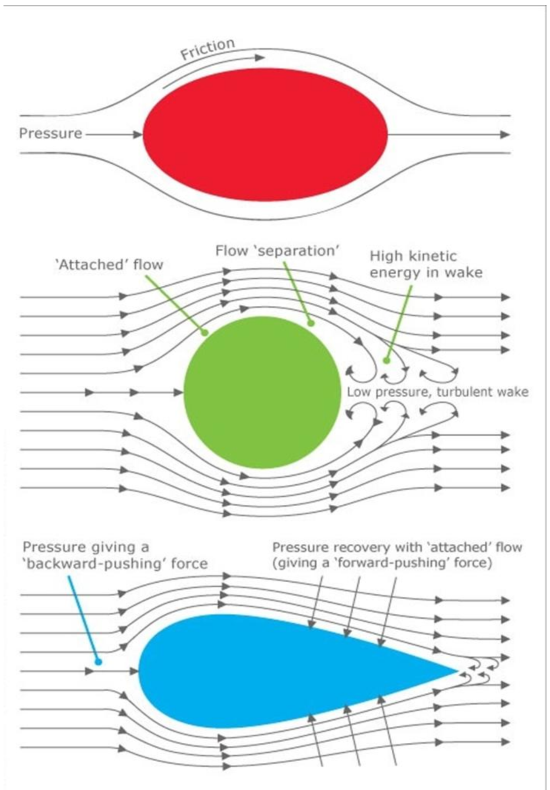


FIGURE 17 – STREAMLINING AN OBJECTS SHAPE REDUCES THE LEVEL OF TURBULENCE AND FLOW DRAG AND LOWERS PRESSURE LOSS

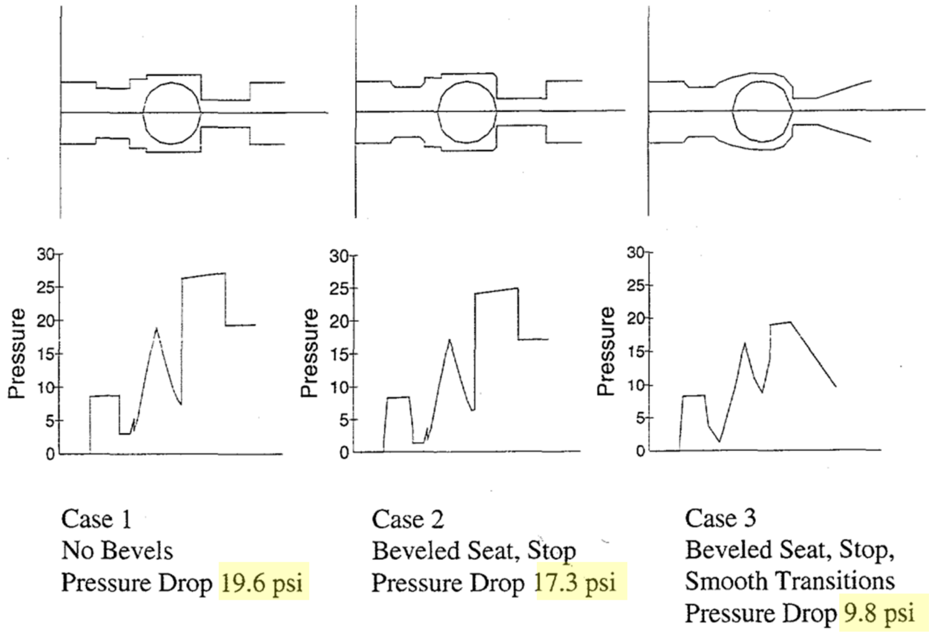


FIGURE 18 – FLOW PATH CONDUIT CONFIGURATION AND SHAPE IMPACT PRESSURE LOSS THROUGH A TRAVELLING VALVE

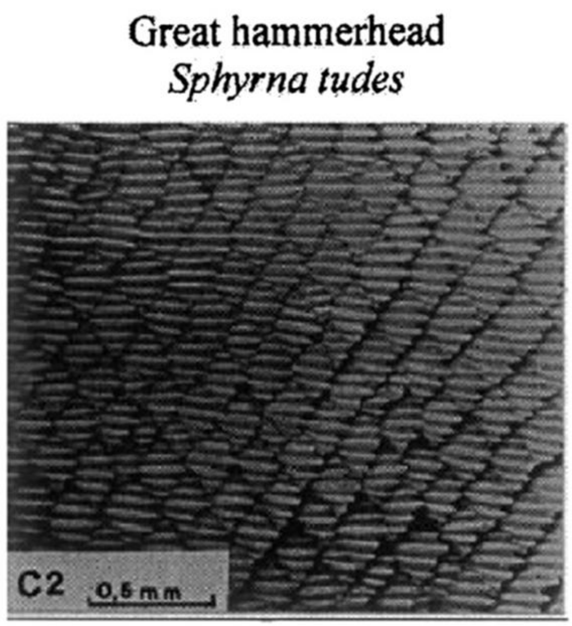


FIGURE 19 – SURFACE TREATMENTS SUCH AS RIBLETS ON A SHARK'S SKIN CAN REDUCE SKIN FRICTION FLOW DRAG

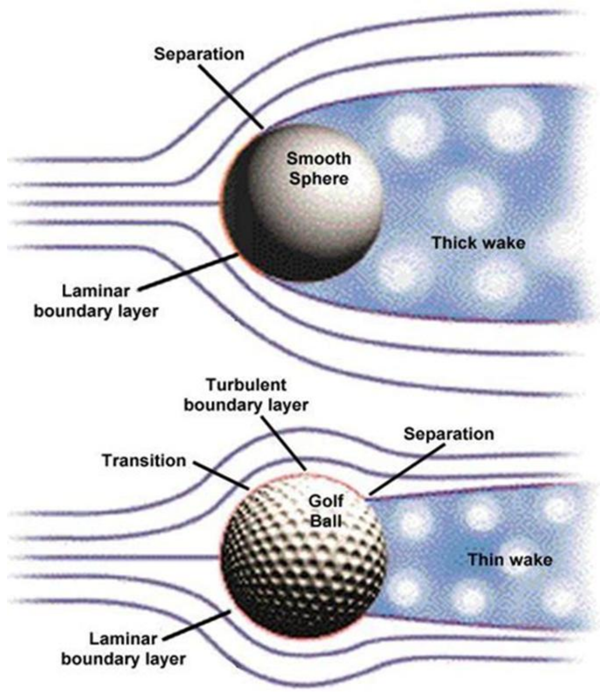


