# THE GEARED CENTRIFUGAL PUMP
# A NEW HIGH-VOLUME LIFT SYSTEM

John C. Patterson, Conoco-Phillips Co.
William B. Morrow, Harrier Technologies, Inc.
Michael R. Berry, Mike Berry Consulting LLC

## ABSTRACT

The Geared Centrifugal Pump (GCP) is a high volume artificial lift system consisting of a progressive cavity pump style rotating rod string driving a bottom intake ESP style multi-stage centrifugal pump via a downhole speed increasing transmission. The heart of the system is the unique transmission that utilizes a novel gearing configuration that allows high torque and power. The GCP provides the high volume lift of an ESP but with better gas handling, simpler operation, and lower capital and operating costs. Additionally, since all downhole components are mechanical the GCP can operate effectively at very high temperatures.

## INTRODUCTION

The Geared Centrifugal Pump (GCP) is an artificial lift system consisting of the rotating rod string drive of a progressive cavity pump (PCP), driving the multi-stage centrifugal pump of an ESP via a down-hole speed increasing transmission (Figure 1). The transmission is required to increase the relatively slow rotational speed of the PCP rod string, typically less than 500 rpm, up to the 3500 rpm operational speed of the centrifugal pump.

The GCP was developed as an artificial lift system to provide the high volume lift of an ESP without the expensive and troublesome downhole electric motor and cable of the ESP system. The GCP utilizes efficient and relatively inexpensive industrial electric motors in the surface drive head, requiring none of the special controls or transformers necessary for high voltage ESP motors. The rotating rod string transmits power downhole with low power losses. This system combines the highly efficient GCP transmission assembly with a modern ESP pump. This combination results in a system with lift efficiency greater than 50%. Because all of the downhole components of the GCP are mechanical, the design is adaptable to very high temperature applications such as SAGD projects.

## GCP Components

### Drive head

The geared centrifugal pump utilizes the same drive system as a progressive cavity pump system. A surface drive head turns a rotating rod string that drives the downhole assembly. The only difference between a PCP drive head and one built for the GCP is the load capacity requirements of the thrust bearing carrying the rod load. In a PCP drive head the thrust bearing must carry the weight of the rod string as well as the fluid load. Since the rod string of the GCP does not carry any fluid load, the thrust bearing in the GCP drive head can have a correspondingly lower capacity. Another characteristic of both PCP and GCP drive heads is prime mover flexibility. Although the majority of well locations have electric power, the GCP can be powered by an IC engine, or a hydraulic motor, as well as by an electric motor. In addition, by changing sheave sizes on the drive head, input rotational speed can be varied, providing an inexpensive (although less convenient) VSD.

One of the features of the drive configuration of the GCP is the ability to change the pump rotational speed without having to install a permanent variable speed drive. The typical GCP drive head uses a belt drive speed reducer between the prime mover and the rod string. If the sizing of the GCP installed in a well turns out to be less than optimum, the pump speed can be adjusted to optimize the lift by merely changing the size of the motor sheave on the drive head. This capability allows the use of a portable VSD to determine the optimum pump speed based on the well's productivity, and then size the motor sheave to give that optimum speed. This capability gives most the operational flexibility of having a permanent VSD on the well, without the cost.

If the sizing of an ESP is likewise found to be less than perfect, optimizing lift by changing the pump speed requires a permanent variable speed drive installation, and may also require changing the downhole motor voltage to optimize the power requirement. Optimizing a GCP to maximize a well's productivity should be simpler and less expensive than for an ESP.

## Drive string

The GCP has the same rotating rod drive string as a PCP. The principal difference between the two systems is that the GCP rod string does not carry the fluid load of the pump (product of the pump differential pressure and the cross section area of the rotor), so the axial load on the rod string is significantly reduced, and hence the rod tension is reduced. In a PCP application the tension on the rods at the pump is equal to the fluid load. At the surface the tension is the fluid load plus the buoyant rod weight. A rod string in tension is pulled taut against the wall of the tubing if there is any deviation at all. This results in increased wear frequency and severity. The worst place for side loading in a PC pump is high (close to the surface) where the rod string tension is highest. High side load and the resulting rod-on-tubing wear is a function of the rod tension and the well deviation. Depending on the well profile and the dog-leg-severity, high wear areas can occur high in the well at a kick-off point or low in the well at high inclinations. The rod string in a GCP carries none of the fluid load. Rod tension at the bottom hole assembly is nil and increases to the rod string weight at the surface. This lower rod string tension results in much less forceful rod-tubing contact and significantly less severe tubing wear.

The downside to low rod tension is reduced rod rotational stability. Rod rotational stability decreases with increasing rod string rotational speed and increases with string tension. The GCP rod string turns at a relatively fast 500 rpm and tension at the pump is nil, so rotational stability can be an issue. Another issue is the natural frequency of the rod string at rotational speeds below 500 rpm. With 25 foot rods the natural frequency of the rod string plus the second and third order of the natural frequency can occur at or below the proposed 500 rpm operating speed.

The most troublesome part of the GCP rod string is at the pump where the rod tension is at a minimum. In the current 4600 foot installation, the bottom 250 feet of the string consists of guided rods with three spin-through centralizers on each rod (Figure 2). The last rod above the GCP assembly is a 10’ pony rod that is unguided to act as a flex shaft between the tubing center line and the offset to the stab-in receiver. Each of the remaining rods is equipped with one centralizer per rod located 1/3 of the length from the rod coupling. This position has the greatest effect on disrupting the first and second order frequency modes of rotational instability.

As an alternative to the above rod string configuration, running sinker bars for the last few hundred feet will also help stabilize the rod string by significantly stiffening the lower part of the string, as well as adding some weight at the bottom to help with rod stability up the hole. Since increased stiffness improves rotational stability, it may also be advisable to run larger diameter rods than are needed, just to get more stiffness.

The torsional strength for various rod sizes and alloys are given in Table 1. The approximate maximum operating GCP output horsepower for each size and strength, assuming 500 rpm and 85% downhole assembly efficiency is also given. It is assumed that the maximum operating rod string torque is 67% of the yield torque value given in the table to allow for potential over-torque conditions at start-up. In many areas start-up torque is essentially the same as operating torque so the maximum delivered horsepower to the pump is as much as 20% to 30% higher than the values given in the Table 1. The principal power limitation for a GCP is usually the strength of the rod string that will fit in a given tubing and casing, not the GCP itself.

Figure 3 illustrates the capacity of the GCP with depth with the rod torque limitation of 1000 ft-lbs, a GCP rating of 100 hp, a 250 psi surface tubing pressure, and an effectively nil pump intake pressure. The pump is operated at 3500 rpm with the rod string at 500 rpm. Each point is determined from the performance curve of the appropriate available pump with tubing flow pressure losses included. The capacity of the system decreases with increasing depth. At 1000 feet with a 500 rpm, 1000 ft-lb (100 hp) limitation, the GCP can produce 4500 BFPD. Rate capacity decreases with depth to 1000 BFPD at 7000 feet.

