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Development and Testing of a High-Pressure Downhole Pump for Jet-Assist Drilling

Topical Report - Phase II

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By
FlowDril Corporation
Kent, Washington

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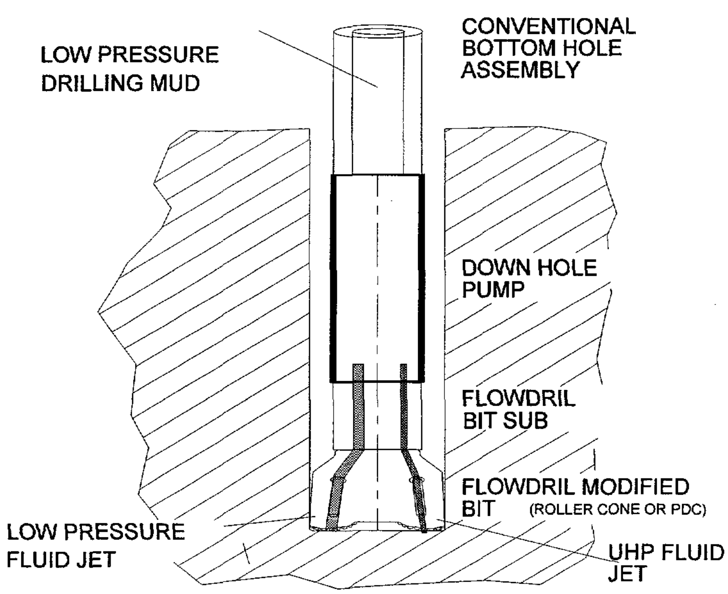
EXECUTIVE SUMMARY

The goal of jet-assisted drilling is to increase the rate of penetration (ROP) in deeper gas and oil wells, where the rocks become harder and more difficult to drill. Increasing the ROP can result in fewer drilling days, and therefore, lower drilling cost.

In late 1993, FlowDril and the Gas Research Institute (GRI) began a three-year development of a down hole pump (DHP®) capable of producing 30,000 psi out pressure to provide the high-pressure flow for high-pressure jet-assist of the drill bit. The U.S. Department of Energy (DOE) through its Morgantown, WV (DOE-Morgantown) field office, joined with GRI and FlowDril to develop and test a second prototype designed for drilling in 7-7/8 inch holes. This project, "Development and Testing of a High-Pressure Down Hole Pump for Jet-Assist Drilling," is for the development and testing of the second prototype. It was planned in two phases. Phase I included an update of a market analysis, a design, fabrication, and an initial laboratory test of the second prototype. Phase II is continued iterative laboratory and field developmental testing. This report summarizes the results of Phase II.

In the downhole pump approach shown in the following figure, conventional drill pipe and drill collars are used, with the DHP as the last component of the bottom hole assembly next to the bit. The DHP is a reciprocating double ended, intensifier style positive displacement, high-pressure pump. The drive fluid and the high-pressure output fluid are both derived from the same source, the abrasive drilling mud pumped downhole through the drill string. Approximately seven percent of the stream is pressurized to 30,000 psi and directed through a high-pressure nozzle on the drill bit to produce the high speed jet and assist the mechanical action of the bit to make it drill faster.



Down Hole Pump Approach to Jet-Assisted Drilling

The power required to drive the DHP is obtained from increasing the operating pressure of the surface pumps and delivering it downhole via the conventional mud stream. The surface mud pump standpipe pressure necessary to drive the DHP was between 3,400 and 3,500 psi.

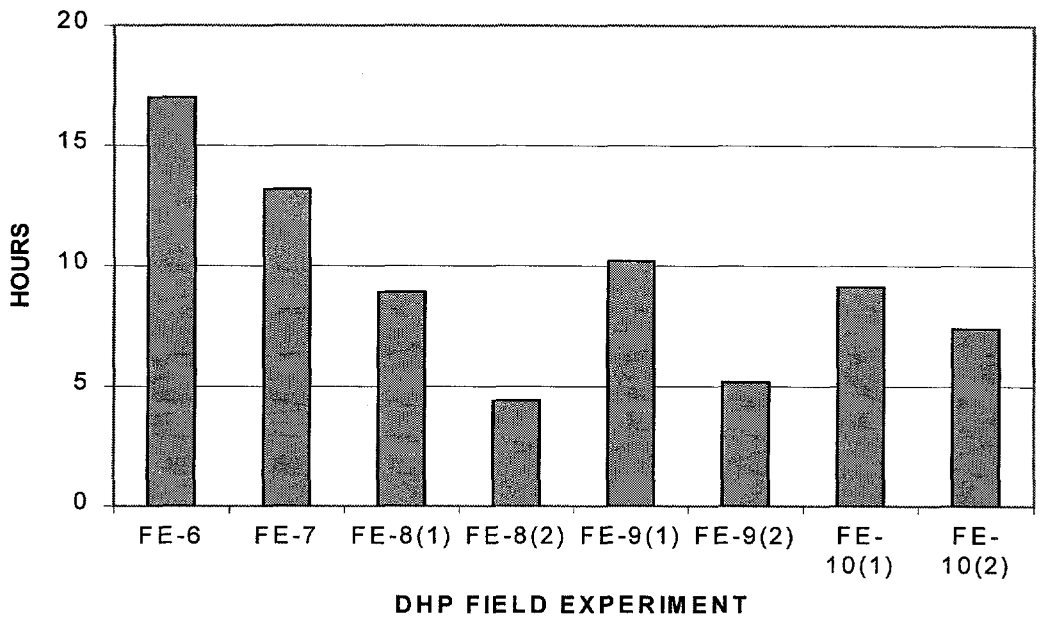
The outer diameter of the second prototype DHP is 6-3/4 inches. It was designed to drill in 7-7/8 inch diameter holes. During the design phase, attempts were made to improve several components and several aspects of the design over the 7-5/8 inch first prototype DHP that was designed to drill in 8-3/4 inch holes. These included increasing the cycling rate capability of the pump such that it could be operated at full specified cycle rate, improving the overall efficiency by reducing hydraulic losses, making it shorter as well as smaller in diameter, and improving the fishability of the tool. In particular, the mechanical design of the center connecting drive tube and connections through the drive pistons was changed significantly.

Over 50 laboratory test runs were conducted with the 6-3/4 inch second prototype, accumulating 480 hours of laboratory test time. A number of proprietary design improvements were introduced during laboratory testing to address mechanical strength and dynamic loading issues, low cycle fatigue, fabrication difficulties, and both drive fluid and high pressure sealing issues. The second prototype was an improvement over the first prototype with better relative performance and improved hydraulic efficiency. Near the conclusion of laboratory testing, the second prototype had achieved run times of about 40 hours with laboratory drilling mud.

Five field experiments were conducted with the 6-3/4 inch second prototype, all in 1996. They were designated FE-6 through FE-10. A total of 75 downhole test hours were accumulated over eight DHP runs, with the longest run of 17 hours occurring on drilling mud in a well in East Texas.

Downhole experiments FE-6, FE-7 and FE-8 were conducted in East Texas in the Travis Peak formation at depths between about 6,200 and 7,300 feet. As described by Watson et al (1996), this formation generally starts at depths of about 6,400 feet and is composed of laminated sections of shale, some limestone, sandy mudstones, muddy sandstones, and sections of very abrasive fine-grained sandstone. Downhole experiments FE-9 and FE-10 were supported in part by Statoil of Norway and conducted in a test well operated by RF-Rogaland Research in Stavanger, Norway. The DHP tests were at a depth of about 4,100 feet (1,250 meters) in a very abrasive granite formation.

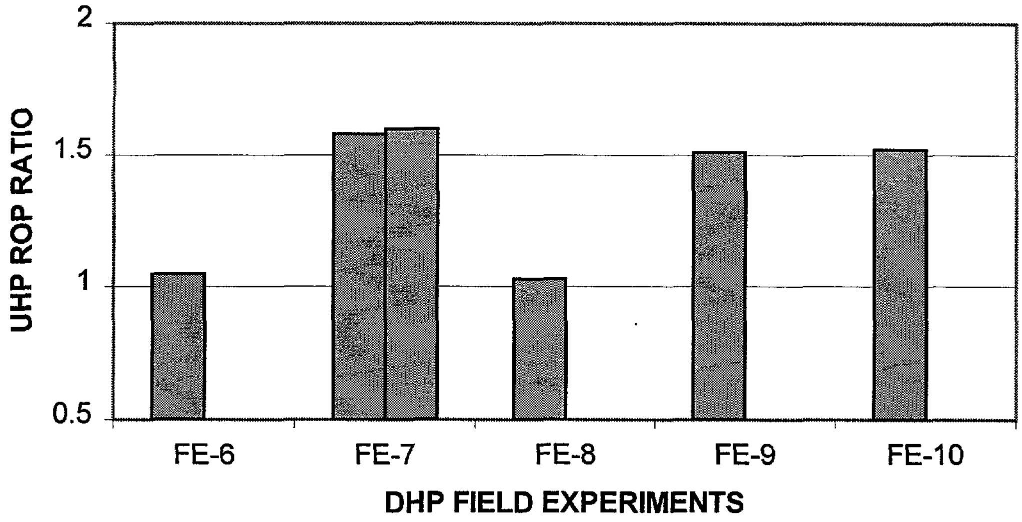
Hours achieved in each of the downhole field experiments are shown in the following figure. Failures in FE-6 and FE-7 were due to an unreliable sleeve retention method in the main valve in one case and overstressing the main valve drive rod in the latter case. These failures were resolved with design changes that were introduced and confirmed through laboratory testing, concluding in the 45- and 35-hour laboratory test runs mentioned earlier.



Second Prototype DHP Downhole Hours at Pressure.

The failures limiting downhole pump life in the last three downhole experiments, FE-8, FE-9 and FE-10, were all due to fluid erosion or abrasion within the DHP. Beginning with FE-8, it became apparent that the drilling mud in the field was more abrasive than in the laboratory. The field drilling mud became generally more abrasive with each successive field experiment, resulting in more severe erosion and more apparent abrasion.

For each of the DHP field experiments, ultra-high pressure (UHP) jet-assisted ROP data was gathered from either the operators, or from the rigs. In each case, conventional ROP data was also acquired from either a nearby offset well, or from either immediately higher or lower in the same well bore and formation. A comparison was then made between the UHP jet-assisted ROP and the conventionally drilled ROP data at the corresponding depths to determine the ratio of UHP jet-assisted ROP to the conventional ROP. Shown in the following figure is the ratio of UHP jet-assisted average ROP to the average conventional ROP, where the average ROP was determined over the range of depth drilled with UHP.



Jet-Assisted ROP Ratios during DHP Field Experiments.

The drill bits were tested in FlowDril's laboratory drilling facility before FE-7, FE-9, and FE-10 to ensure that the UHP jet-assist was effective. When the bits were tested prior to the field experiment, and the DHP was able to provide expected output pressures as in FE-7, 9, and 10, the ROP ratios were between 1.5 and 1.6.