FIGURE 20 – A DIMPLED SURFACE ON A GOLF BALL CAN REDUCE FLOW DRAG BY UP TO 50%

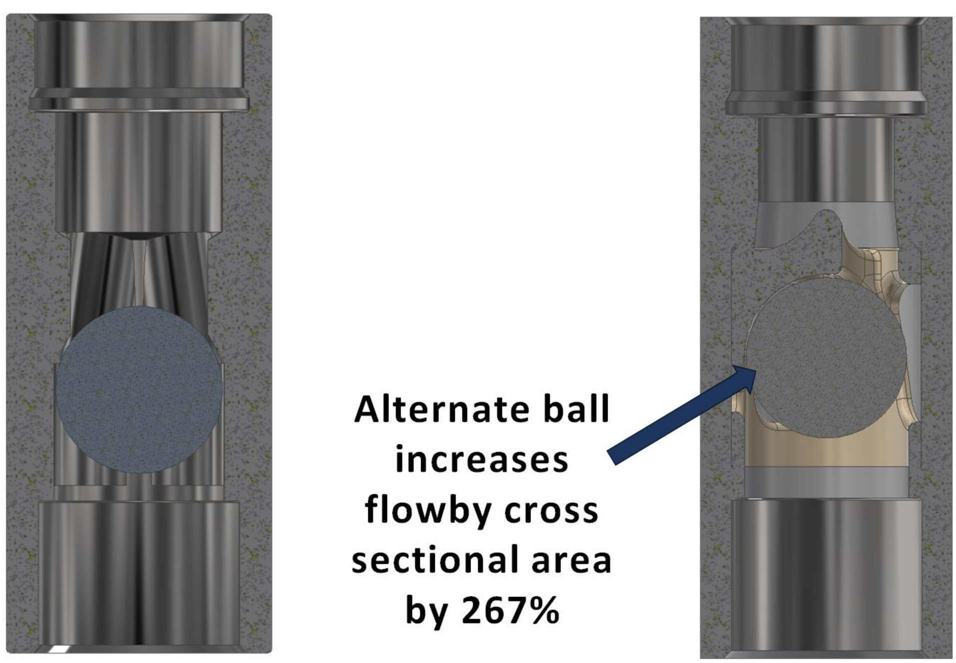


FIGURE 21 – LEGACY FULL FLOW CAGE WITH API BALL AND INSERT GUIDED CAGE WITH ALTERNATE BALL

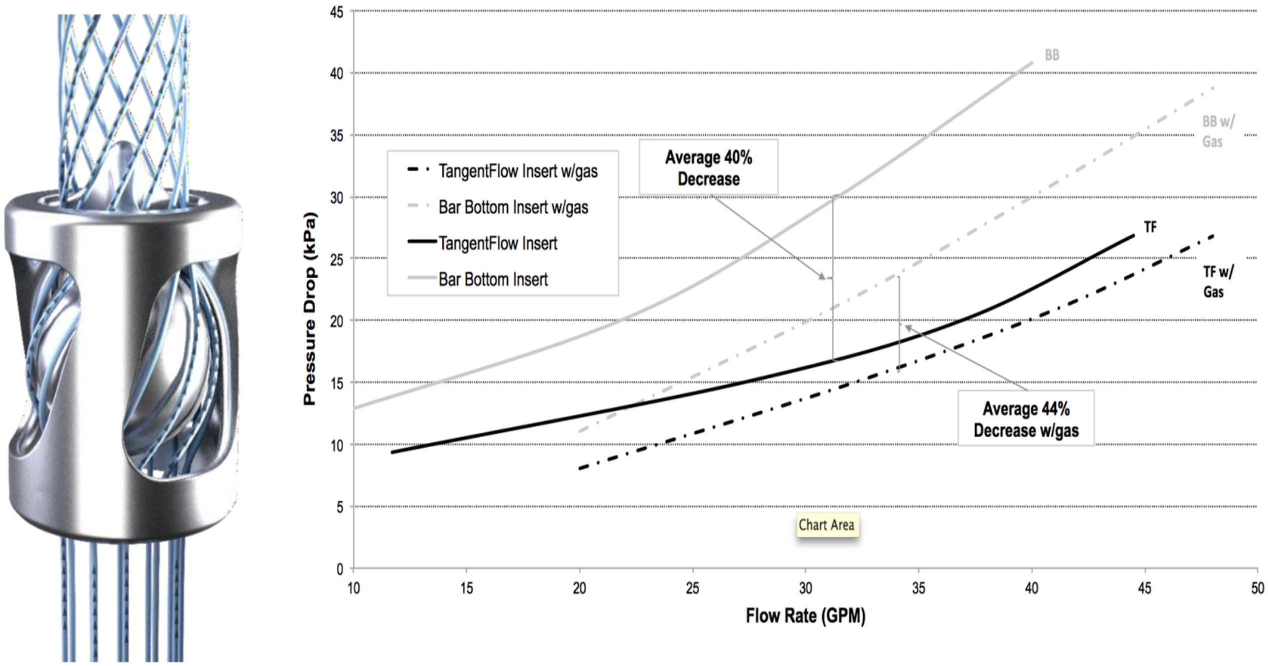


FIGURE 22 – VORTEX FLOW THROUGH A SUCKER ROD PUMP VALVE REDUCES PRESSURE LOSS AND IMPROVES RELIABILITY

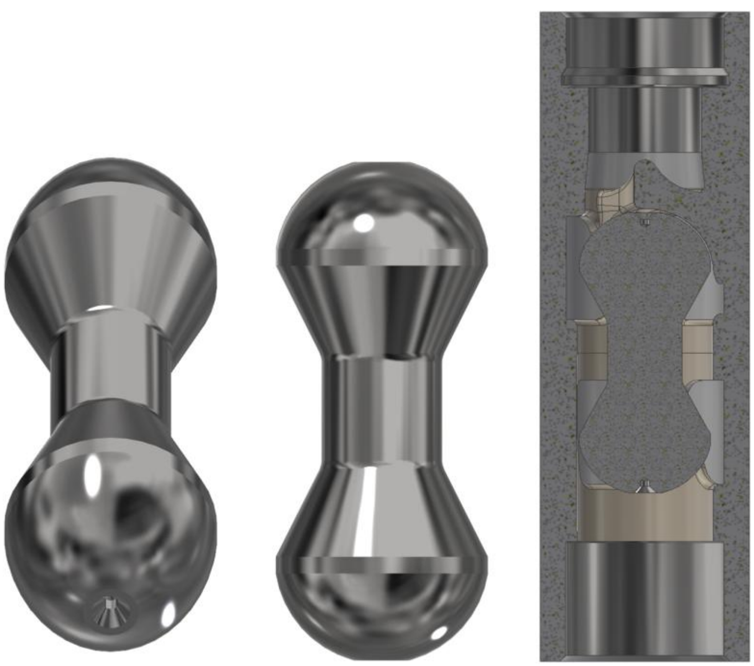


FIGURE 23 – BARBELL VALVE (Q2 ALS HVS^{TM}) INSERT DESIGN AND DUAL INSERT GUIDED CAGE FOR HIGH INCLINATION PUMP PLACEMENTS