## Downhole assembly

The downhole assembly of the GCP consists of a stab-in receiver for the rod string, a top seal /compensator section, the transmission, a bottom seal/thrust bearing section, and the pump with an optional tail pipe or gas anchor, as shown in Figure 1. Produced fluid flows from the pump outlet into lower seal section cavity where it flows in the annular space between the shaft seal housing and the outer pressure housing, then through the transmission via D-shaped tubes running along side the gear train, then through the upper seal section cavity in the space between outer pressure housing and the shaft seal housing and compensator, and into the tubing and on to the surface. The rod string drives the transmission via a square-section stab-in connection in the receiver at the top of the upper seal section. Notice in Figure 1 that the tubing and the GCP assembly are not concentric. This is due to the fact that the transmission has a multi-stage parallel shaft gear train, inside a circular housing, and the power input is therefore off-center. Despite this eccentricity, the OD of the tubing is still within the OD of the transmission assembly, so the offset does not limit the minimum casing size, nor affect the running of the assembly.

### Stab-in Receiver

The stab-in receiver accepts the rod string stab-in connector and channels the fluid discharge into the tubing string. The downhole pumping assembly is tubing conveyed and run without the installation of the rod string. Once the tubing is landed, the square section stab-in connection is run with the rod string to mate with the stab-in receiver which connects the rods to the downhole assembly. The stab-in receiver and stab-in connection have sufficient length to allow for growth of the tubing due to fluid loading and thermal expansion at operating conditions.

As discussed previously, the fluid load is carried by the tubing and not the rod string so the axial load on the rods is their buoyant weight. The stab-in receiver is not on the centerline of the tubing, due to differing offsets of the transmission input shaft and the tubing string. Since the rod string is centralized in the tubing by spin through guides, this off-center location of the stab-in connector is compensated for by the last ten feet of rods, which are not centralized and act as a “flex shaft”.

### Upper Seal Section

The upper seal section houses the offset drive shaft extending from the rod string stab-in receiver to the off-center transmission input. It also houses the main transmission pressure compensator. The drive shaft runs through a cylindrical housing, containing multiple shaft seals to protect the transmission from well fluid contamination. Each seal has its own pressure compensator, and is isolated from both the produced fluid and the other seals. Contamination of the transmission could occur only with the serial failure of all three seals. On the drive shaft below the last seal is a high capacity thrust bearing that protects the transmission input shaft from both compressional and tensional loads from the drive shaft that may be due to incorrect installation and space-out of the rod string.

The transmission pressure compensator maintains the oil-filled interior of the transmission at the same pressure as the produced fluid flowing through the assembly. It is offset in the seal section housing, parallel with the drive shaft. The compensator design used in the current prototype has a very large working volume to accommodate the large changes in oil volume due to the significant variation in expected operating temperatures of a SAGD installation. For normal temperature applications, the main pressure compensator is a conventional design, with a much smaller working volume reflecting the small temperature changes of non-thermal use.

### Transmission or Speed Increaser

Below the upper seal section is the transmission. In the current design, the transmission is an oil-filled 5-stage, parallel shaft gear train, with an eccentric input shaft. The off-center output from the gear train is centered via a crown-spline centering shaft so that transmission output is concentric with the lower seal/thrust bearing assembly and pump (Figure 4a). The configuration of a parallel shaft gear train housed within a circular casing, results in significant open space between the gear train and the outer housing. This roughly D-shaped space is utilized for the produced fluid flow channels. These flow channels are two relatively thin-walled D-shaped tubes that pass along sides the gear train (Figure 4b). The D-tubes are isolated from the interior of the transmission and carry the produced fluid from the lower seal section, through the transmission and into upper seal section. Figure 5 is a photograph of the output end of the transmission showing the inlet to the D-tubes from the lower seal section chamber. The input end of the transmission looks similar, except the input shaft is offset to one side. This flow path configuration, as well as being an efficient use of available space, provides extremely effective cooling of the gear train.

The geometry of the transmission and the large D-tubes for fluid flow is a significant advantage over planetary type transmissions. Fluid flow for planetary transmissions requires a shroud to route fluid around the transmission. This shroud either increases the OD of the equipment which impacts the required casing size, or decreases the size of the transmission, reducing its power capacity.

The concern with the D-tubes will be corrosion and erosion. With the large cross sectional area of the D-tubes the velocity is low which should minimize the erosion potential. Corrosion can be addressed by material selection and the use of stainless steel. Additional corrosion and erosion protection can be provided by coatings.

The pressure compensator in the upper seal section keeps the interior of the transmission and the flowing fluid in the D-tubes at essentially the same pressure. The relatively thin walls of the D-tubes have significant flexibility and the tubes act as fast reacting ‘bellows’ reducing the severity of pressure surges in the system due to shut-downs or power failures.

### Lower Seal Section

The lower seal section houses the shaft carrying the transmission output to the pump. This shaft, like the drive shaft in the upper seal section, is equipped with multiple shaft seals to protect the transmission from well fluid contamination. In addition, this section houses the main pump thrust bearing.

### Centrifugal Pump

Below the lower seal section is the pump, which is identical to those used in ESPs. The intake is at the bottom of the pump, and is the lowest component of the entire GCP assembly. This configuration allows the fluid intake to be located below the bottom well perforation. Or, if needed, the intake can be equipped with a gas anchor (Figure 1).

Both configurations provide greatly improved gas handling capability compared to conventional ESP installations. If desired, the pump intake can be equipped with a rotary gas separator. To set the intake of a conventional ESP below the bottom perforation the pump and motor would need to be shrouded or a fluid bypass system would be needed to keep the motor cool. Shrouds increase the equipment OD limiting their application, depending on the casing.

Since the GCP is a bottom intake system a tail pipe can be run from the bottom of the centrifugal pump to below the bottom perforation. This maximizes the area between the casing and the tail pipe, which reduces the downward liquid velocity and increases the gas separation efficiency.

The entire 575 GCP assembly, less the pump, is about 42 feet long. The 456 units will be about 10% shorter. Both the 575 and 456 GCPs have field connections between the receiver and top seal section, the transmission and the lower seal section and between the lower seal section and the pump.

### High temperature applications

The GCP was intended from the outset to be easily adaptable to very high temperature use. All components of the transmission and seal sections are designed to be manufactured from either high temperature steel, ceramic or other high temperature resistant material. Due to the large temperature fluctuations expected in SAGD use, each seal has its own relatively large volume compensator, and the main pressure compensator, which maintains pressure equilibrium between the interior of the oil-filled transmission and the produced fluid, has a very large working volume. All components are designed to operate continuously at 250 °C (482 °F) when fabricated from the appropriate high temperature material.

## GCP TRANSMISSION

The heart of the GCP is the speed increasing transmission. Transmissions that have constraints on the size of the gears due to space limitations, such as the internal diameter of oilfield casing, but are nonetheless required to transmit significant power, present a difficult problem. The most obvious potential solution is to use a planetary or related epicyclic configuration, which have both a high power density and concentric input and output. However, the small diameters of oil field casing severely limit even the power of a planetary transmission – a situation worsened by the fact that significant annular space must be provided between the transmission case and the outer pressure housing as a produced fluid flow channel, further reducing the already small gearbox diameter. The problem is that an epicyclic transmission has only one gear set to handle all the input, and there just is not enough steel in the single gear set in these small diameters to handle significant torque and power.