Design and reliability issues remaining to be resolved for the DHP design are abrasion and wear of some of the low pressure dynamic components and erosion of the fluid valve manifolds. Specific areas that need to be addressed are erosion of the valve manifolds, upper and lower, erosion of the main valve drive rod, erosion within the low pressure drive pistons; abrasion between the connecting tubes and low pressure dynamic seals, and abrasion between and low pressure liners and the drive piston dynamic seals. The approaches being evaluated to address the fluid erosion and abrasion issues are more erosion-resistant materials and erosion-resistant coatings. It is believed, however, that to properly resolve the erosion and abrasion issues, several areas of the design of the DHP will require significant revision. As a result, it is planned to design, build and test a third prototype that is expected to be a pre-production prototype DHP.

1. INTRODUCTION

The goal of jet-assisted drilling is to increase the rate of penetration (ROP) in deeper gas and oil wells, where the rocks become harder and more difficult to drill. Increasing the ROP can result in fewer drilling days, and therefore, lower drilling cost. In late 1993, FlowDril and the Gas Research Institute (GRI) began a three-year development program of a downhole pump (DHP®) capable of producing 30,000 psi output pressure. This program was based on UHP mud pumping and sealing technology previously developed by FlowDril. The U.S. Department of Energy (DOE) through its Morgantown, WV (DOE-Morgantown) field office, joined with GRI and FlowDril to develop and test a second prototype designed for drilling in 7-7/8 inch holes.

This project, "Development and Testing of a High-Pressure Down Hole Pump for Jet-Assist Drilling," was planned in two phases. Phase I included an update of a market analysis, a design, fabrication, and an initial laboratory test of the second prototype. Phase II was continued iterative laboratory and field developmental testing. This report summarizes the results of Phase II.

The project was originally proposed to extend the DHP and jet-assisted drilling technology to drilling slimholes. Results of the market analysis for DHP jet-assisted slimhole drilling indicated that the slimhole market would be small (about 1/20th) compared to 7-7/8 inch hole size, the largest market segment. As a consequence of the market size and a desire to promote earlier commercialization of the DHP jet-assisted drilling technology, this project was re-directed from slimhole applications to development of a second prototype DHP for 7-7/8 inch hole size.

1.1 Jet-Assisted Drilling Background

In the early 1970s, the potential advantages of applying high-pressure (15,000 psi) jet technology to increase rates of penetration (ROP) were demonstrated by Maurer et al (1973) and by Fair (1981). They were able to demonstrate ROP enhancements between 1.2 and 2.9 times conventional rates in tests conducted in Florida and Texas. Both Maurer and Fair required that the entire fluid stream be pressurized. This resulted in extremely high power requirements (2,800 and 11,200 hydraulic horsepower), reliability problems, and safety concerns.

During the late 1980s and early 1990s, FlowDril developed a system for ultra-high pressure, 34,000 psi, jet-assisted drilling. A number of field projects were conducted demonstrating the effectiveness of ultra-high pressure jet-assist in improving ROPs at depths up to 11,000 feet. The system was described by Butler, et al (1990) and the most recent results were discussed by Veenhuizen et al (1993). About 30 to 40 gpm of the downhole mud stream was pressurized with pumps at the surface developing 600 to 700 hydraulic horsepower, and conducted to the drill bit through a dual-conduit drill string. The system required separate high pressure surface piping, standpipe, and kelly hoses, a dual swivel, and a dual conduit kelly. This allowed a high-velocity jet of drilling mud at the bit to be directed at the bottom of the hole to assist the mechanical action of the bit. The jet assists the mechanical action of the bit to achieve approximately 1.5 to 2.5 times conventional penetration rates.

Twenty-two field projects, 11 in West Texas and 11 in East Texas, totaling about 90,000 feet drilled, were conducted with this system through Grace/FlowDril, a joint venture with Grace Drilling Company. Two of the projects in East Texas in the Carthage field were performance bench mark wells: ET-3 drilled in 1990 and ET-9 drilled in 1992.

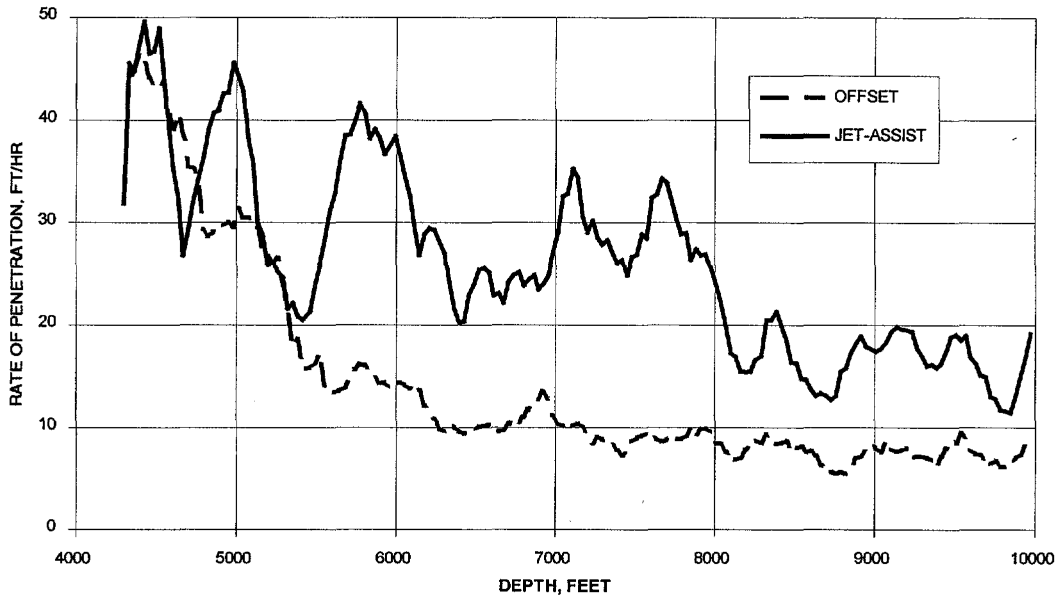


Figure 1. Ultra-High Pressure Jet-Assisted Rate of Penetration in a Gas Well in East Texas (ET-3) using the FlowDril® System.

As an example of the effectiveness of drilling with UHP jet-assisted drilling, the average ROP for each pipe joint achieved with UHP jet-assist for test well ET-3 and a conventionally drilled offset is shown in Figure 1. The overall ROP ratio of the UHP jet-assisted ROP to the ROP of the offset between 4,300 and 10,000 feet was 2.2. A second example, shown in Figures 2 and 3, is from test well ET-9. ET-9 was drilled after reliability of the UHP pumping and delivery systems had been improved to about 88 percent up-time. The offset ROP data is a composite average of the three fastest of eight wells drilled in the same area of the Carthage field by the same rig in the previous 12 months. The overall ROP ratio of UHP jet-assisted ROP to the offset between 4,500 and 10,000 was just over 1.5. As shown by the drilling curves in Figure 3 for ET-9 and the composite average offset, the increased ROP resulted in 4 days less time to drill the well.

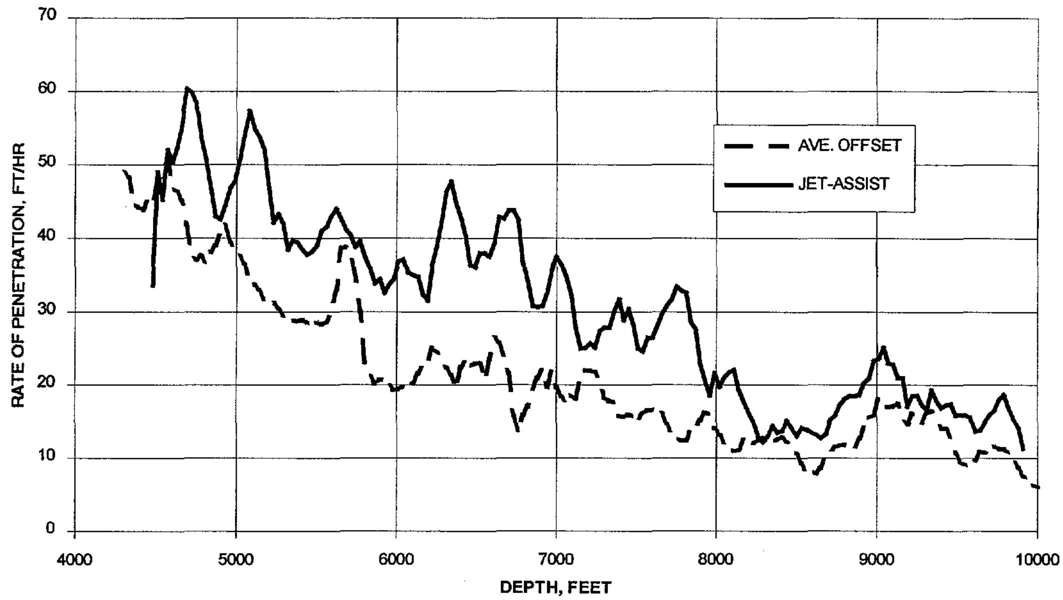


Figure 2. Ultra-High Pressure Jet-Assisted Rate of Penetration in a Gas Well in East Texas (ET-9) using the FlowDril System.

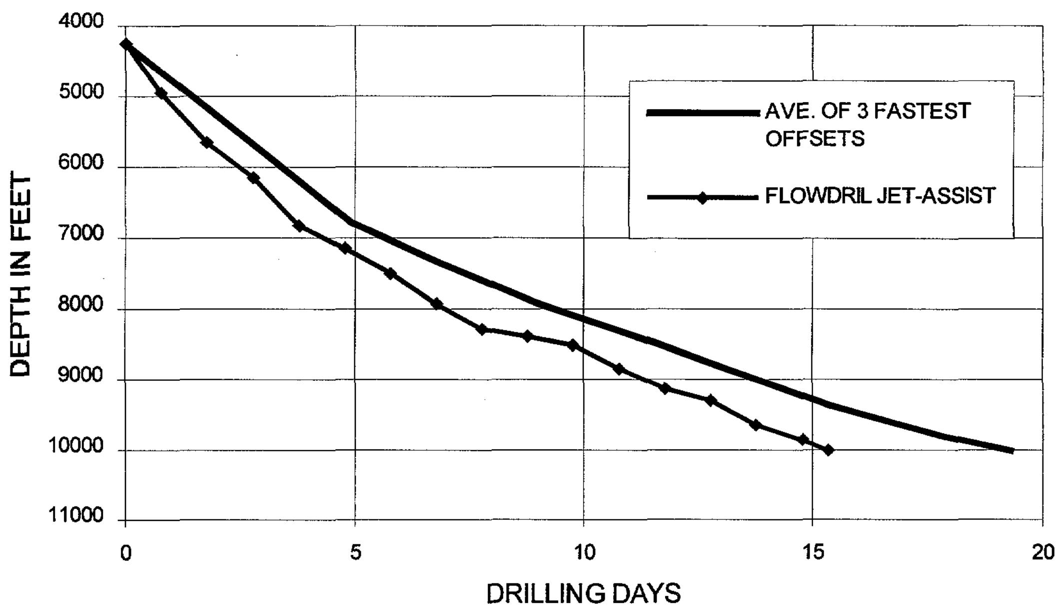


Figure 3. Drilling Curve from a Gas Well in East Texas (ET-9) using the FlowDril System.