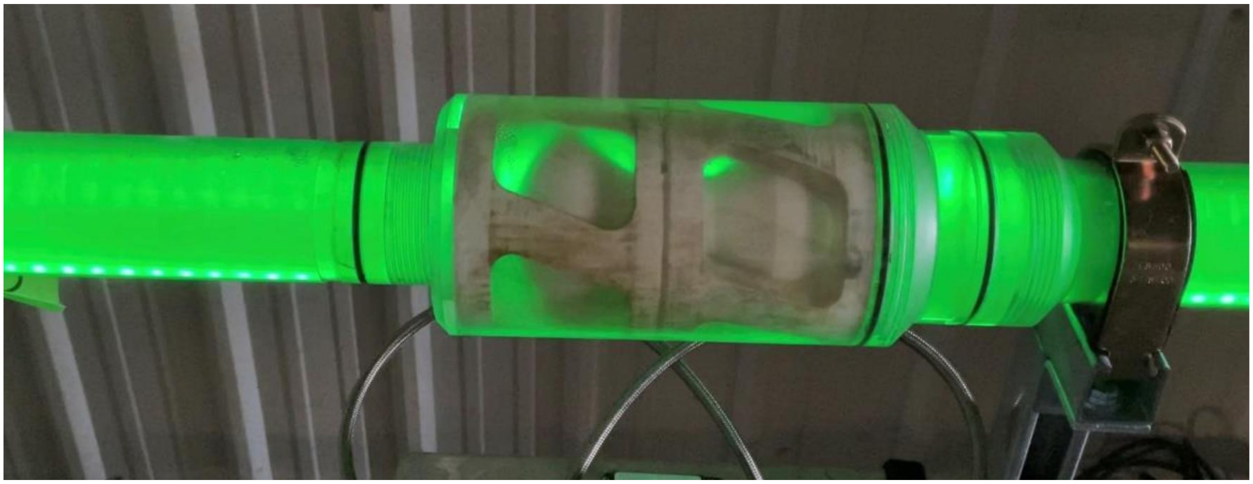


FIGURE 24 – BARBELL FLOW LOOP TESTING PROVED VALVE FUNCTIONING BENEFICIALLY USES HYDRODYNAMIC FLOW DRAG

FIGURE 25 – VORTEX BARBELL ITERATIONS WITH HYDRODYNAMIC ENGINEERED PRESSURE LOSS REDUCTION FEATURES

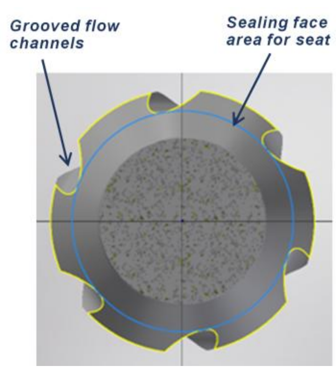


FIGURE 26 – VORTEX BARBELL FRONT FACE WITH HYDRODYNAMIC ENGINEERED PRESSURE LOSS REDUCTION FEATURES

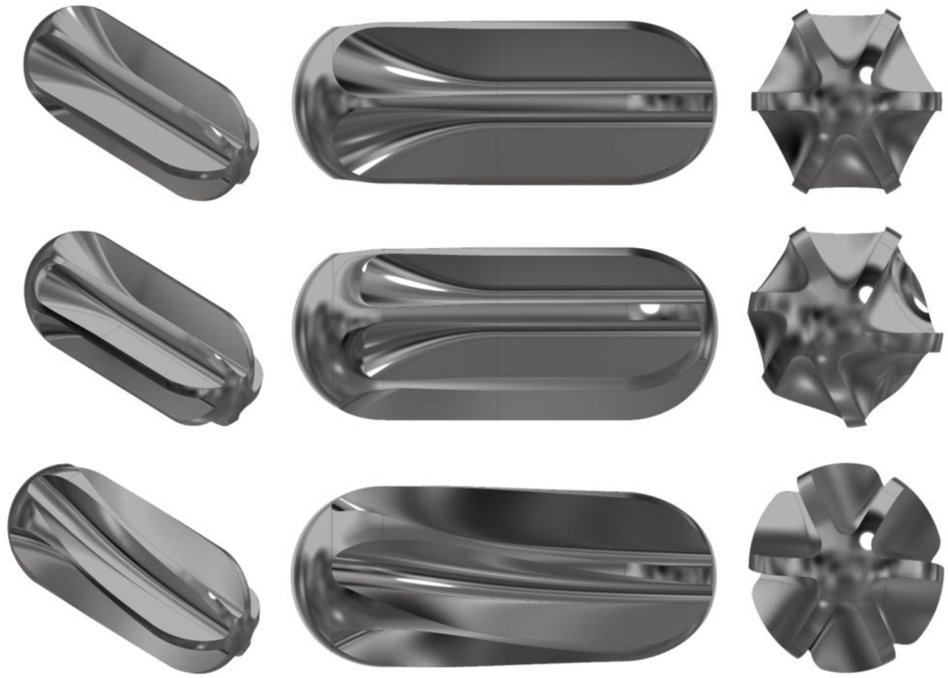


FIGURE 27 – ENGINEERED HYDRODYNAMIC VALVE INSERT DESIGN ITERATIONS USING 3D METAL PRINTING

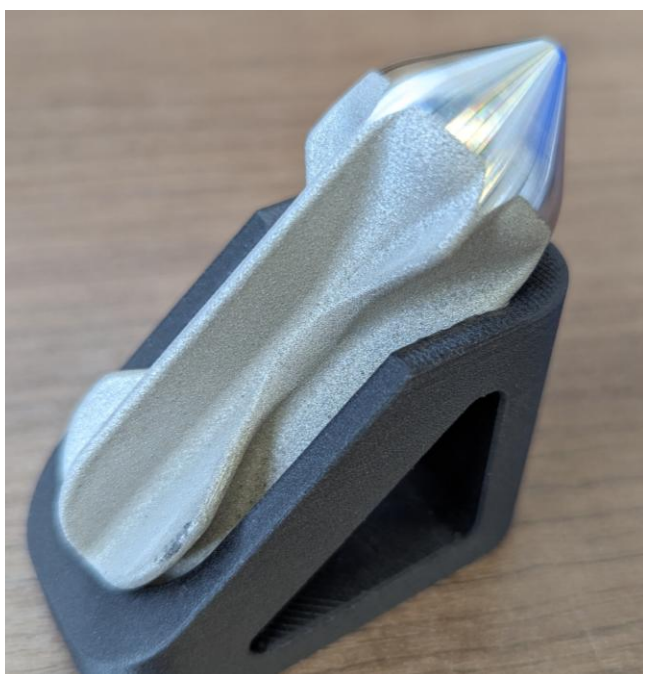


FIGURE 28 – FINAL VERSION FULLY ENGINEERED HYDRODYNAMIC SUCKER ROD PUMP VALVE INSERT FOR MINIMIZING PRESSURE LOSS AND MAXIMIZING SOLIDS TOLERANCE

FIGURE 29 – DIAMOND LIKE CARBON (DLC) COATING APPLIED TO PROTOTYPE 3D PRINTED HYDRODYNAMIC VALVE FOR ABRASION/WEAR RESISTANCE