One potential way to get more gear sets and hence more steel in the gear train is to utilize a parallel shaft configuration, with multiple gears on a common shaft. Although this option is appealing in concept, it has proven, in the past, to be impractical because of the difficulty in achieving load sharing among the multiple gears. That is, due to manufacturing tolerances, the multiple gears are not likely to be perfectly aligned and do not all engage simultaneously. The misalignment causes the gears to engage unevenly, resulting in some of them being loaded more heavily than they are designed for, causing premature failure due to excessive load or wear.

## Paired Helical Gearing

The solution to this dilemma is a new gearing configuration termed Paired Helical Gearing, or PHG. A PHG gear train consists of pairs of helically cut gears arranged along common shafts as shown in Figure 4a. Each half of the pair is cut in the opposite sense, so they look like a herringbone gears, except the two halves are not attached, and are free to move axially relative to one another. The shaft is splined, as are the gears, so that the gears are free to move laterally along the shaft, but can drive or be driven by the shaft. The basic principal that results in load sharing in a PHG gear train is that helical gears exert an axial force that is proportional to their radial, or torque load. This axial force is in a direction that would move the gear away from engagement were the gear free to move axially on its shaft. Conversely, if the gear were pushed axially in the opposite direction, the gear would move into greater engagement.

If one were to impose a load on such a gear train, for example the second stage in the transmission in Figure 4, shown in more detail in Figure 6a, the first thing to happen, before any significant rotation occurred in either shaft, would be the gears reacting to the tooth loads and immediately moving axially along their respective shafts until they bumped into one another, (Figure 6b). At this point, the gears would begin to bear increasing but varying tooth loads, depending upon the relative and uneven timing of each gear. The gears that bear the higher loads push axially more strongly than their neighbor gear, moving away from engagement, while at the same time pushing those neighbor gears into more engagement. This process continues among all the gear pairs until the axial forces are opposite and equal, and no more lateral movement occurs. Since the axial force of a helical gear is proportional to the torque load on the gear, when axial forces among the gears are balanced, gear torsional loading is also balanced.

This load sharing was demonstrated by an instrumented four-pair model of the first stage of a 100 HP transmission. Torque transducers were located in the hollow splined drive shaft, between each gear pair, to measure the torsional load each pair of gears was imparting to the shaft. The result of one run is shown on Figure 7. Input torque was about 920 ft-lbs as shown by Bridge 2 located between the input and the first pair of gears. The fourth pair of gears was putting some 220 ft-lbs of torque into the shaft (Bridge 5), the third pair was contributing 450 ft-lbs (Bridge 4) less 220 ft-lbs or 230 ft-lbs, the second pair 680 ft-lbs (Bridge 3) less 450 ft-lbs or 230 ft-lbs, and the first pair was adding 920 ft-lbs less 680 ft-lbs or 240 ft-lbs. The fact that the torque split between the four pairs is not perfect was due to spline twist and binding, a situation remedied in the current designs. Nonetheless, even in this early PHG configuration, the load sharing among the four pairs of gears was outstanding.

Since multiple gears can be mounted on a common shaft, and load sharing assured, the capacity of a PHG transmission is not limited by gear strength or wear issues, but by the ultimate strength of the input shaft – a much less restrictive criterion. If a more powerful transmission is desired, all that is required is the addition of more pairs of gears until the input shaft strength is reached. Little additional design work is needed to significantly increase the capacity of the transmissions, and the incremental manufacturing costs are minimal.

The ability to add pairs of gears as needed also has profound effects on bearing issues. Bearing life and capacities are frequently the limitation of conventional transmissions, particularly epicyclic gear trains where the heavily loaded planet gear bearings are the Achilles heel of the designs. The PHG allows the addition of pairs of gears solely to increase bearing capacity and life.

The PHG gearing layout allows easy cooling of the gear train. Planetary transmissions with the heavily loaded planet gears ‘buried’ deep inside the gearbox and shrouded by the planet carrier and annulus gear, have real problems with cooling. With the parallel shaft configuration of the PHG transmissions all the gears and bearings are in close proximity to the external cooling medium - in the case of the GCP, the lubricating oil the gear train is immersed in and the D-tubes filled with the relative cool produced well fluid. This close proximity allows rapid and effective dissipation of the heat generated in the gear meshes and bearings. Keeping the gear train, and hence the lubricating oil, cool is critical to transmission life.

## Transmissions Capacities

The first GCP transmissions built and tested were 100 HP 7 in. models (i.e. designed for 7 in. casing) and were prototypes for a high temperature GCP intended for use in heavy oil thermal projects, principally SAGD. These prototypes were constructed from normal steel, but designed from the outset to be fabricated from high temperature steel to withstand the 250 °C (482 °F) expected SAGD operating temperature. The first prototype 7 in. GCP is currently installed in a well in West Texas, producing over 1400 BFPD from 4600 feet. Since then, a normal temperature 200 HP 7 in. transmission has been designed and is in the construction and assembly phase. All 7 in. transmissions have a 5.75 in. OD housing. In addition to the 7 in. models, a 75 HP 5-1/2 in. model (i.e. designed for 5-1/2 in. casing with 4.56 in. OD housing) is nearing the end of surface testing, in preparation of downhole installation. Construction of a 120 HP 5-1/2 in. model is complete, and surface testing will begin in the near future. Conceptual studies of larger sizes have been made showing that GCPs designed for 8-5/8 in. casing with power ratings in the 300 to 400 HP range are entirely feasible. All GCP transmissions are designed for a minimum 17,000 hours (two years) life at full power.

Table 2 summarizes the GCP sizes in current development, along with the casing, tubing and rod sizes required for the rated horsepower. For clarity, GCPs are identified by the OD of the assembly and the rated horsepower. GCPs designed for 7 in. casing have an OD of 5.75 in., so a 100 hp unit is a ‘575-100’, for example. Units designed for 5-1/2 in. casing have an OD of 4.56 in., so the 75 hp GCP for 5-1/2 in. casing is a ‘456-75’.

Table 2 is for a rod rotation speed of 500 rpm. Experimentation at rod rotation speeds in excess of the design speed of 500 rpm is ongoing to determine whether stable rod rotation can be achieved at significantly higher speeds. There is a great incentive to increase rod speed, as downhole horsepower is the product of torque and speed. Increasing the rotational speed allows increasing the power with the same peak torque. Since input shaft strength, i.e. shaft diameter, is the limiting factor on input torque in a PHG transmission, and transmission diameter, i.e. casing size, is the limiting factor on input shaft diameter and strength, increasing the operating speed allows greater power in the same size casing. For example, if the 456-120 transmission were run at 650 rpm instead of 500 rpm, it would deliver over 150 hp. Revising the current 120 hp transmission to handle 150 hp at 650 rpm requires merely the addition of a few gears to maintain gear tooth wear and bearing life.