Although the FlowDril system had demonstrated technical success and achieved marketable reliability, the high capital costs restricted commercial implementation to high day-sensitive cost drilling markets. It is believed that a downhole pump that develops the ultra-high pressure downhole near the bit would eliminate the need for the costly dual conduit system and enable UHP jet-assisted drilling technology to be more cost-effective across a broader portion of the drilling market.

1.2 The Downhole Pump Approach

In the downhole pump approach, conventional drill pipe and drill collars are used, with the DHP as the last component of the bottom hole assembly next to the bit, as shown in Figure 4. The DHP is about the same size as a conventional drill collar and is handled like a drill collar. The power required to drive the DHP is obtained from increasing the operating pressure of the surface pumps and delivering it downhole via the conventional mud stream. Estimated rig surface mud pump pressures necessary to drive the DHP are approximately 3,500 to 4,000 psi.

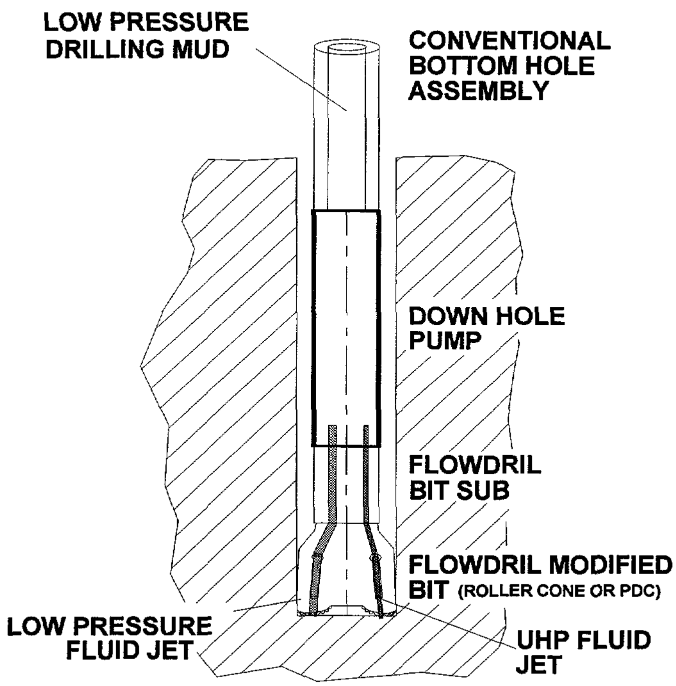


Figure 4. Jet-Assisted Drilling with a Downhole Pump (DHP®).

Within the DHP, the fluid stream is divided. Approximately seven percent of the stream is pressurized to 30,000 psi and directed through the high pressure nozzle on the drill bit to produce the high speed jet. The remainder of the fluid stream passes through the bit nozzles at pressures comparable to conventional bit nozzle pressure drops.

1.3 Intensifier Type Ultra-High Pressure Pump

The FlowDril DHP is a reciprocating, intensifier type ultra-high pressure pump. Flow International, a sister company to FlowDril, has manufactured and marketed this type of ultra-high pressure pump commercially for factory floor water-jet cutting applications since 1974. It is a fluid driven pump where the drive fluid is hydraulic oil. The output high pressure fluid is a non-abrasive fluid, generally filtered water. A schematic of an intensifier pump is shown in Figure 5. The drive fluid is a higher volume, lower pressure fluid that drives a large diameter low pressure piston. Connected to the large diameter piston is a smaller diameter plunger rod that when driven into the high pressure cylinder, pressurizes a smaller volume of fluid in the cylinder to higher pressure (up to 55,000 psi) and pumps it out through an outlet check valve. The ratio of the area of the larger diameter piston to the area of the smaller diameter plunger rod approximately determines the ratio of the output pressure to the drive flow pressure, and the ratio of drive flow rate to the output flow rate.

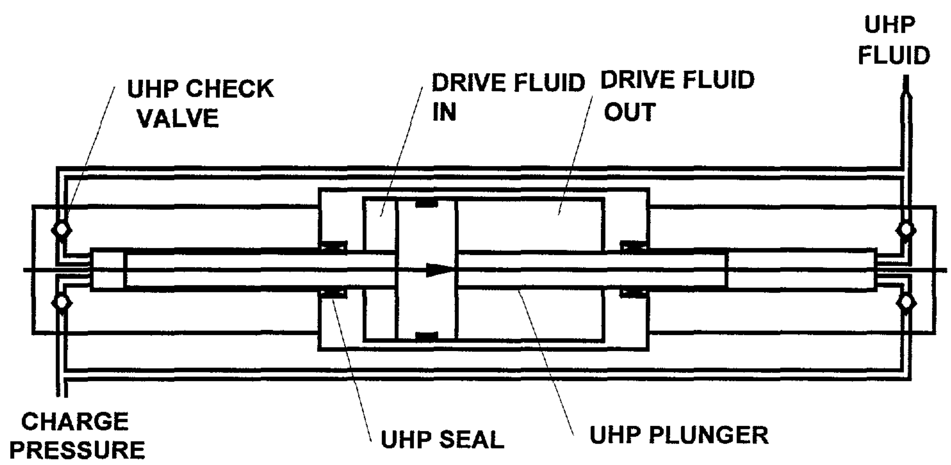


Figure 5. Intensifier Pump Principle.

As the low pressure piston is driven to the end of its stroke, a pilot valve is mechanically activated, which in turn shuttles a four-way directional main control valve (not shown in Figure 5). The main control valve, when shuttled, reverses the direction of low pressure flow within the intensifier, driving the low pressure piston in the opposite direction. When operating, the low pressure piston-UHP plunger assembly reciprocates back and forth. Output ultra-high pressure flow is produced in both directions. Because it is driven hydraulically, it is possible to achieve power conversion efficiencies between input hydraulic horsepower and output hydraulic horsepower of over 70 percent.

1.4 Downhole Pump Concept

Both the first prototype and the second prototype DHPs, described in later sections of this report, are reciprocating double ended, intensifier style positive displacement, high pressure pumps with the same operating principle as shown in Figure 5. A schematic of a downhole pump using this principle is shown in Figure 6. In the case of the downhole intensifier pump, the drive fluid and the high pressure output fluid are both derived from the same source, the abrasive drilling mud pumped downhole through the drill string.

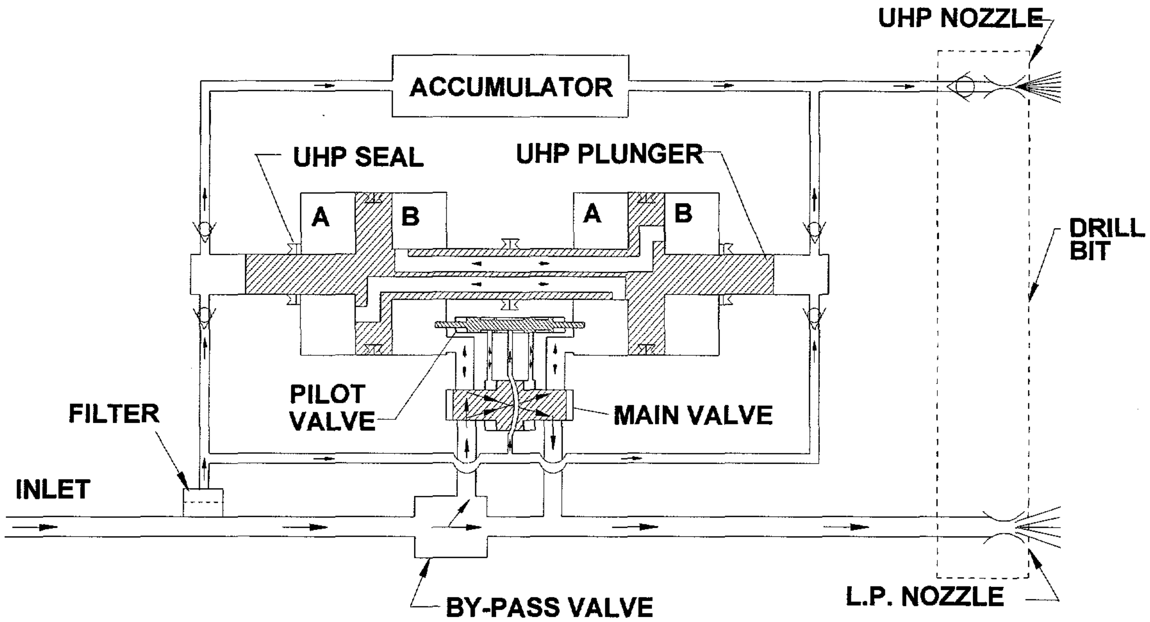


Figure 6. FlowDril DHP Concept.

Because of space limitations within the envelope of a drill collar, the downhole intensifier employs multiple low pressure drive pistons and fluid porting through the shaft connecting the drive pistons. Also shown in Figure 6 are the main control valve that directs the drive fluid to either chambers "A" or "B", and the pilot valve (also called trigger valve) that hydraulically shifts the main control valve. Other elements shown that are required for a practical downhole pump are a by-pass valve, an ultra-high pressure accumulator and a filter on the inlet to the UHP portion of the fluid circuit. The by-pass valve allows for circulation in the hole without the DHP operating, particularly in the event the DHP stalls. The attenuator maintains UHP flow during the non-pumping period at the end of stroke when the intensifier style pump changes direction. The filter reduces the size of particles allowed through the components in the UHP portion of the fluid circuit, minimizing the opportunity for plugging and reducing wear of the check valves and UHP nozzle.

1.5 GRI Development Program

Under the GRI project, a prototype 7-5/8 inch DHP for drilling 8-1/2 or 8-3/4 inch diameter holes was designed, fabricated, and a breadboard version tested in the laboratory. Subsequently, five field experiments were conducted with the 7-5/8 inch prototype during which seven down hole pump trials were conducted. Results of the field experiments with the 7-5/8 inch prototype have been presented by Veenhuizen, et al (1996). The downhole configuration of the 7-5/8 inch DHP for the field experiments is shown in Figure 7.

The assembly consisted of the bit, a bit sub, the DHP section, the attenuator collar and attenuator hanger sub, a float sub, and a screen sub. The hanger sub supports the UHP attenuator within the outer housing of the assembly. Connection to the drill collars above the pump assembly was usually with a crossover sub.