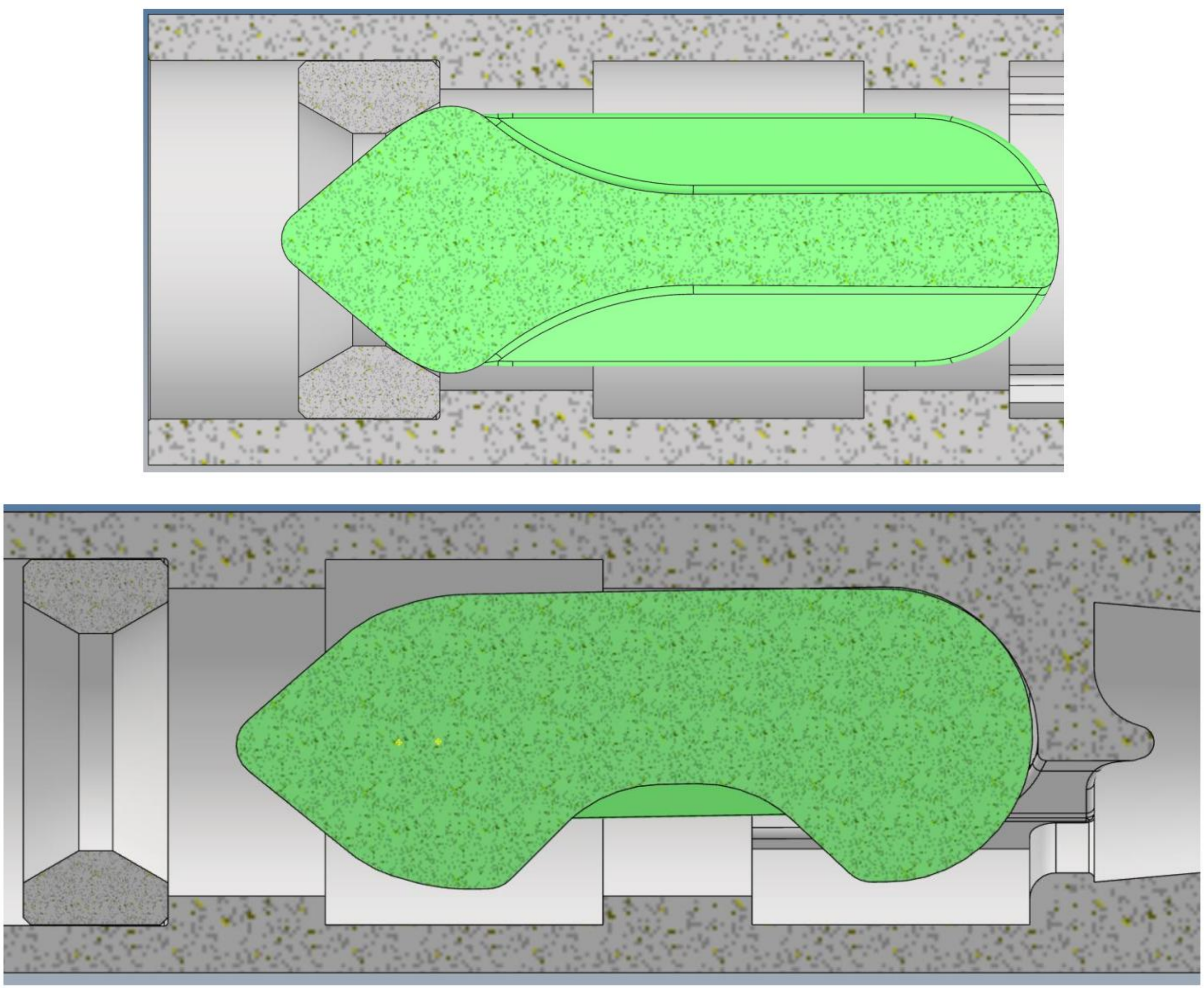


FIGURE 30 – BEVELLED SEAT DESIGN AND HAMMERHEAD VALVE ON AN OFF SEAT IN Q2 ALS HVS DOUBLE INSERT CAGE

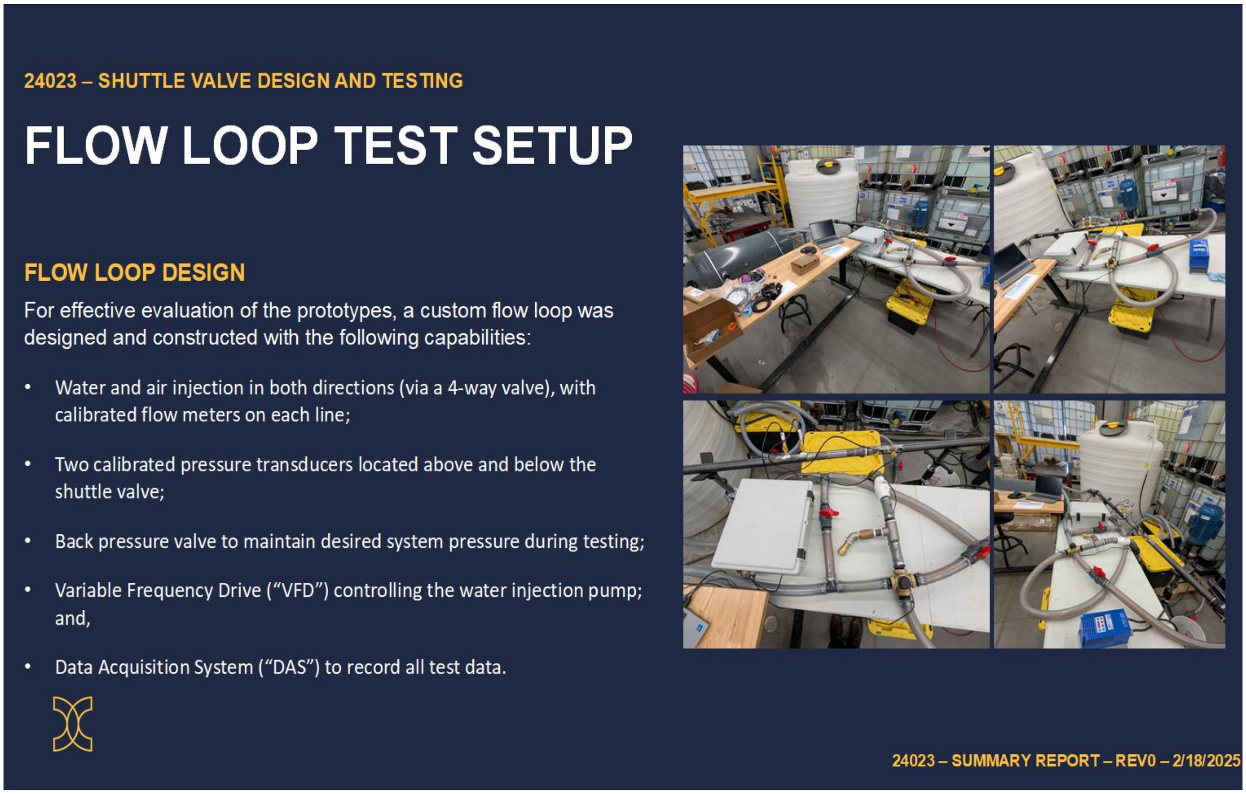


FIGURE 31 – FLOW LOOP TEST SET-UP

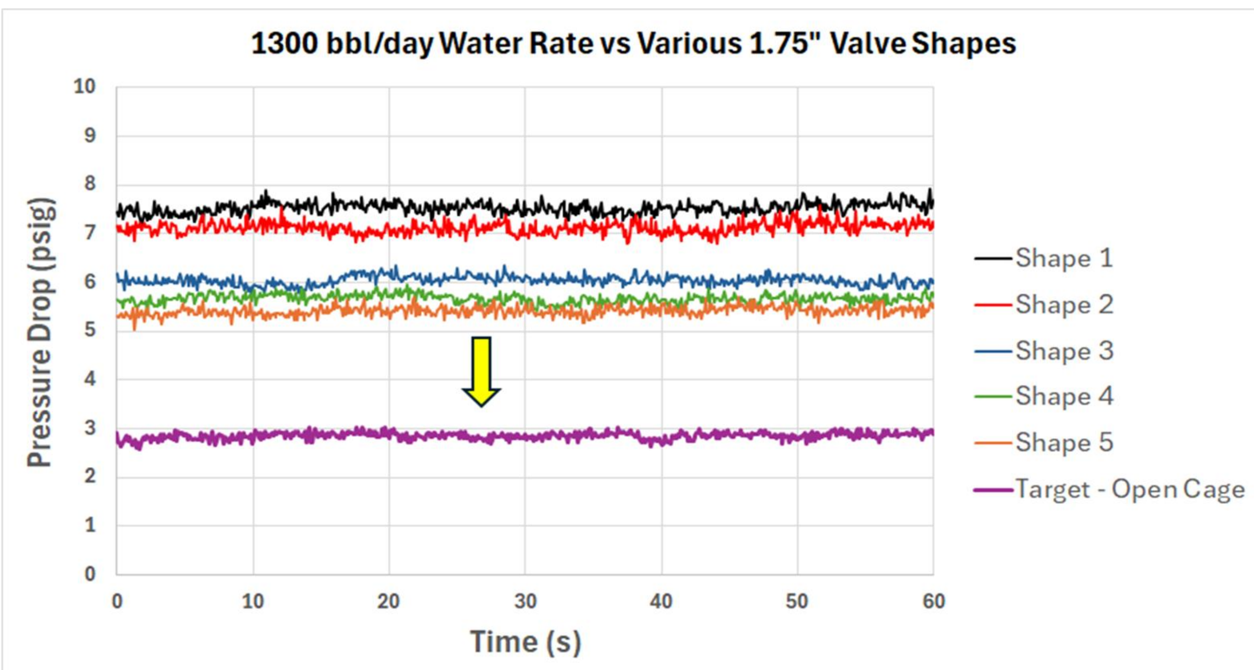


FIGURE 32 – FLOW LOOP TESTING (PRESSURE LOSS) VARIOUS HYDRODYNAMIC VALVE SHAPES WITH WATER