### ESPCP Transmission

Although the current transmissions have been designed as speed increasing units for the GCP, the PHG design can easily be adapted for use in ESPCP systems. Currently, ESPCPs use planetary transmissions, either two-stage for use with a conventional 3500 rpm ESP motor, or single-stage when used with a motor designed to run at half speed. Gear reduction ratio in the two-stage units is typically 11.5:1, giving a PC pump speed of about 300 rpm. Due to the limitations of small diameter planetary transmissions, the power of these ESPCPs is quite limited. For example, ESPCPs for 5-1/2 in. casing typically have capacities less than 40 HP, and ESPCPs for 7 in. casing units are typically limited to 80 hp. A GCP transmission adapted for ESPCP use by adding an additional stage to bring the normal 7:1 ratio to 11.5:1, would more than double the available horsepower downhole compared to these existing planetary transmission units. In addition, due the flow-through feature of the GCP transmissions, the fluid intake can be located between the transmission and motor, allowing the installation of a rotary gas separator run at motor speed at the fluid intake (Figure 8), a capability not available with the existing planetary designs.

## TESTING OF THE GCP

There has been significant testing of the GCP system components to quantify efficiency and evaluate potential operating problems. One of the advantages of a rotating rod string drive is the ‘softness’ of the start-up. Since the rod string under full torque load in a typical installation may have more than one full wrap per 100’, start-up to full torque may require 10 seconds or more to reach full load at the pump. This very soft start results in equally gentle pressure changes – on start-up. Shut-downs, from power loss, or rod breakage, on the other hand, will result in a very rapid drop in pressure, potentially overwhelming the compensation system for both the seals and transmission. Much of the testing of the GCP has focused on the behavior of the both the individual components and as well as the full assembly during these inevitable shut-downs.

The D-tubes have been tested to determine whether shut-in induced pressure surges would cause weld fracturing or permanent deformation of the tubes. No problems were encountered despite the extreme level of pressure change imposed on the tubes. The ‘D’ shape provides significant and instantaneous volume change with only minor elastic deformation. This characteristic was proven during full assembly testing, when an operational problem caused a pressure surge inside the D-tubes of over 350 psi – more than 10 times the maximum expected downhole surge. The D-tubes flexed to such an extent that they actually impinged on the gear train, as evidenced by loud grinding noises and later seen in the post-test tear-down. Neither tube leaked and the test was completed successfully.

The flow path through the GCP assembly, particularly the upper and lower seal sections, is tortuous, with several areal restrictions and many 'unstreamlined' components in the flow path. Testing was done to determine the amount of pressure drop due to this 'dirty' flow path. The incremental pressure drop through the assembly was only 9 psi at a 6000 BFPD flow rate – representing an insignificant power loss at this high rate.

In addition to these tests of individual components, full GCP assembly tests have been conducted with numerous starts and stops to document system performance and the ability of the seals to withstand the rapid pressure changes during shut downs. The seals and seal compensators in both the upper and lower seal sections, as well as the main transmission compensator, performed perfectly despite imposed pressure changes that were much more abrupt and of much greater magnitude than anything possible in a downhole application.

Surface testing of both the transmissions and the full pump assembly have shown that the GCP is a very efficient pumping system. Transmission only testing results in energy efficiencies ranging between 85% and 87%, for transmissions with normal surface finishes. Testing of the full GCP assembly, including pump, gave overall system efficiency of 53%, using a pump with a pump-curve efficiency of 60% at the operating rates and pressures. The transmission was the same as tested alone, except that the gears and bearing surfaces had been isotropically finished to 6 micro-inches. This result gives an overall efficiency of the GCP system - transmission, seals and thrust bearing - of 88%, not including the pump. Assuming the surface drive head and rod string drive efficiency is 90%, and pump efficiency is 65%, the GCP will deliver 50% total lift efficiency.

## FIELD TRIAL

The first field trial of a GCP is being carried out in a former ESP-equipped well in West Texas. The well had been producing about 1300 BFPD with a 98% water cut and a very low gas rate from 4600 feet, when it was converted to GCP lift. The principal purpose of the conversion was to understand the installation and operational issues of a GCP in an actual oil well, determine the 'real world' efficiency of this new artificial lift system, and test some of the features unique to the high temperature GCP.

The well, a producer in a carbonate waterflood, was chosen because it has 7 in. casing through the productive interval and it has sufficient total fluid PI to test the operational lift capacity of the GCP. The GCP installed in the well is a normal temperature version of the 100 HP high temperature model being planned for use in Conoco-Philips' SAGD project in Alberta, and hence is designed for the 7 in. or larger casing used in that project.

The system installed in the well consists of a transmission and seal sections with the same large capacity pressure compensators that will be used in the high temperature applications. The transmission has a nominal rating of 100 HP. The pump used is a 147 stage 400 series Green Country GCP4-31 with a 3500 rpm operating range of 1000 BFPD to 1675 BFPD. Since there was not enough rathole to install the entire unit below the productive interval, the GCP assembly was set above the perforations and the pump inlet was extended with a 68 ft. 2-7/8 in. tubing tail extending below the perforations.  The tailpipe was equipped with an eccentric torque anchor that allows vertical movement but not rotation of the pump assembly or tubing and a screened intake.

The drive string consists of 1 in. Grade KD drive rods fitted with centralizers on each rod. Most of the centralizers used are spin-through type, but several of the molded on centralizers are also installed (Figure 2) for evaluation purposes. The bottom 200 ft. of rods are equipped with three spin-through guides per rod, with the bulk of the remaining rods having a single spin-through located 1/3 of the length from the rod coupling to disrupt second-order harmonic rotational instability, as described above.

The drive head has a 500 rpm, 125 hp capacity with a single 100 HP side-mounted variable speed rated electric motor. Speed reduction ratio is about 3:1 with a design input drive speed of 490 rpm at 75 Hz. A variable frequency drive is installed which provides easy speed changes as well as detailed drive motor performance and diagnostics.

The unit was started at about 270 rpm and ramped up to the full 490 rpm over an eight-day period. Initial fluid rates were about 850 B/D increasing to nearly 1500 B/D at the full 490 rpm drive speed. The VFD continuously provides read-outs of amperage and voltage, current frequency, and motor output power and torque calculated form manufacturer supplied motor performance curves. The well is also equipped with downhole pressure sensors on the pump intake and outlet. Unfortunately only the pump outlet pressure sensor is operational.

At the surface, the well is equipped with a turbine meter for fluid rate and cumulative production measurement. The well makes very little gas and well testing at the tank battery has shown the turbine meter to be reasonably accurate. The well also has continuous side-stream chemical treating down the casing annulus for both scale and corrosion control. Fluid level shots have been erratic and the flush is believed to be responsible.

The performance of this first installation of a GCP has been excellent. Some vibration occurred in the surface equipment and piping at certain speeds but none is evident at 490 rpm. The overall efficiency of the system as well as the drive train efficiency has been very good. Table 3 gives representative examples of calculated GCP performance at three pump speeds. Overall system efficiency is the pump output hydraulic power divided by the input power to the motor, and increases to 50% at higher loads. Drive train efficiency, including the drive head, rod string, and GCP down assembly less pump, is the overall system efficiency divided by pump efficiency and exceeds 75% at higher loads.