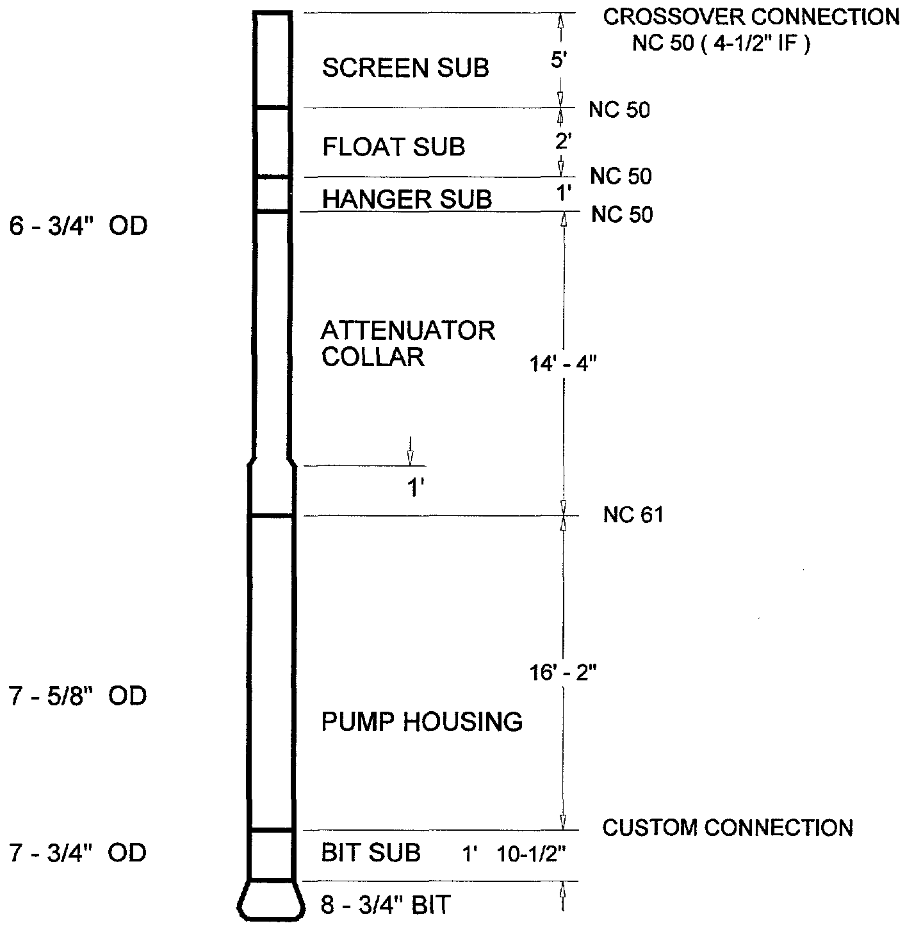


Figure 7. GRI 7-5/8 Inch DHP Field Experiment Configuration

The length of the assembly, not including the drill bit, was 40 feet 4-1/2 inches. The bit sub was the largest in diameter at 7-3/4 inches. The next largest in diameter was the DHP section at 7-5/8 inches. The top of the attenuator collar, the hanger sub, the float sub, and the screen sub were all 6-3/4 inches in diameter and could be fished with a full strength overshot. The top of the DHP section, however, was too large and would have required a custom slimhole overshot, or a spear to be fished. Had it been required, the entire assembly could have been washed over with a custom slimhole style washover pipe. All of the connections between sub sections of the assembly were standard API connections, except for the custom connection between the bit sub and the pump housing. This was a custom connection similar to that used in downhole mud motors.

A photograph of the 7-5/8 inch DHP is shown in Figure 8. It is from the first field experiment that was conducted at Amoco's Catoosa Drilling Test Facility in Oklahoma. Visible in the photograph is the 8-3/4 inch bit, the bit sub, and most of the lower portion of DHP section. The UHP extended nozzle is also visible on the bit.

A close up of the UHP jet-assisted drill bit is shown in Figure 9. It was a commercially available 8-3/4 inch Smith International F3 tungsten carbide insert bit to which the UHP extended nozzle tower and nozzle were added by FlowDril. The UHP nozzle tower extended from a UHP outlet in the bit sub and out through an existing mud nozzle port on the bit. The nozzle used was a 0.064 inch diameter proprietary nozzle developed by FlowDril.

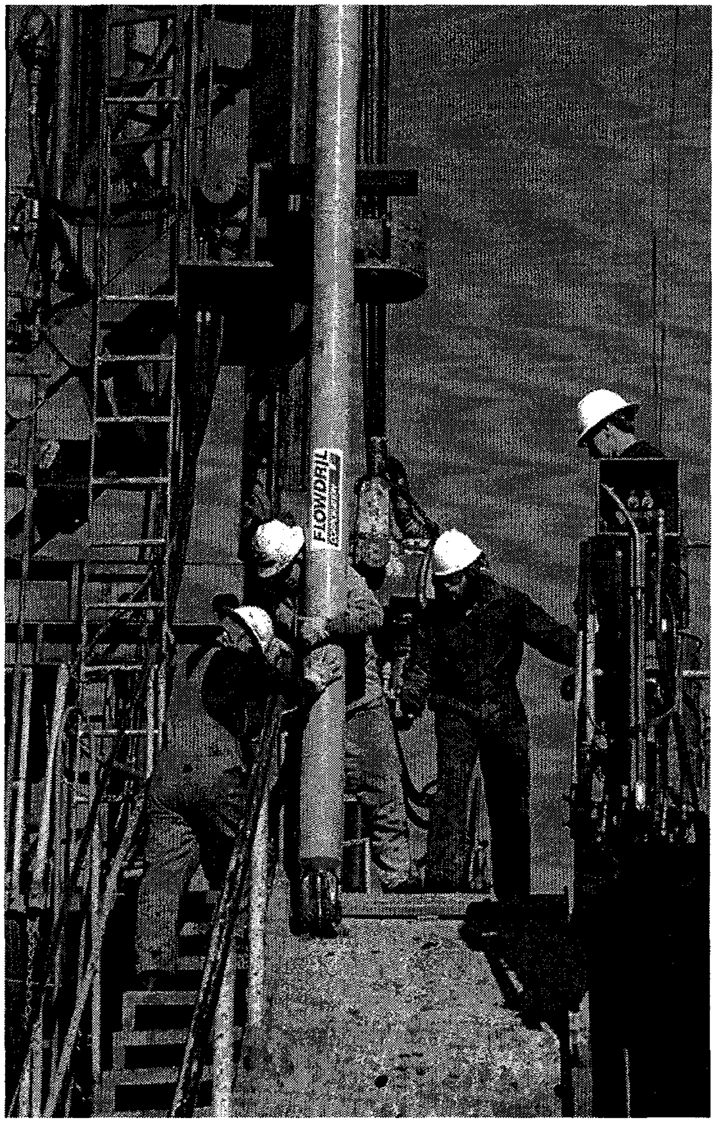


Figure 8. Photograph of First Prototype, 7-5/8 Inch, High Pressure Down Hole Pump during First Field Experiment under GRI Program

Figure 9. Smith F2 8-3/4 Inch Drill Bit with Extended UHP Nozzle used during First Field Experiment under GRI Program

2. OVERVIEW OF DHP DESIGN

Both the 7-5/8 inch and the 6-3/4 inch DHPs utilize the same basic concept shown in Figure 6. During the design phase for the 6-3/4 inch DHP, attempts were made to improve several components and several aspects of the design over the 7-5/8 inch DHP. These included increasing the stroking rate capability of the pump such that it could be operated at full specified stroke rate, improving the overall efficiency by reducing hydraulic losses, making it shorter as well as smaller in diameter, and improving the fishability of the tool. In particular, the mechanical design of the center connecting drive tube and connections through the drive pistons was changed significantly. The design outlined in this section generally describes the 6-3/4 inch second prototype DHP design. Whenever a significant difference between the first and second prototypes occurs, these differences are noted.

The second prototype was designed for 7-7/8 inch holes to better match the DHP market (see Phase I report). The general specifications for the 6-3/4 inch second prototype are given in Table 1 along with those for the first prototype.

Table 1. DHP Specifications.

	1st Prototype	2nd Prototype
Hole Diameter	8-3/4"	7-7/8"
Housing Length	33', 3-Piece	25', 1-Piece
DHP Max. O.D.	7-5/8"	6-3/4"
Area Ratio	14:1	14:1
Max. Stroke Rate	112 spm	120 spm
Max. Output Pressure	30,000 psi	30,000 psi
Max. Output Flow	23.7 gpm	21.4 gpm
Ave. Efficiency	70%	72%

2.1 DOE/FlowDril DHP Basic Arrangement

The basic arrangement of the components is shown in Figure 10 as they occur in the more recent 6-3/4 inch second prototype DHP design. At the lower end of the DHP, the fluid outlet end, is the bit sub (to which the drill bit is attached). In the bit sub and extending into the DHP is the UHP inner conduit (IC) that carries UHP fluid from the DHP to the drill bit. Above the bit sub is the pumping section of the DHP within the DHP housing. The pumping section consists of a lower and an upper end. The lower end of the pump generally consists of the lower UHP cylinder and two lower end low pressure drive chambers. Between the lower and upper drive chambers is the center valve (manifold) section where the by-pass valve and the DHP shifting control valves (pilot valve and main valve) are located. The upper end of the pump, as does the lower end, generally consists of the two upper drive chambers and the upper UHP cylinder. Above the pumping section and still within the housing are the shear filter, the UHP accumulator (also called an attenuator as in Figure 6), a float valve (or spacer), and a coarse inlet screen.

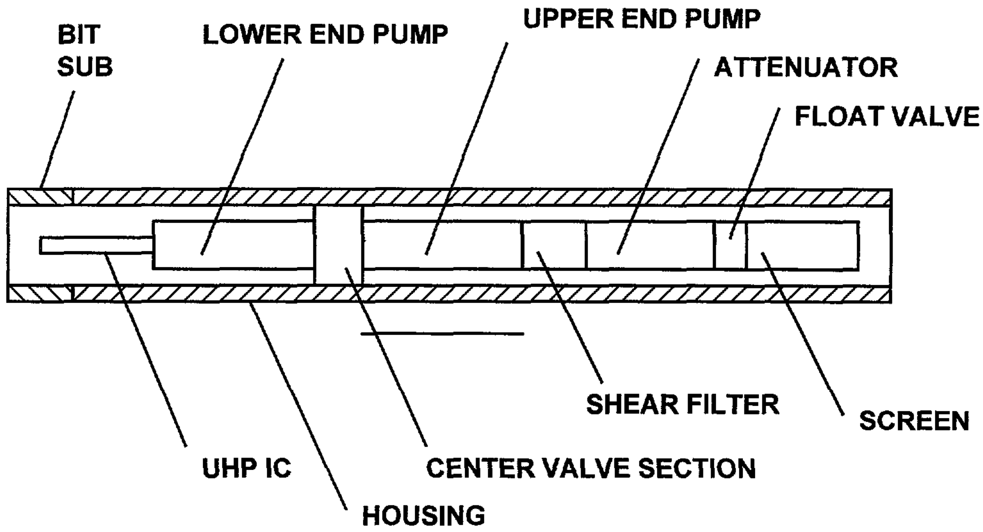


Figure 10. General Arrangement of 6-3/4" Second Prototype DHP.