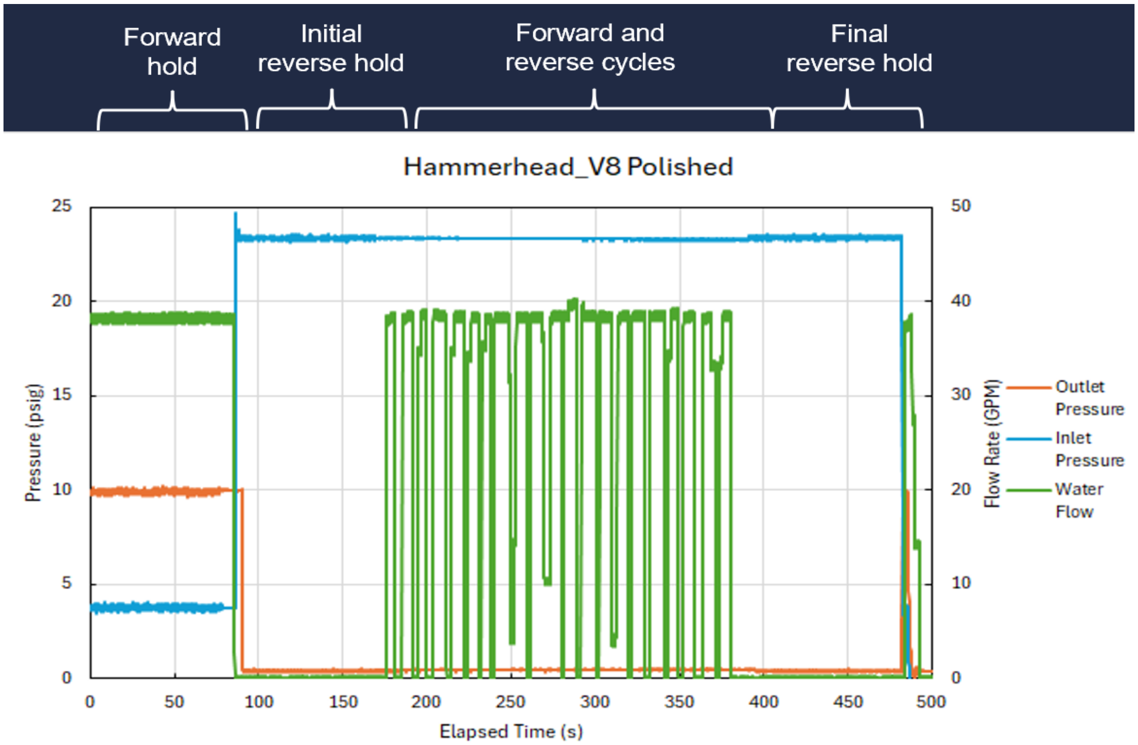


FIGURE 33 – FLOW LOOP TESTING VALVE CLOSING AND OPENING EFFICACY AND EFFICIENCY WITH WATER

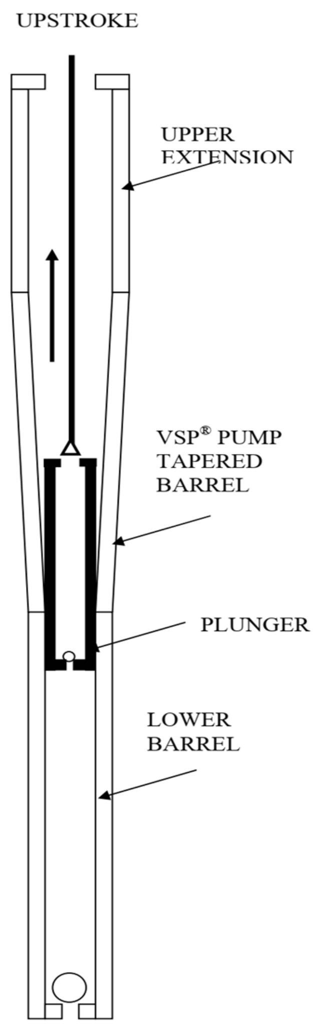


FIGURE 34 – PRESSURE BALANCING PUMP USING A TAPERED TOP BARREL SECTION FOR INCREASING PLUNGER FLUID SLIPPAGE