Note that the pump output differential pressure, a critical factor in the calculation of hydraulic power output, was taken from the manufacturer’s pump performance curve. The pump actually used in this installation had been independently flow tested before installation and performed essentially the same as given on Green Country’s pump performance curve. However, as a check on the validity of the calculated performance from the pump curves, the GCP performance was also determined from producing fluid level data and measured pump outlet pressure. This independent determination corroborated the efficiency calculations from the pump curves.

### Future Field Tests

The next test in West Texas will be in a 5-1/2 in. cased well that produces at a high GOR which is causing significant problems with conventional ESP run life. The GCP, with its better gas handling capability by virtue of the bottom intake, as well as its ability to tolerate gas locking without damage to the downhole equipment (no motor to overheat), should perform well in these difficult wells.

## CONCLUSIONS

1. The Geared Centrifugal Pump, GCP, is a new high volume lift system that combines the drive of a progressive cavity pump system and the multi-stage centrifugal pump of an ESP via a downhole speed increasing transmission.
2. The heart of the GCP, the downhole transmission, incorporates paired helical gearing, PHG, that allows very high torque and horsepower, giving the GCP lift capacities competitive with ESPs
3. The pump inlet in the GCP is below all other system components, allowing the fluid intake to be located below the producing perforations. This configuration provides greatly improved gas handling capability compared to conventional ESP installations.
4. All downhole components of the GCP are mechanical, allowing the GCP to be adapted to use very high temperature environments, such as SAGD projects.
5. The GCP has proven to be very efficient in surface and downhole testing, and is expected to meet or exceed the energy efficiency of other high volume lift systems.
6. The PHG transmission can be easily adapted for use in an ESPCP system. Use of a PHG transmission would more than double the available power downhole, compared to the currently used planetary transmissions. In addition, the PHG transmission would allow the use of a rotary gas separator run at motor speeds – a capability not available with the existing designs.

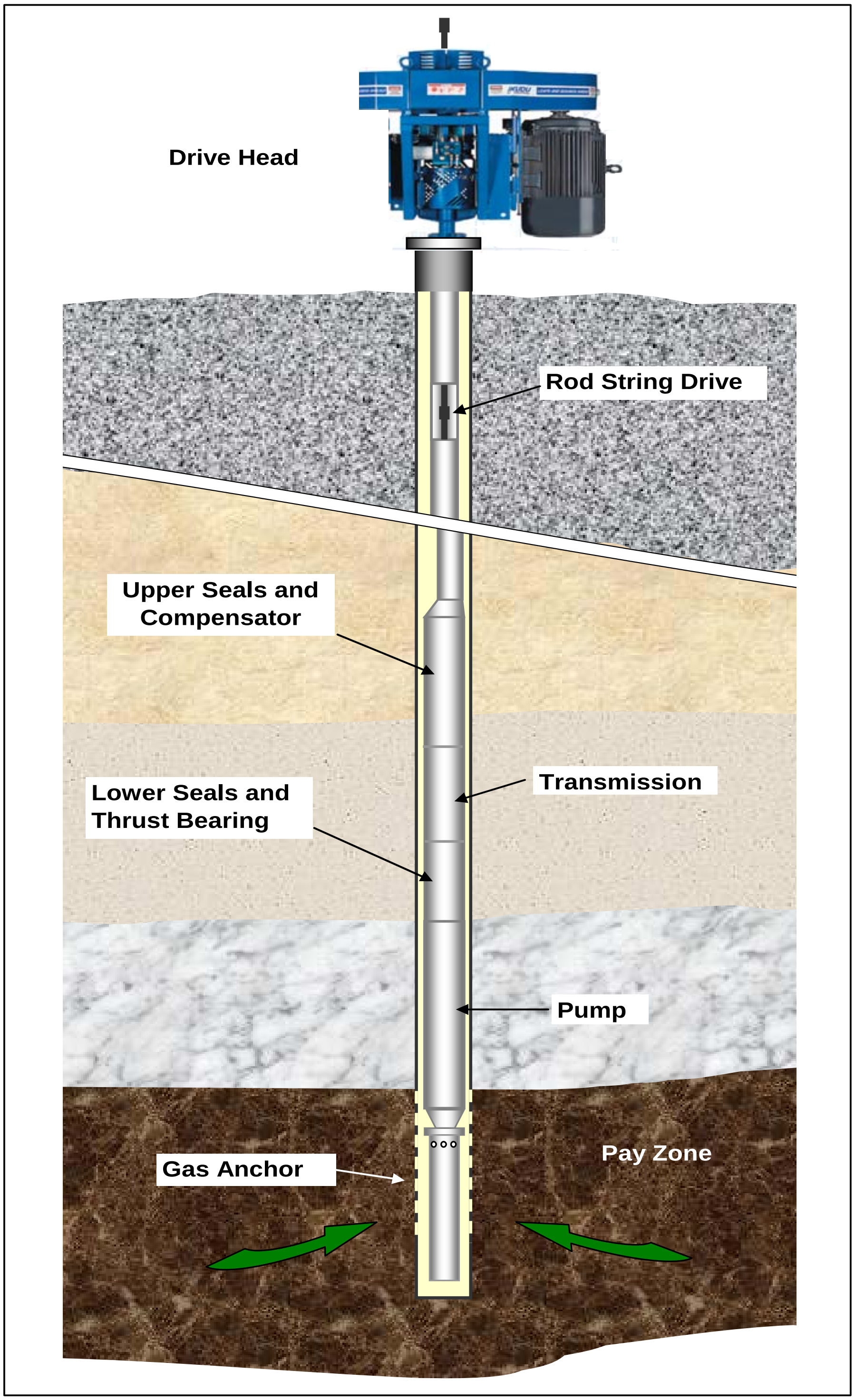



Figure 1 - Schematic of a Geared Centrifugal Pump

## Table 1

### Maximum Torque and GCP Power for Various Ros Sizes

Max GCP HP = 85% trans eff x 67% rod derate x (max rod torque x 500 RPM / 5250)

| | Grade D | | Special Service Alloy | |
|---|---|---|---|---|
| Rod Size | Max. Torque ft-lbs | Max GCP Power, HP | Max. Torque ft-lbs | Max GCP Power, HP |
| 3/4" | 460 | 25 | 500 | 27 |
| 7/8" | 735 | 40 | 800 | 43 |
| 1"+* | 1100 | 60 | 1200 | 65 |
| 1-1/8"+** | 1570 | 85 | 1700 | 92 |
| 1-1/4"+** | 2100 | 114 | 2500 | 136 |
| 1-1/2"+** | 3000 | 163 | 3750 | 203 |