During assembly, the UHP inner conduit, pumping section, shear filter, and UHP attenuator all slide into the housing from the bit sub end as a single assembled unit. This assembly is stabilized within the housing with three longitudinal centralizers located at equal spaces around the circumference of the assembly. The bit sub retains this assembly with a seat at the upper end of the attenuator against a shoulder within the housing. The coarse screen is inserted into the DHP housing from the upper end. In the 7-5/8 inch first prototype, the UHP attenuator was suspended in a separate housing with a hanger sub, and the float sub and screen sub were separate subs from the pumping section as shown in the comparison sketch in Figure 11.

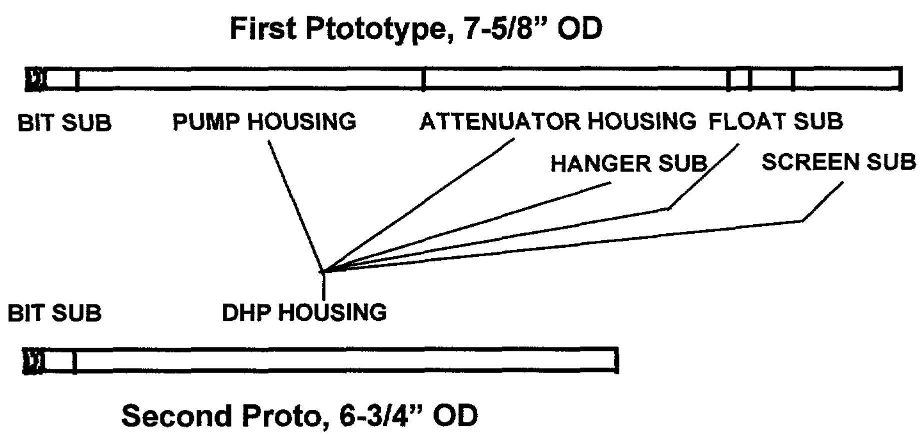


Figure 11. First Prototype Multiple Component Housing and Second Prototype Single Piece Housing Arrangements.

A schematic of the pumping section, consisting of the lower end of the pump, the center valve (manifold) section, and upper end of the pump is shown in Figure 12 for the first prototype and in Figure 13 for the 6-3/4 inch second prototype. The apparent difference is that the first prototype utilized three drive pistons, one toward the lower end of the pump, and two at the upper end. As indicated in Figure 13, there are four drive pistons in the second prototype. Each drive piston operates in a separate drive cylinder. The drive pistons are connected together with the connecting tube. The connecting tube also conveys fluid between drive cylinders and the main shifting valve, connecting chambers "A" to "A" and "B" to "B", as indicated conceptually in the sketch in Figure 6. Outboard of the drive pistons are the UHP plungers, one on either end of the pump section. The UHP plungers run in the UHP cylinders through the UHP dynamic seals. At the outlet end of the UHP cylinders are the UHP inlet and outlet check valves. The drive pistons operate hydraulically in parallel to generate the force conveyed through the connecting tubes to the UHP plungers to pressurize the UHP fluid in the UHP cylinders.

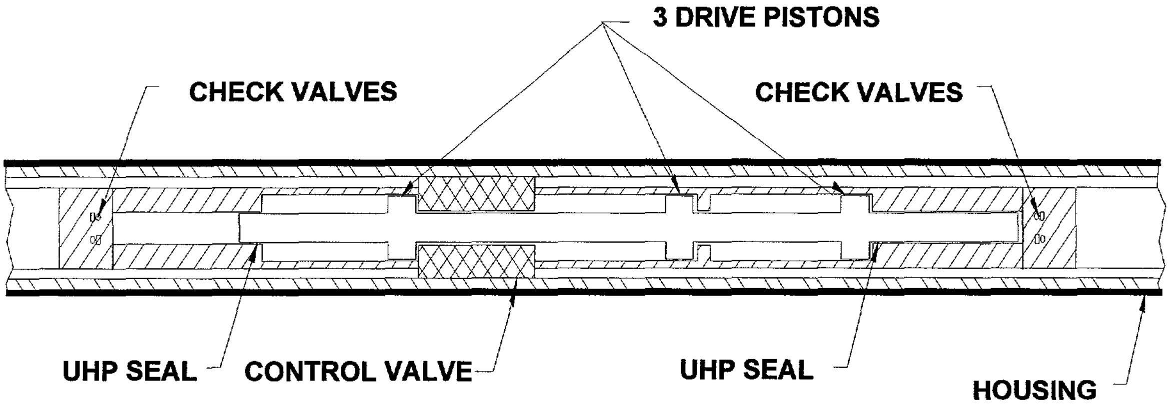


Figure 12. Pumping Section of the 7-5/8" First Prototype DHP.

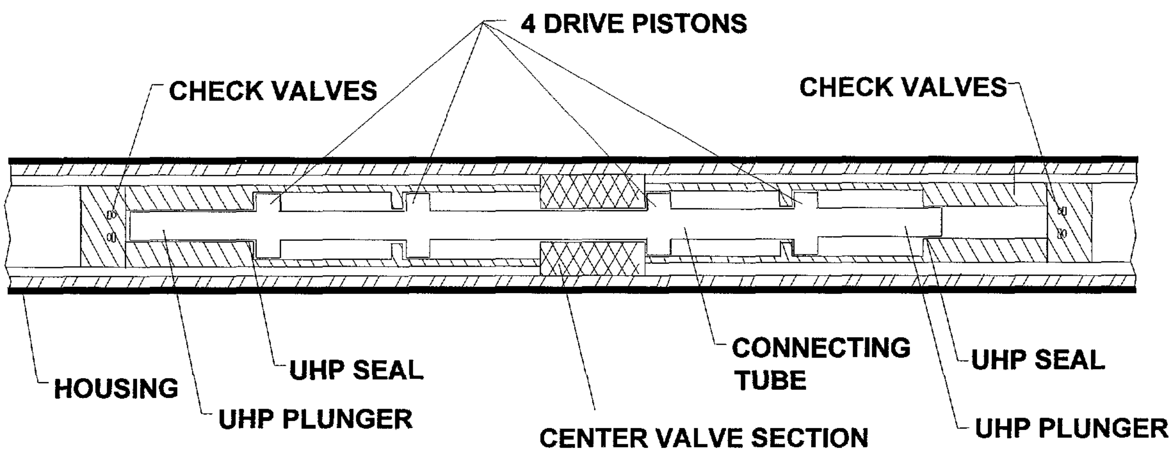


Figure 13. Pumping Section of the 6-3/4" Second Prototype DHP.

The lower pressure drive fluid pumped down the drill string from the surface enters the DHP from the upper end and encounters the coarse screen first. It then flows over the UHP attenuator, over the shear filter, over the upper end of the pumping section, and into the main control valve. The control directs the flow within the pumping section to drive the drive pistons either upward or downward. Fluid downstream of the drive pistons is exhausted through the main control valve downstream of the center valve section into the lower end of the pumping section. It then flows over the lower end of the pump, through the bit sub to the drill bit, and out the conventional nozzles of the drill bit. The inlet drive fluid is isolated from the lower pressure outlet fluid (exhausted fluid) by seals between the center valve manifold section and the housing. The inlet to the main control valve is upstream of these seals, and the outlet of the main control valve is below these seals.

As entering drive fluid flows over the shear filter, a small portion is drawn through the filter to supply the UHP fluid and the fluid used to hydraulically shift the main control valve. The UHP inner conduit, lower UHP cylinder outlet, upper UHP cylinder outlet, and UHP attenuator are connected by two UHP outer conduits passing through the annular region between the inner pumping section assembly and the outer housing and through the center valve section. A similar arrangement provides lower pressure fluid from the shear filter to the center valve section for shifting the main control valve and to the inlet to the lower end UHP inlet check valve and UHP cylinder.

2.2 Connecting Tubes

Running through the center of the pumping section of the DHP are the connecting tubes. They conduct drive fluid between the drive cylinders and the main control valve. They also transmit the force to drive the UHP plungers into the UHP cylinders. There are three of the them in the 6-3/4 inch DHP design: two outboard connecting tubes and one center connecting tube. The center connecting tube is longer than the outboard ones and passes through the center valve section.

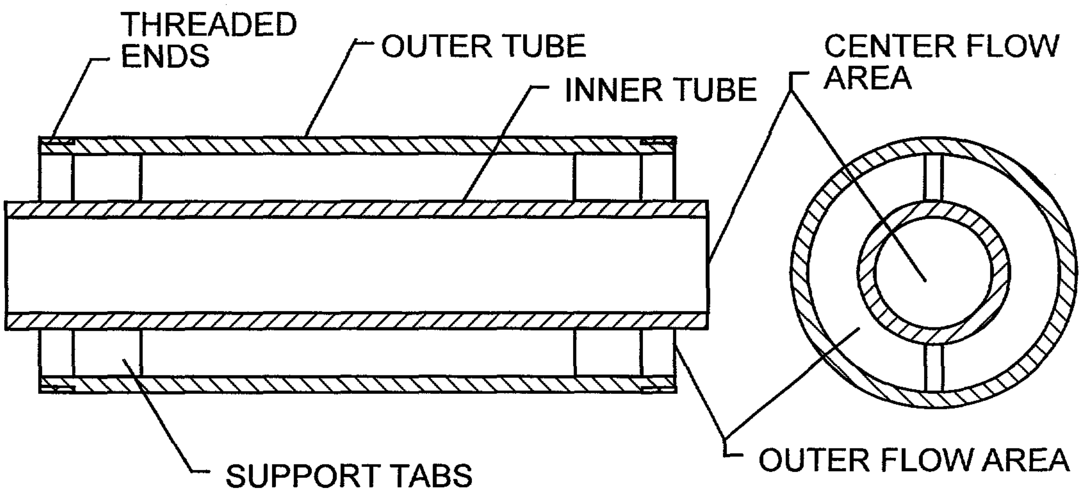


Figure 14. Connecting Tubes.

In the 6-3/4 inch second prototype, the connecting tubes consist of two concentric tubes, welded together at the ends with support tabs as shown in Figure 14. The bore of the inner tube is the center flow area for the passage that connects to the lower side of each drive piston and drive chamber to the main control valve. The annular region between the inner and outer tubes, the outer flow area, is the fluid passage that connects the upper side of each drive piston and drive cylinder to the main control valve. In each passage, fluid flows in either direction, depending on whether the pump is stroking upward, or stroking downward. The fluid flow direction in the outer flow area, however, is always in the opposite direction to the flow direction in the center flow area. The pressure, as does the fluid flow direction, alternates in each flow passage between drive pressure and outlet (exhaust) pressure. The outer tube transmits the force to drive the UHP plunger. The outboard connecting tubes carry the largest load, as it is accumulated for the pressure generated by three drive pistons.

The exterior ends of the outer tubes are threaded for connecting to the drive pistons. Initially in the 7-5/8 inch first prototype, a tie rod extending the length of the pumping section through both UHP plungers and the connecting tubes was used to hold the plunger/connecting tube/piston assembly together. The resulting restriction of the fluid flow through the inner connecting tube with the tie rod caused hydraulic pressure losses within the DHP to be larger than acceptable. The tie rod was subsequently replaced with a tie rod tube arrangement where the inner concentric connecting tube was used as the tie rod arrangement. The design of the connecting tubes depicted in Figure 14 was adopted for the 6-3/4 inch second prototype.