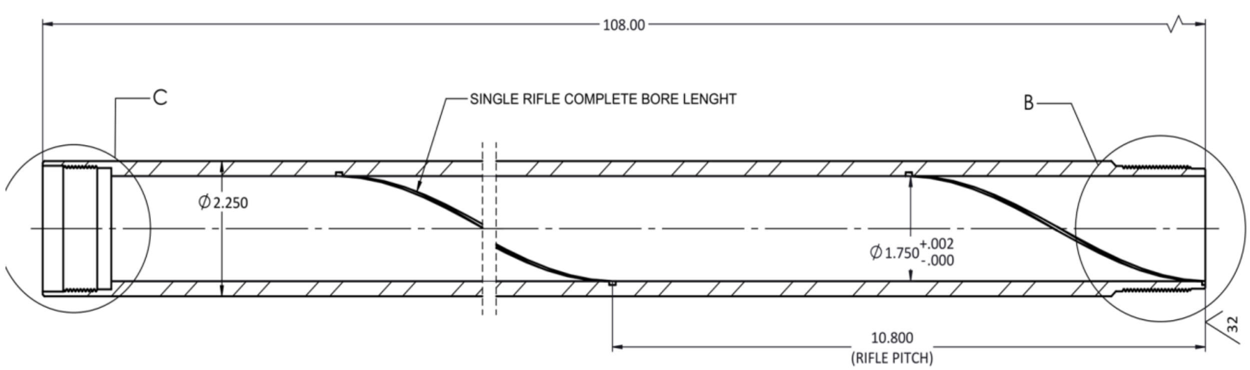


FIGURE 35 – SPACE OUT AND SOLIDS TOLERANT PRESSURE BALANCING RIFLED TOP SECTION FOR A PUMP BARREL

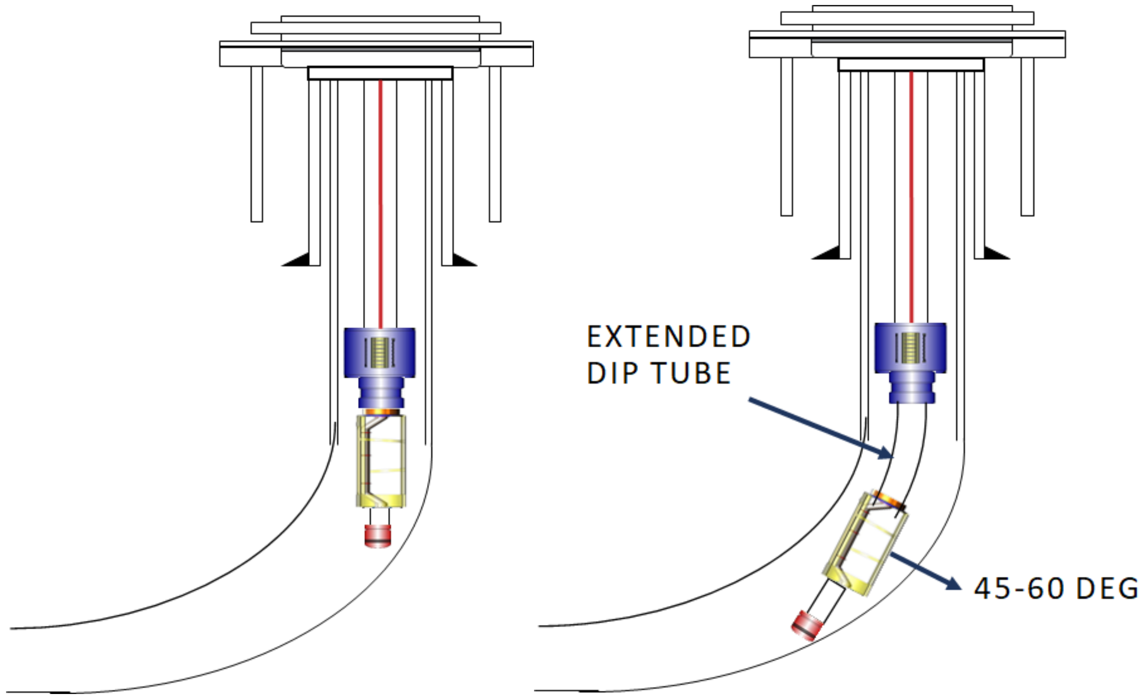


FIGURE 36 – SEPARATOR/PUMP AT KOP VERSUS EXTENDED DIP TUBE BHA WITH PUMP AT KOP AND SEPARATOR AT 45^{O} INCLINATION CONFIGURATIONS