* 2-7/8" tubing or larger
* 3-1/2" tubing or larger
\+ drive rods

## Table 2

### GCP Installation

**Recommended Casing, Tubing, and Rods**

| GCP Model** | GCP Size OD, in. | GCP operating max torque[t] ft-lbs | Output Power, HP | Min. Csg. Size, in. | Min. Rod Size, in. | Min. Tbg. Size, in. |
|---|---|---|---|---|---|---|
| 456-75 | 4.56 | 788 | 65[tt] | 5-1/2 | 1.0* | 2-7/8 |
| 456-120 | 4.56 | 1260 | 120 | 5-1/2 | 1-1/4 | 3-1/2* |
| 575-100 | 5.75 | 1050 | 92[tt] | 7 | 1-1/8 | 3-1/2 |
| 575-100 | 5.75 | 1050 | 100 | 7 | 1-1/4 | 3/1/2 |
| 575-200 | 5.75 | 2100 | 200 | 7 | 1-1/2 | 3-1/2 |

* slim hole couplings required
** BHP rating based on max input torque and 500 RPM
[t] maximum torque for start-up = GCP operating max torque x 1.5
[tt] rod limited

Figure 2 - Rod Centralizers

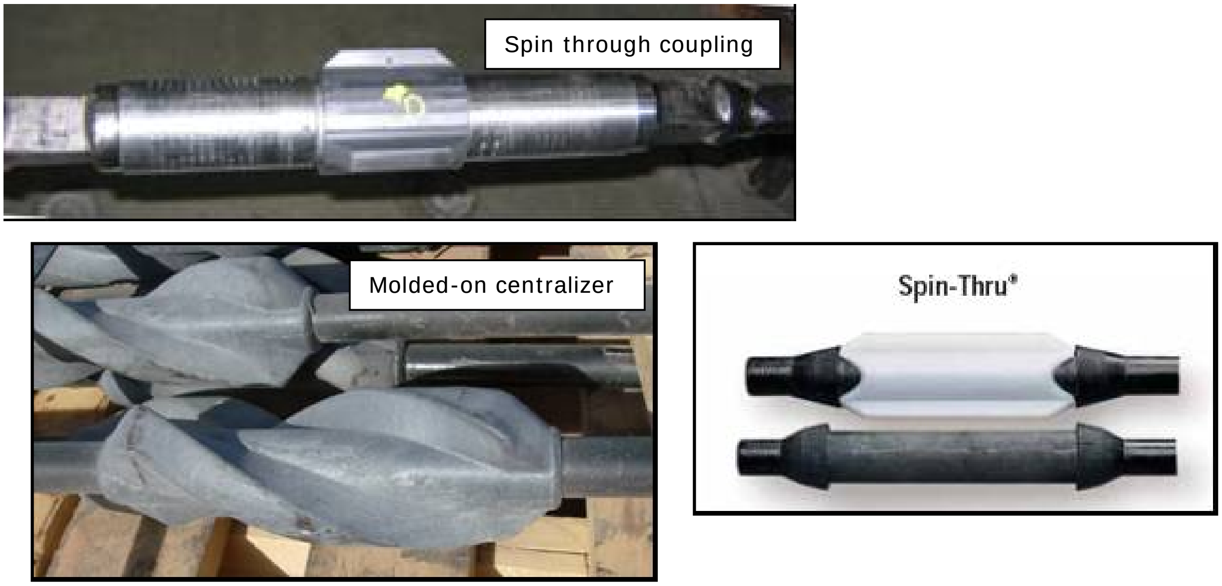



Figure 3 GCP Rate vs Depth and Torque

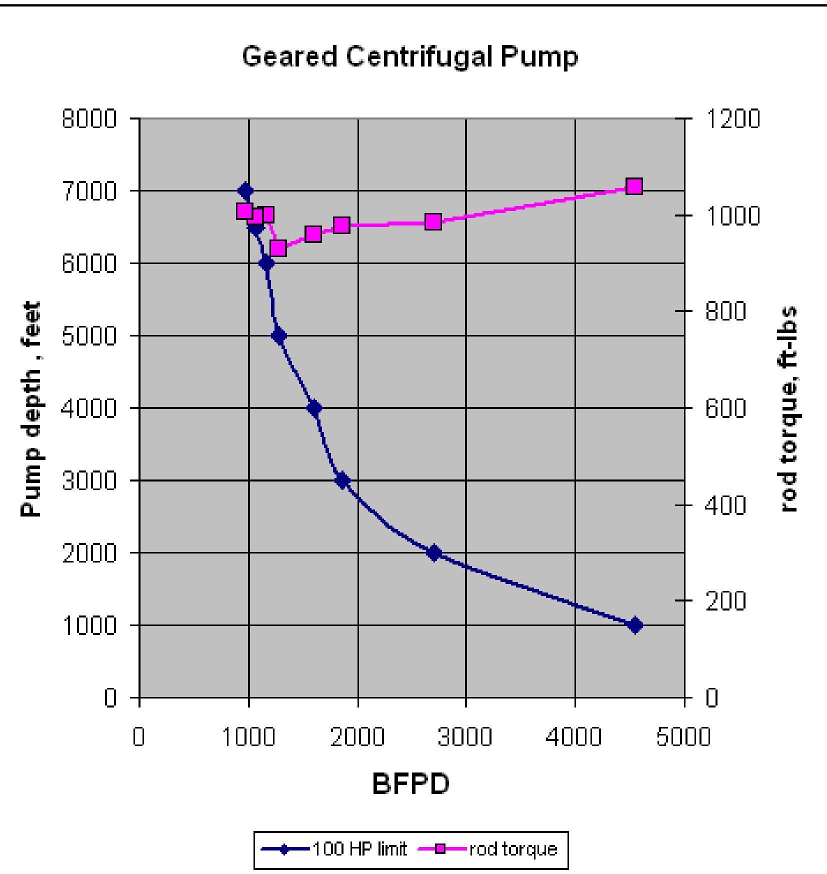


**Figure 4a**

**Schematic of GCP Transmission side view**

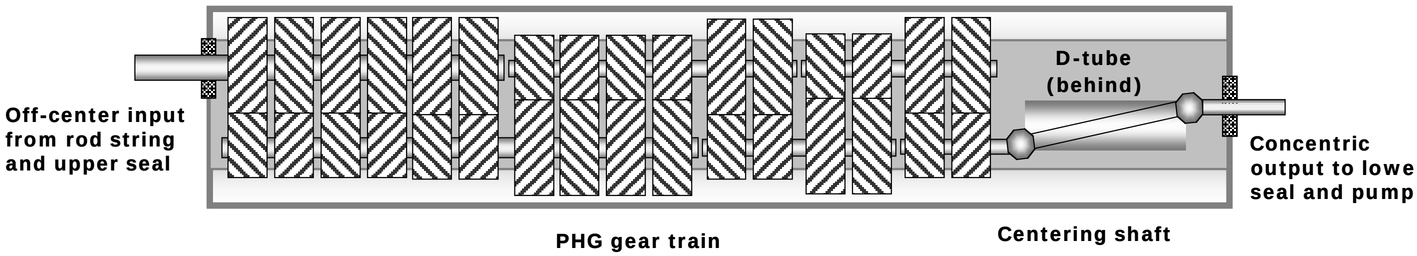



**Figure 4b**

**Schematic of GCP Transmission top view**

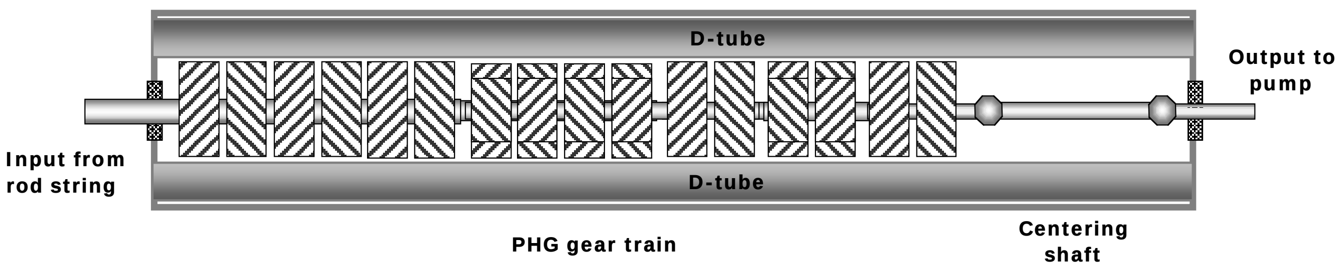



Figure 5
Photo of GCP Transmission

D-tubes at output end

**Figure 6a**

## Schematic of PHG Gear Train

**before any load is imposed**

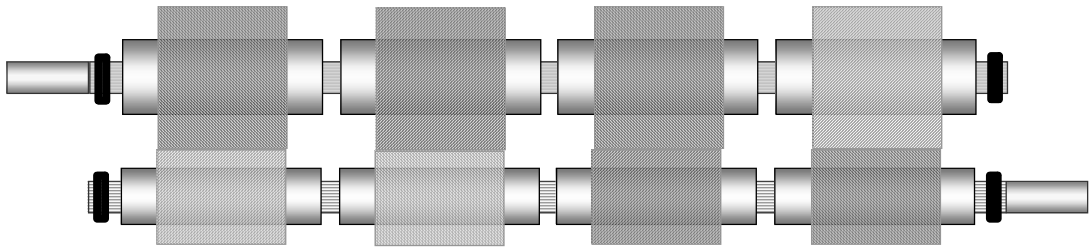


**Figure 6b**

**after load balance is achieved**

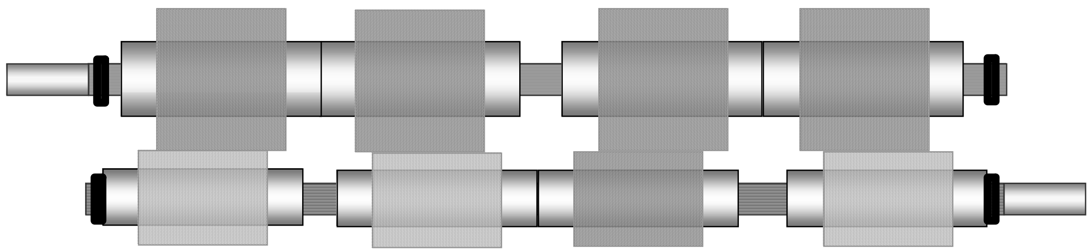
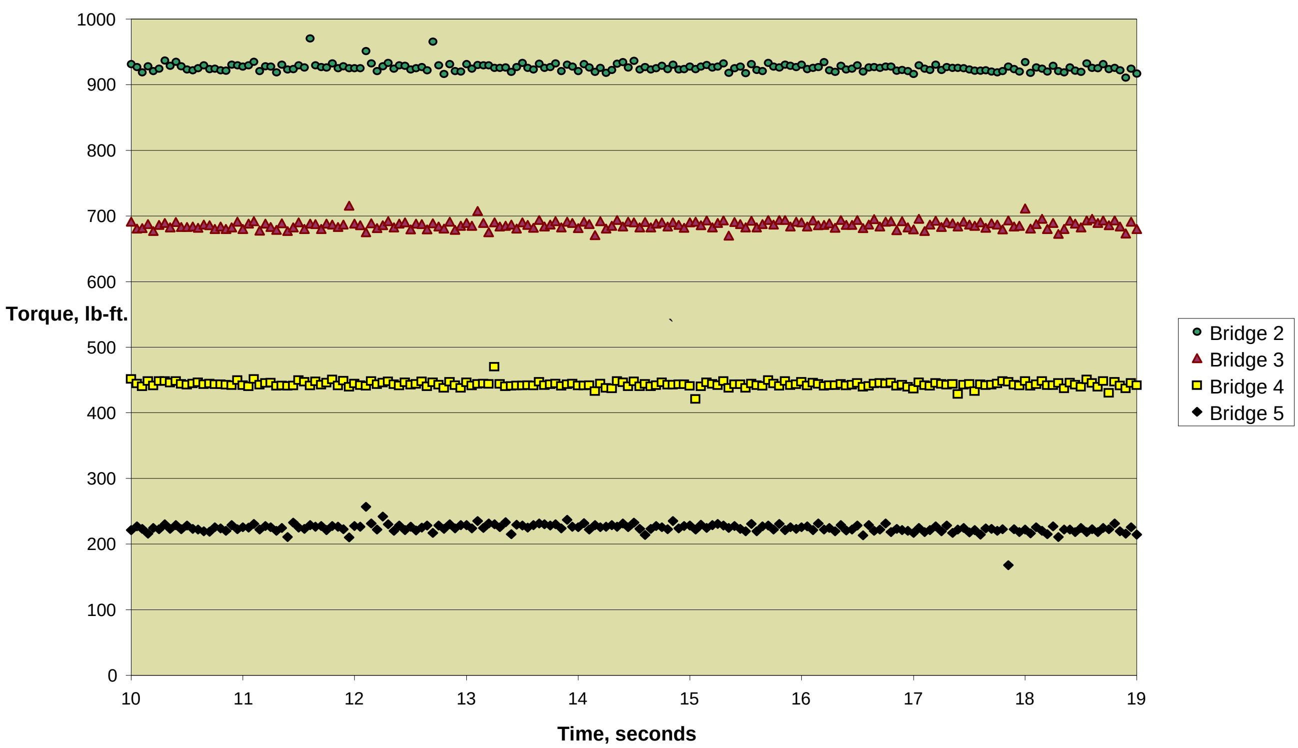