2.3 Low Pressure Drive Cylinders/Pistons/Seals

Attached to each end of each connecting tube is a low pressure drive piston. These drive pistons run in the low pressure liners, or cylinders. The arrangement is shown in Figure 15. The low pressure liners/pistons comprise the fluid drive cylinders that hydraulically drive the connecting tube - drive piston - UHP plunger assembly up and down (or back and forth) in the DHP.

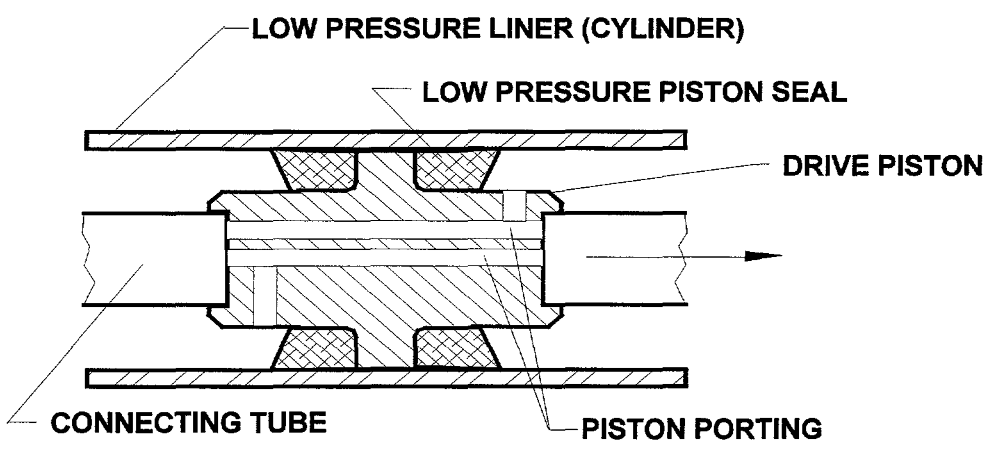


Figure 15. Low Pressure Cylinder/Piston/Seals

When drive fluid at drive pressure is on the upper side of each piston, the lower side of each piston is at outlet (exhaust) pressure. To seal these pressures from each other, seals mounted on the pistons are used. These dynamic piston seals run on the interior surface of the liners. Fluid passages are ported through the pistons to connect all the upper sides of the pistons together, and all of the lower sides of the pistons together. Porting in the end pistons is slightly different as they connect to the UHP plungers. The pistons are connected to the connecting tubes using a threaded connection. Between the two low pressure drive cylinders on both the lower pump end and upper pump end are liner separators with rod seals (not shown) through which the connecting tube travels. These liner separators support and align the ends of the low pressure cylinder liners as well as isolate each drive chamber from the others.

2.4 UHP Cylinders/Plunger/Seals

Outboard of the two end drive pistons are the UHP plungers. The end of the low pressure drive piston, the UHP plunger, and UHP cylinder arrangement are shown in Figure 16. The UHP plunger is retained in the drive piston with a threaded-in tie rod through the hollow UHP plunger.

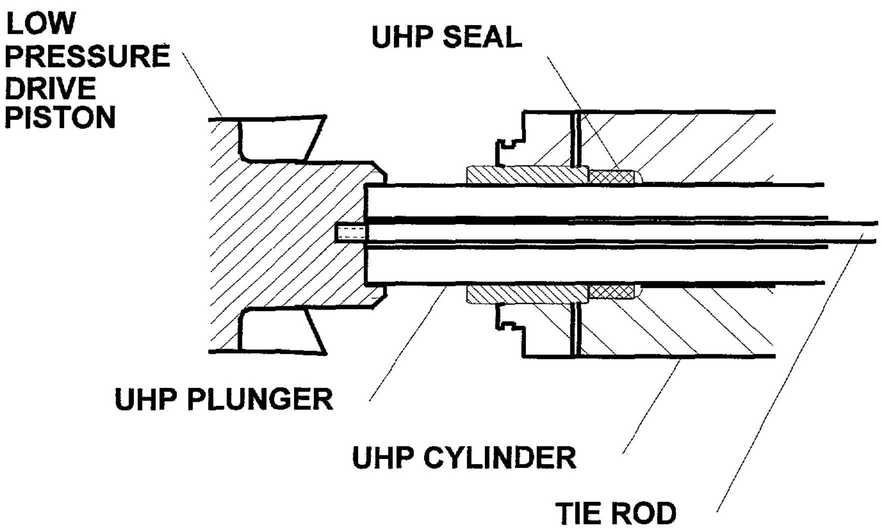


Figure 16. UHP Cylinder/Plunger/Seal

The UHP dynamic seals are located where the UHP plungers enter the UHP cylinders. Between the two outboard drive cylinders and the UHP cylinder is an end bell (not shown) through which the UHP plunger travels and which supports the UHP cylinder and the outboard end of the outboard low pressure cylinder liners.

2.5 UHP Check Valves

At the outboard end of each UHP cylinder is the UHP inlet/outlet check valve assembly. This is shown in Figure 17 with the outlet check valve open and the inlet check valve closed. The type of check valves used are ball and seat. Both the ball and the seats are of very hard materials. The seal is made with a narrow band of contact of the round ball in the cone shaped seat. The check valve housing is again a highly stressed component subject to the same pressure cycling as the UHP cylinders.

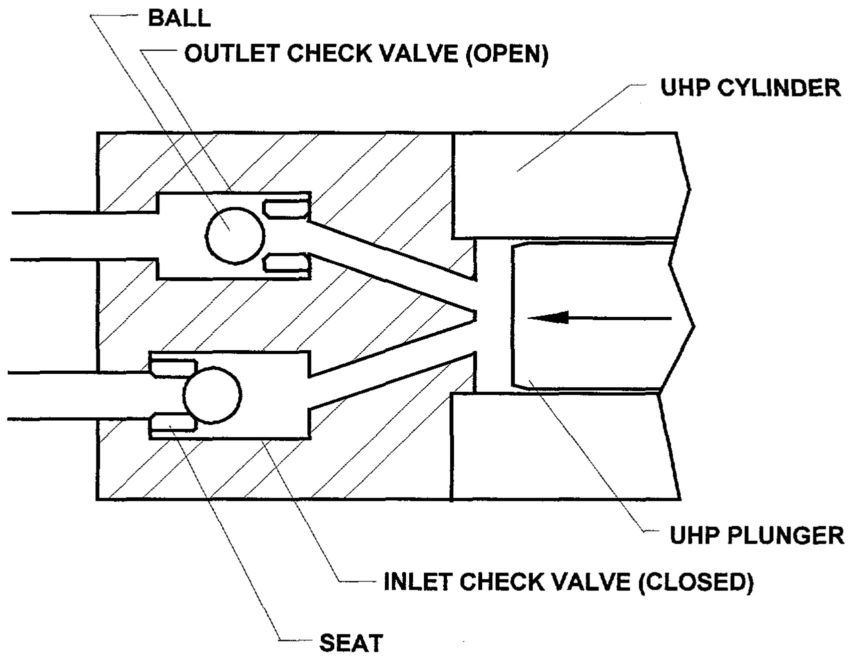


Figure 17. UHP Check Valves.

2.6 Center Valve Manifold Section

Separating the lower end of the pump from the upper end of the pump is the center valve manifold section. The center valve manifold section has three pieces: a lower manifold section, a middle manifold section, and an upper manifold section. The middle valve manifold has straight through bores for the main control valve, pilot valve, and by-pass valve as shown in Figure 18.

Around the outside circumference of the middle manifold are the seals between the center valve section and the housing of the pump that separate the inlet drive flow from the lower pressure outlet flow (exhaust) of the pump. It is across these seals that the pressure to drive the DHP is maintained.

The lower and upper valve manifolds have circumferential internal porting to connect the "A" and "B" main valve ports to the lower and upper low pressure drive cylinders. To fit the valve pieces and provide the porting in the center valve manifold sections required a design compromise between physically fitting the components within the available space, providing enough wall thicknesses to have the structural strength required, and yet providing fluid passages large enough to avoid large pressure losses and minimize fluid erosion. It is the fluid passages in the lower and upper valve manifolds that are the most susceptible to fluid erosion.

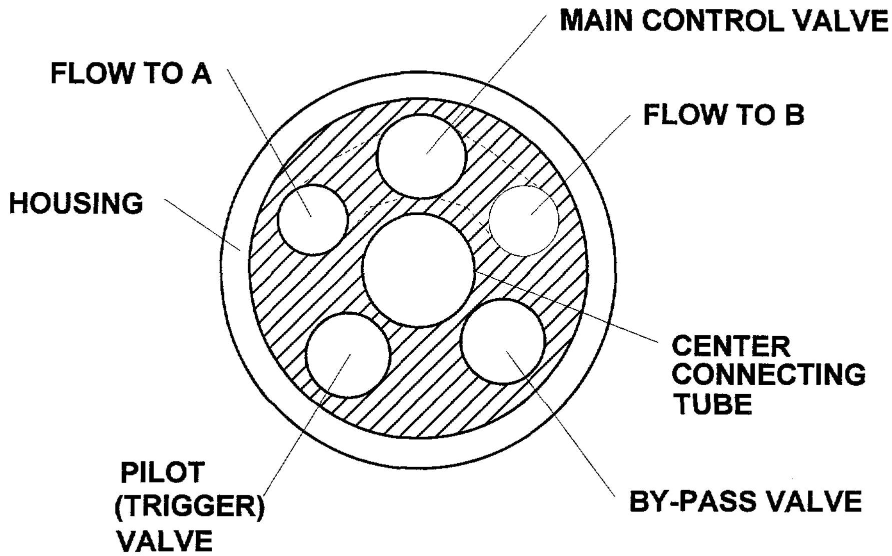


Figure 18. Center Valve Manifold Cross Section.

2.7 Main Valve

The main control valve controls the direction of the drive fluid within the DHP. It is located in the center valve manifold section. It is a four-way directional control valve that is pilot-operated. It is shown in Figure 19, shifted such that the inlet port, the pressure port, is open to "A" (see Figure 6 for "A" and "B" locations in the drive cylinders) and the tank port, or exhaust port, is open to "B." In the DHP, the drive pistons within the drive cylinders are between the pressurized "A" port and the "B" port open to tank (exhaust). When in the opposite position, the spools are moved across the pressure and tank ports opening inlet pressure to "B" and tank (exhaust) to "A" and the fluid flow direction and pump stroking direction are changed to the opposite direction. The effect of the valve is to re-direct the flow within the DHP. When the main valve goes across center, or is in the center position, the spools do not completely close off either the pressure port or the tank port. That is, the flow through the main valve and the pump is never blocked off, or stopped. In this case, the pressure port is momentarily connected directly to the tank port and inlet fluid is lost to the outlet (exhaust) side of the pump.