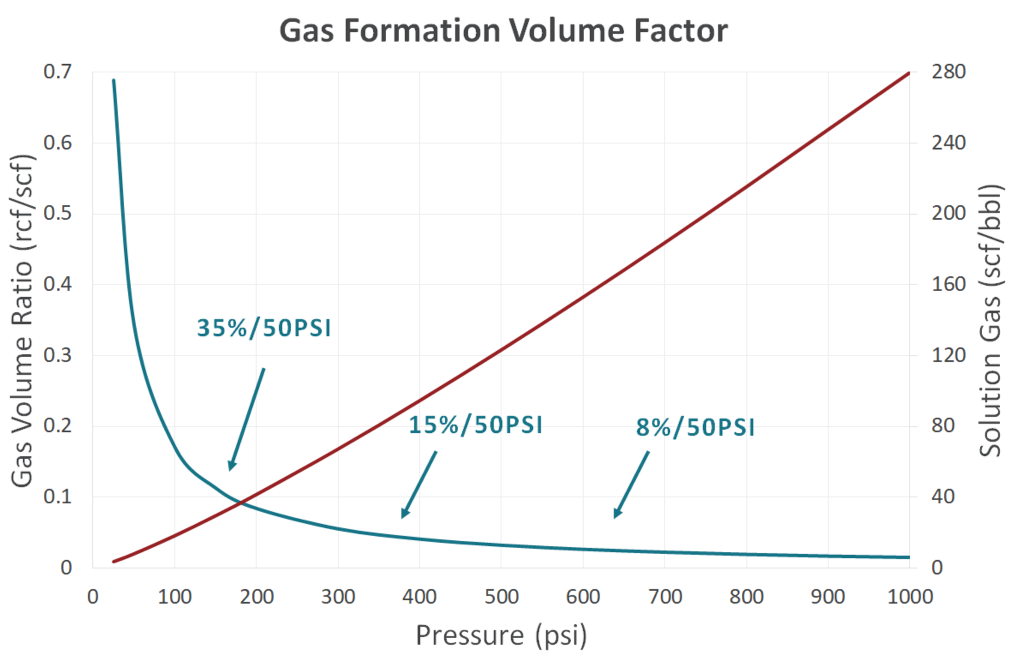


FIGURE 37 – GAS FORMATION VOLUME FACTOR AS A FUNCTION OF PRESSURE

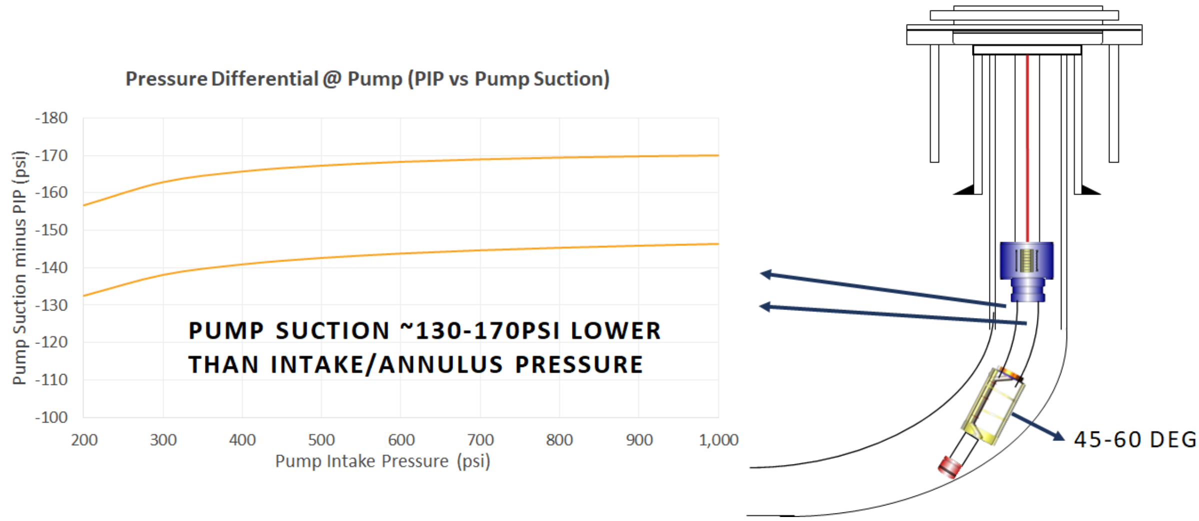


FIGURE 38 – PRESSURE FOR FILLING A PUMP CAN BE 130 PSI TO 170 PSI LESS WITH AND EXTENDED DIP SYSTEM

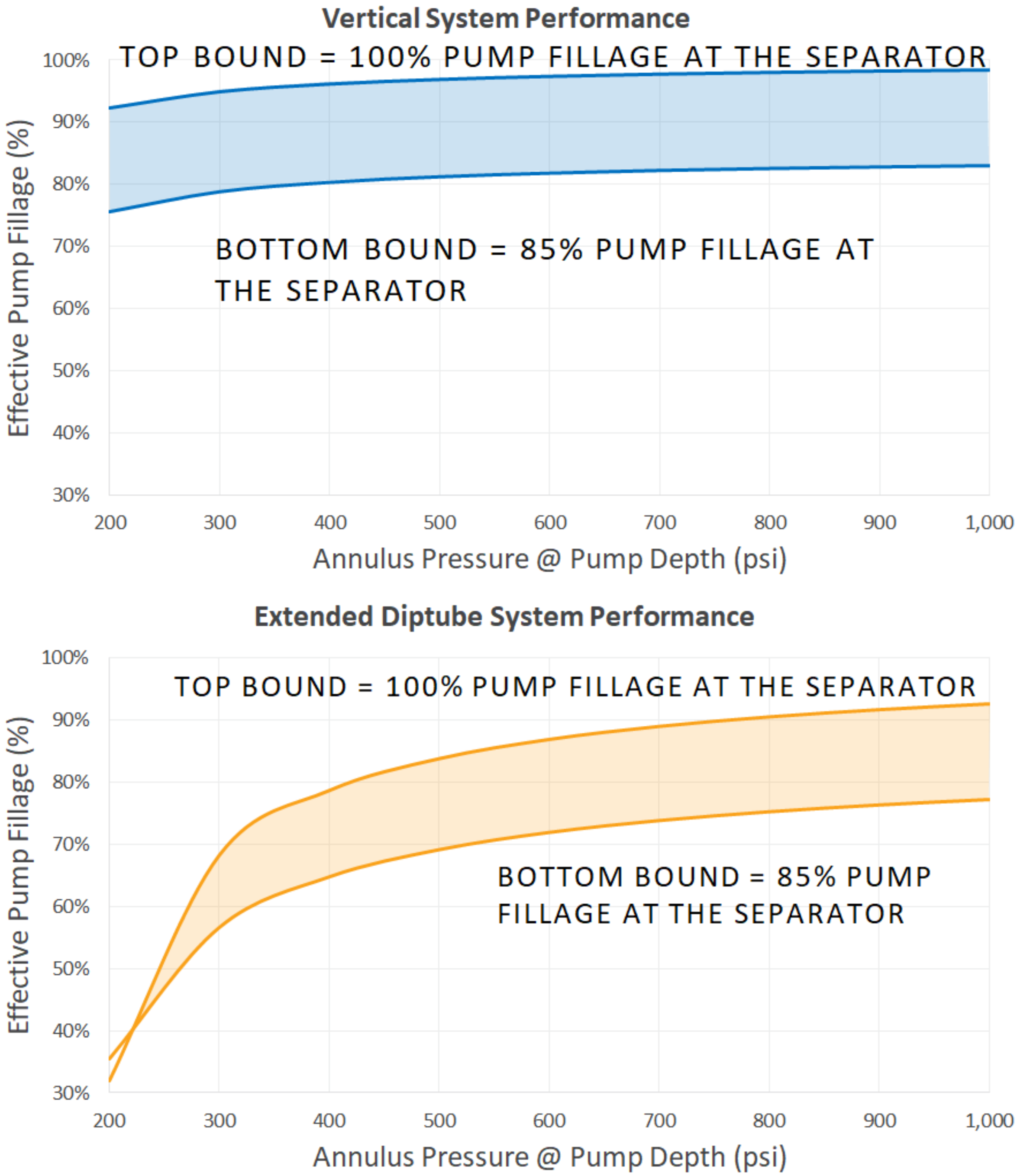


FIGURE 39 – EFFECTIVE PUM FILLAGE WITH AN EXTENDED DIP TUBE SYSTEM'S PRESSURE LOSS RELEASING OF SOLUTION GAS FROM THE OIL