**Figure 7**

## GCP Single-Stage Prototype

**Input Spline Shaft Strain Gauge Torque**



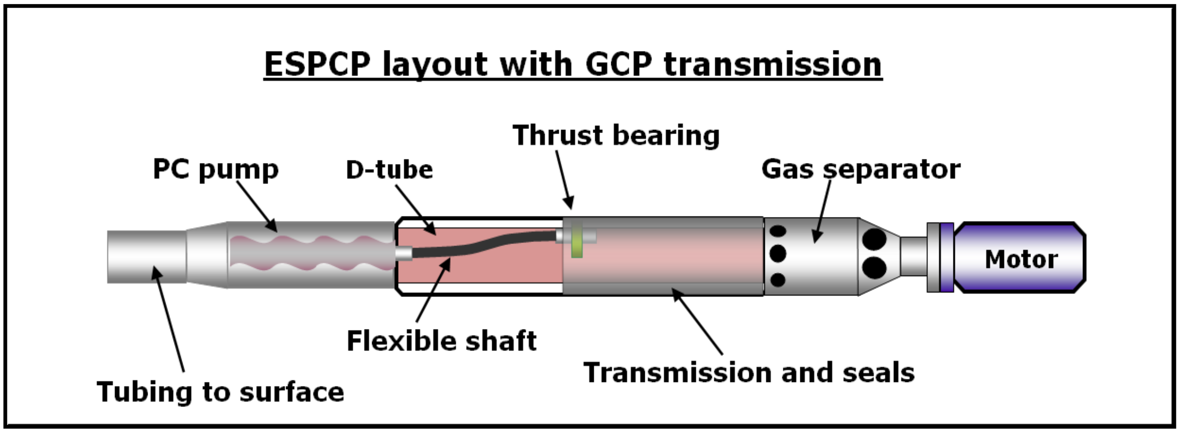


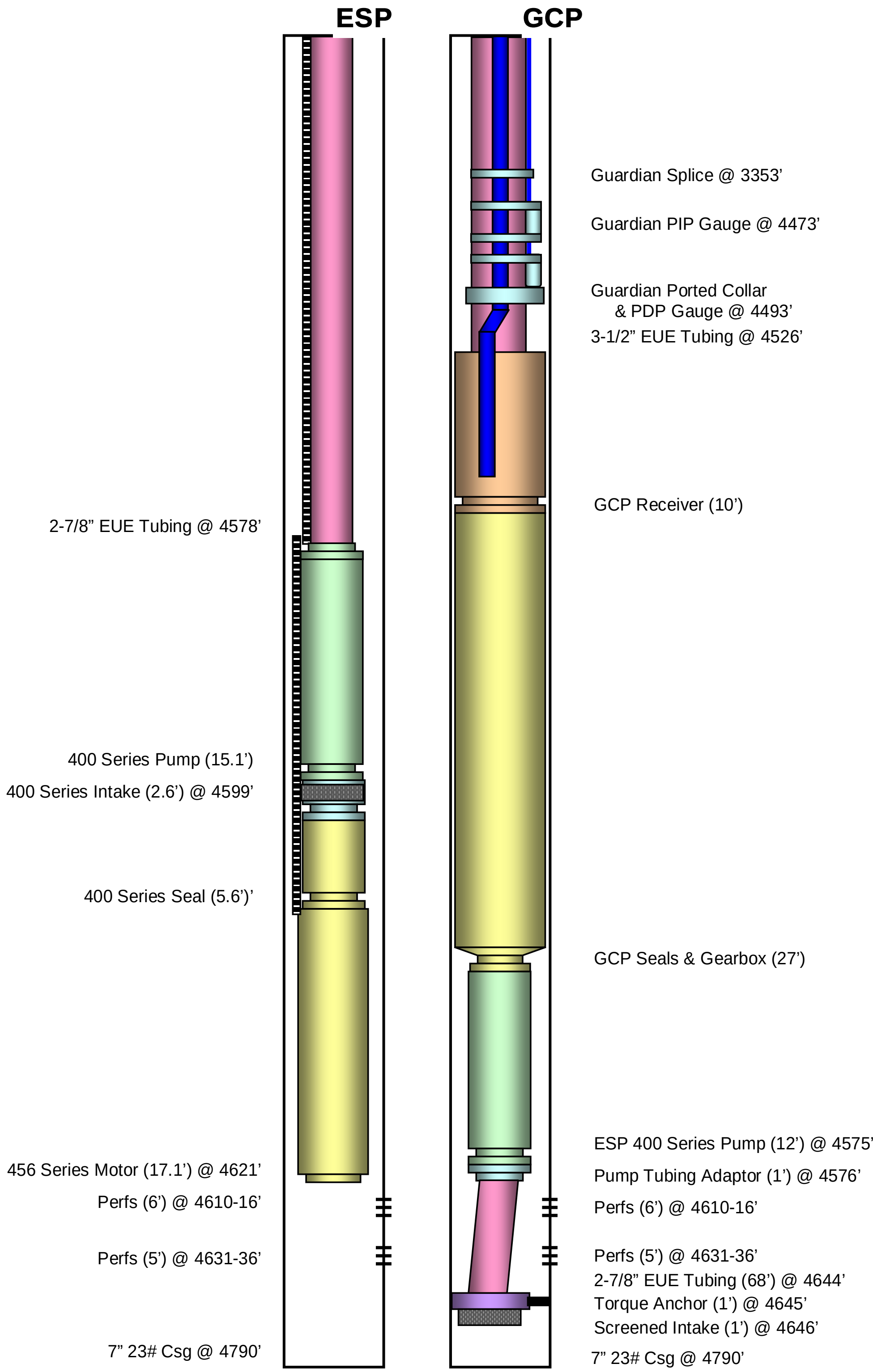

Figure 8

**Table 3**

**Geared Centrifugal Pump**
**Field Test Performance Summary**

| Date | measured values | | | | | | calculated values | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 8 | 9 | 10 | 11 | 12 | 13 | 14 |
| | BFPD (98% WC) | Rod RPM | Rod Torque ft-lbs | Pump RPM | Pump Discharge psi | Motor Output kw | Pump Differential psi | Pump BHP from Pump Curve | Pump Efficiency from Pump Curve | Pump Intake psi (5) - (8) | Drive Train Efficiency | Pump Hydraulic HP | System Efficiency |
| 2/18/09 8:07 | 1135 | 400 | 482 | 2846 | 2245 | 27.2 | 939 | 28.0 | 64.8% | 1306 | 70.6% | 18.1 | 45.8% |
| 2/18/09 14:25 | 1127 | 399 | 485 | 2838 | 2213 | 27.4 | 939 | 27.8 | 64.8% | 1274 | 69.6% | 18.0 | 45.1% |
| 2/19/09 18:25 | 1479 | 450 | 548 | 3201 | 2235 | 35.0 | 944 | 39.8 | 59.7% | 1291 | 77.9% | 23.7 | 46.6% |
| 2/20/09 13:45 | 1307 | 449 | 555 | 3194 | 2221 | 35.4 | 1149 | 39.6 | 64.5% | 1072 | 76.7% | 25.5 | 49.5% |
| 2/23/09 8:30 | 1302 | 449 | 554 | 3194 | 2218 | 35.3 | 1154 | 39.6 | 64.6% | 1064 | 77.0% | 25.6 | 49.7% |
| 2/23/09 13:00 | 1542 | 490 | 650 | 3486 | 2185 | 45.2 | 1219 | 51.4 | 62.2% | 966 | 78.0% | 32.0 | 48.6% |
| 2/25/09 13:30 | 1541 | 490 | 640 | 3486 | 2184 | 44.0 | 1221 | 51.4 | 62.3% | 963 | 80.1% | 32.0 | 49.9% |
| 2/26/09 13:30 | 1488 | 490 | 640 | 3486 | 2212 | 44.3 | 1292 | 51.4 | 63.6% | 920 | 79.6% | 32.7 | 50.7% |

1680 BFPD UT limit on pump at 3500 RPM




Diameters & BHA Lengths More or Less to Scale (Len/Dia=120/1)

Figure 9

Figure 10