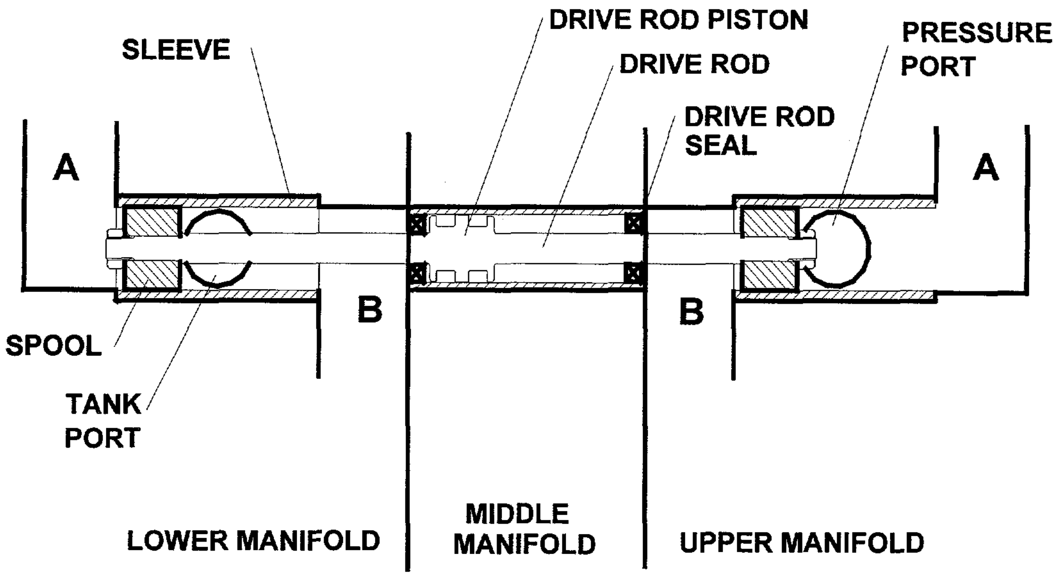


Figure 19. Four-Way Directional Main Control Valve.

During DHP operation, the number of main valve position changes is the same as the stroke rate of the DHP. The time required for the main valve to change positions (shifting time) affects the smoothness of the pump operation and the DHP efficiency. The shorter the shift time, the more efficient is the DHP. Calculated times for the complete travel of the main valve from pressure to port "A" to port "B" is about 25 msec. The entire flow through the DHP is re-directed in about 10 msec. The main pumping assembly in the DHP hydraulically stops and changes direction in about 15 msec. The main valve speed of travel is the highest of any moving component within the DHP. The valve is hydraulically shifted via the main valve drive piston. Fluid used for shifting the main valve is valved through the pilot valve and is part of the filtrate from the shear filter.

The valve spools and the sleeves are made of very hard materials to resist abrasion and erosion. The port sizes are of such size that the fluid velocities are high, causing the main valve and the center valve manifolds to account for over 50 percent of the hydraulic pressure losses through the DHP. As shown in Figure 19, the main valve spans all three sections of the center valve section: the lower manifold, middle manifold, and upper manifold. The valve sleeves are located in the lower and upper manifolds.

2.8 Pilot Valve

The pilot valve is also a four-way directional control valve. It controls the fluid that hydraulically shifts the main control valve. The pilot valve arrangement is shown in Figure 20. The valve actuation method is a spring loaded, mechanically initiated action, that when triggered, shifts the pilot valve spool between one of two positions; pressure (P) connected to port A and tank (T) connected to port B, or pressure connected to B and tank connected to A. The valve positions are bi-stable (either one position or the other) such that the DHP will always start under all conditions.

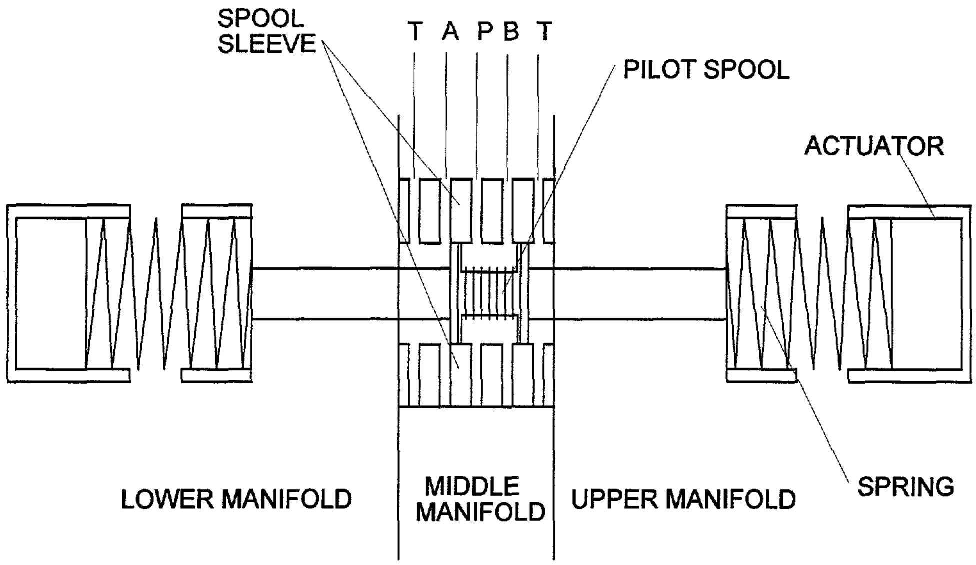


Figure 20. Pilot Valve (P open to B).

The actual valve portion of the pilot valve is located in the middle manifold. The actuators and springs are located in the lower and upper manifolds. During operation, the two interior drive pistons (as shown conceptually in Figure 6) contact the ends of the actuators and depress the springs. When the spring is compressed and the actuator sleeves bottom, the spring forces the spool to travel across, starting the shift of the main valve.

2.9 By-Pass Valve

The by-pass valve is also located in the center valve section. When in the open position, as shown in Figure 21, the inlet flow is connected to the outlet, by-passing the pumping section of the DHP. In the open position, flow may also enter the pumping section of the DHP and cause it to operate slowly at reduced output pressure.

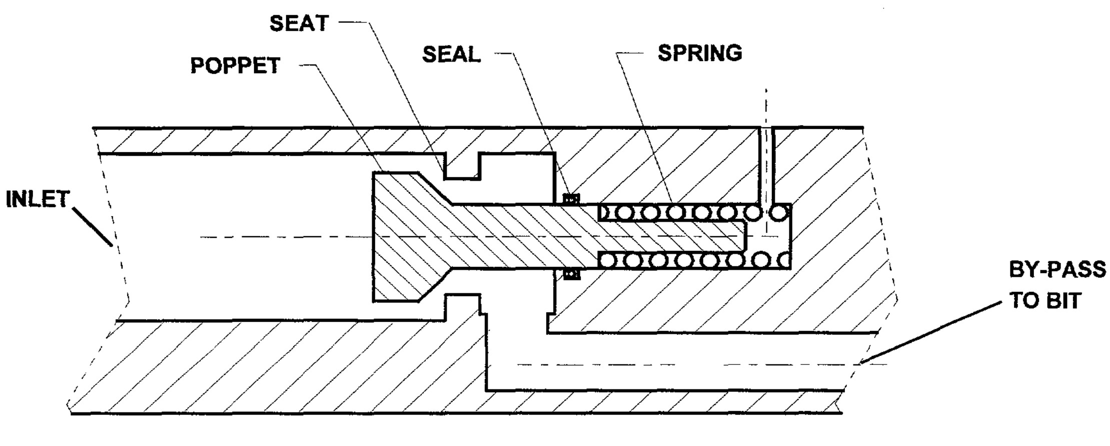


Figure 21. By-Pass Valve (open position).

The by-pass valve is flow-activated. Increasing the fluid flow to the DHP about 10 percent increases the fluid drag on the valve poppet, and forces the valve poppet against the spring, allowing the valve poppet to close off against the valve seat When the by-pass valve is closed, all the inlet flow to the DHP is diverted into the pumping section at the inlet to the main valve. The design includes adjustable stops to adjust the spring tension, thus allowing adjustment in closing flow rate to be made.

2.10 Shear Filter

Above the upper end of the pumping section is the shear filter. A portion of the inlet flow to the DHP is drawn off as filtrate through the shear filter to supply fluid to the pilot valve for shifting the main control valve, and to supply the inlet fluid to both UHP cylinders. All the UHP fluid is from the filtrate through the shear filter.

The shear filter itself is a screen tightly stretched over a slotted liner. It is shown schematically in Figure 22. The slots collect the filtrate after passing through the screen. Material screened is removed from the screen surface by the flow passing over the screen. Tests with an 80 mesh screen showed that approximately 9 percent of the inlet flow could be drawn off as filtrate even when the inlet flow contained 5 pounds of LCM (lost circulation material) per barrel (42 gallons).

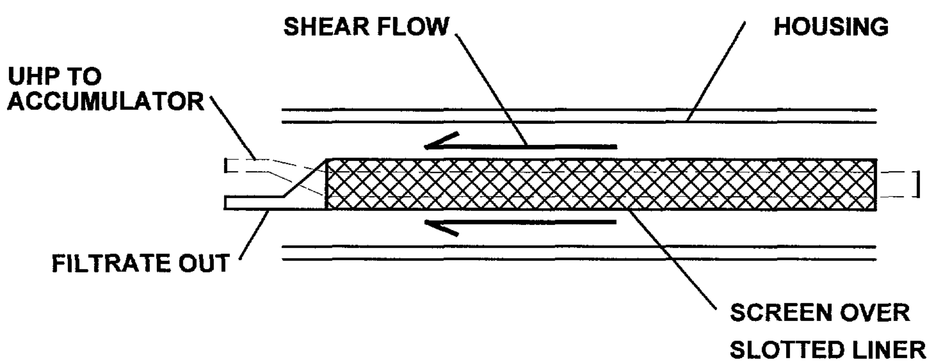


Figure 22. Shear Filter.

2.11 UHP Attenuator

The UHP attenuator (accumulator), shown in Figure 23, is located upstream of the shear filter and provides flow conditioning for the flow over the shear filter. Its two main purposes are to provide flow to the UHP nozzle when the DHP shifts direction and is not pumping, and to minimize the UHP pressure variation in those components external to the UHP check valves to minimize UHP fatigue problems. It is a dead volume attenuator with an enclosed volume of 2 gallons. It provides a volume of UHP fluid large enough that the compression and expansion of the fluid itself provides a sufficient volume of fluid to the UHP jet to maintain the flow of the jet within 10 percent of rated pressure (30,000 psi) during the time when the DHP is not pumping.