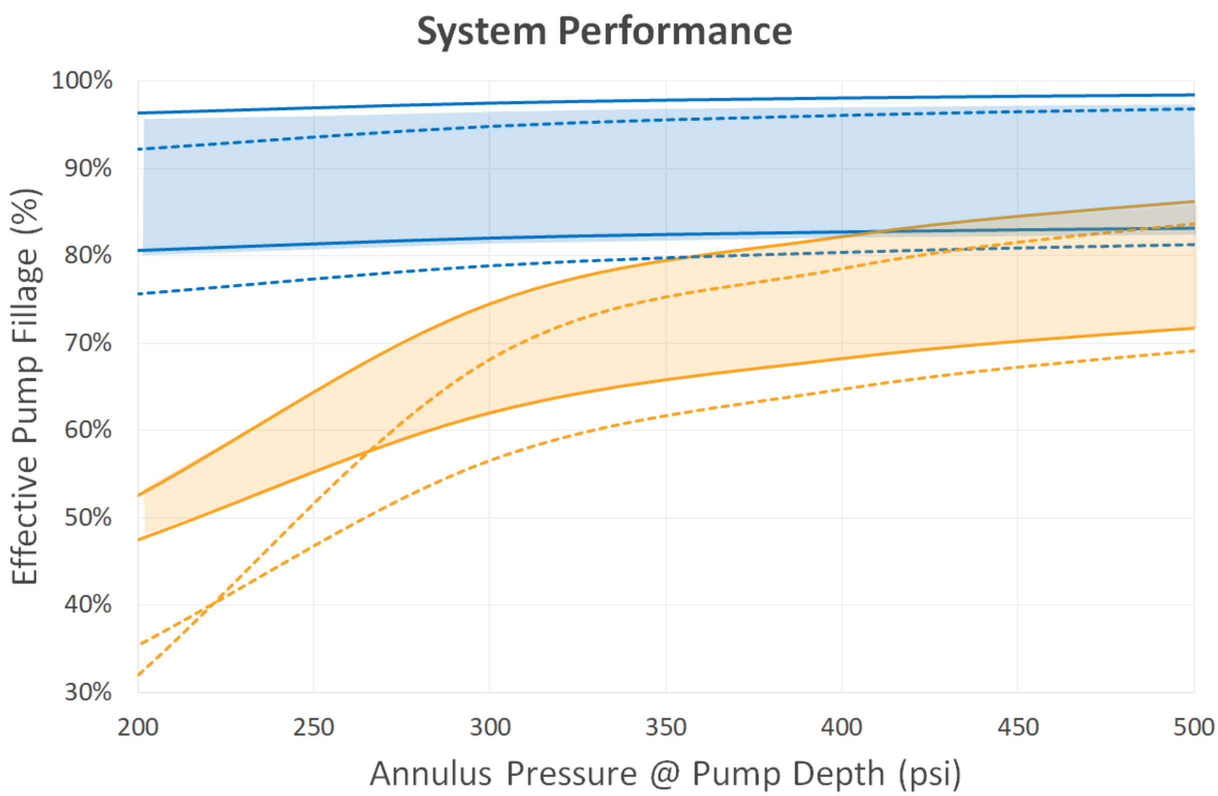


FIGURE 40 – STANDING VALVE PRESSURE LOSS AT LOW PIP'S

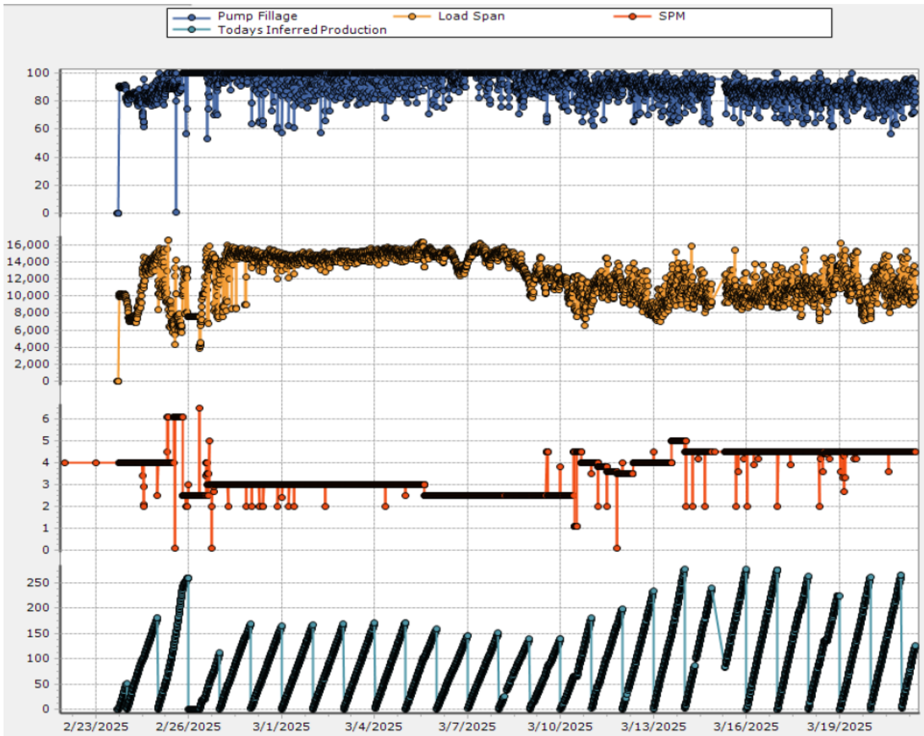


FIGURE 41 – EXAMPLE PUMPING PARAMETERS WITH AND EXTENDED DIP TUBE SYSTEM, PUMP AT KOP AND SEPARATOR AT 45 DEGREES INC

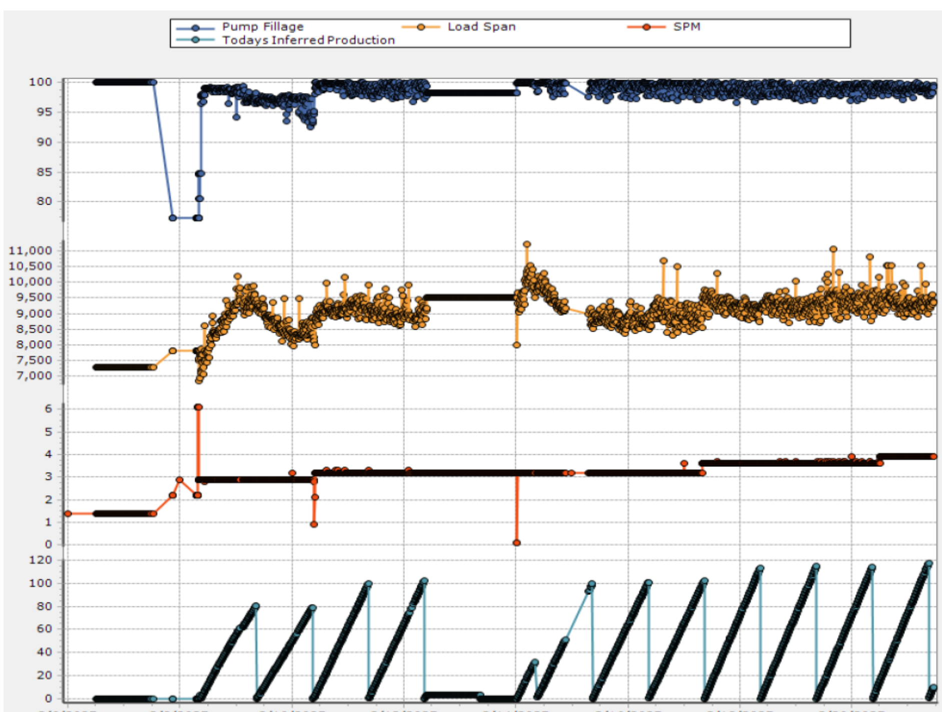


FIGURE 42 – EXAMPLE PUMP/SEPARATOR PLACED DEEP IN CURVE AT 85 DEGREES INCLINATION DEMONSTRATING CONSISTENT HIGH PUMP FILLAGE

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