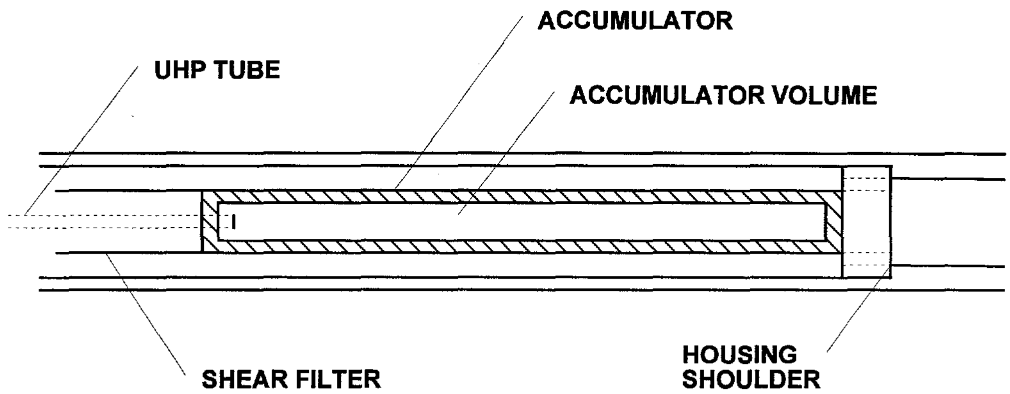


Figure 23. UHP Accumulator (Attenuator).

2.12 DHP Housing

The UHP inner conduit, pumping section, shear filter, and UHP attenuator all slide into the housing from the bit sub end as a single assembled unit. The housings for both the 7-5/8 inch first prototype and the 6-3/4 inch second prototype are shown in Figure 24 with the bit and bit sub attached.

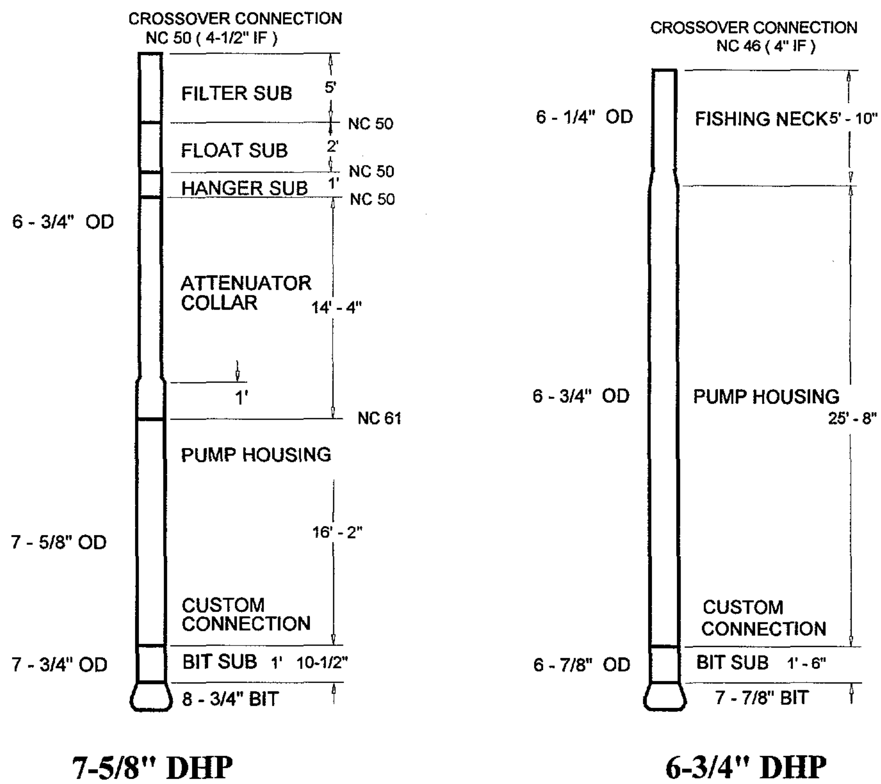


Figure 24. First and Second Prototype DHP Housings.

The connection for the bit sub is a custom API connection that is stronger and has a higher bending strength ratio than the standard NC50 (4-1/2 IF) or the NC46 (4 inch IF) connection at the upper end. The wall thickness of the pumping section of the housing for the 7-5/8 inch first prototype was 11/16 inches and 3/4 inch for the 6-3/4 inch second prototype. Bending stiffness of the housing closely matches a 6-1/4 inch OD standard drill collar in both cases. To minimize the potential for fishing problems, the 6-3/4 inch second prototype was designed as a single piece housing without multiple connections and with a fishing neck at the upper end. As shown, the 5 foot - 10 inches of the housing is a 6-1/4 inch OD fishing neck to allow use of slimhole or semi-full strength overshots in a 7-7/8 inch diameter hole, such as several of the Bowen Overshots - Series 150. For washing over just the DHP, a Tri-State Oil Tool washover pipe, 7-5/8 TSWP would be appropriate.

2.13 Bit Sub

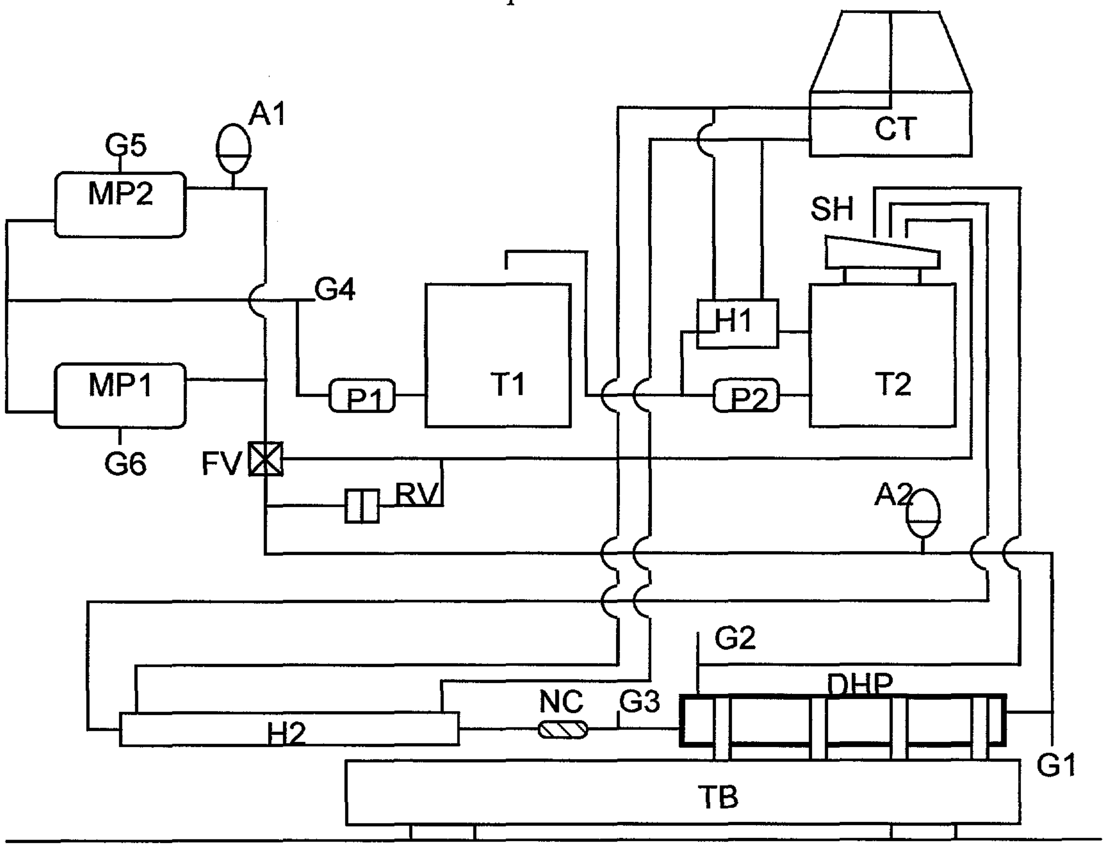
Between the bit sub and the housing is the only joint in the 6-3/4 inch DHP. The UHP fluid is delivered to the bit sub via the UHP inner conduit (IC) that exits the lower end of the pumping section. The lower end of the UHP IC is a female stab connection for the UHP tubing normally installed on the drill bit. A retainer ring is used to pre-load the pumping section, the shear filter, and the UHP attenuator in the housing under a compressive load. It is this pre-load that holds the pumping section, shear filter and UHP attenuator assembly together inside the housing during operation.

3. DHP LABORATORY TESTING

Over 50 laboratory test runs were conducted, accumulating 480 hours of laboratory test time on the second prototype DHP. A number of proprietary design improvements were introduced during laboratory testing to address mechanical strength and dynamic loading issues, low cycle fatigue, fabrication difficulties, and both drive fluid and UHP sealing issues. The second prototype was an improvement over the first prototype with better relative performance and improved hydraulic efficiency. Near the conclusion of laboratory testing, the second prototype had achieved run times of about 40 hours with laboratory drilling mud.

3.1 Laboratory Test Arrangement

The test arrangement and laboratory test facility is shown in Figure 25. Laboratory testing of the DHP was conducted with it in the horizontal position.



TB - DHP Test Bench	
G1 - Inlet L.P. Drive Pressure Gage	MP2 - SPM Mud Pump
DHP - Downhole Pump	G5 - Stroke Counter
G2 - Outlet L.P Drive Pressure Gage	A1 - First Accumulator
G3 - UHP Output Pressure Gage	G4 - Mud Temperature
NC - UHP Nozzle and Jet Catcher	P1 - Centrifugal Charge Pump 1
H2 - Heat Exchanger for UHP Fluid	T1 - Mud Tank 1, Supply
A2 - Second Low Pressure Accumulator	SH - Shale Shaker
FV - System Flow By-Pass Valve	P2 - Centrifugal Charge Pump 2
RV - Drive System Relief Valve	H1 - Heat Exchanger Mud Tanks
MP1 - Gardner Denver Mud Pump	T2 - Mud Tank 2, Return
G6 - Stroke Counter	CT - Cooling Tower

Figure 25. Laboratory Test Arrangement for Testing DHP.

Pressure gages were installed at the inlet (G1), at the outlet (G2) ahead of a back-pressure orifice, and on the UHP output (G3) ahead of the UHP nozzle to monitor testing of the DHP. During part of several of the runs, electronic pressure transducers were installed at these locations to record pressure signals with the laboratory digital data acquisition system. Flow rate was determined by a stroke counter (G5) on MP1 and an rpm meter (G6) on MP2. Near the end of testing, a magnetic flow meter was installed. Temperature was monitored with a gage (G4) mounted in the laboratory plumbing between mud tank 1 (T1) and the inlet to the mud pumps. The stroking of the DHP was audible and the stroke rate was determined by counting against a stop watch. General run parameters during laboratory testing of the DHP are listed in Table 2. The UHP output nozzle was generally a 0.064 inch diameter FlowDril nozzle.

Table 2. Second Prototype Performance Test Parameters.

Max. UHP Output Pressure	28,000 - 31,000 psi
Inlet Drive Pressure	2870 psi
Outlet Pressure	450 psi
Inlet Drive Flow Rate	270-275 gpm
Mud Weight	8.7 ppg
Mud Temperature	120°F

3.2 Second Prototype (6-3/4" DHP) Operating Characteristics

Pressure signals were monitored and recorded via the laboratory data acquisition system at the DHP low pressure drive flow inlet (G1), at the DHP outlet (G2) just upstream of the back-pressure orifice (exhaust from the pump), and at the UHP outlet just upstream of the UHP nozzle (G3). Some of the data illustrates the operating characteristics of the DHP. The pressure transducers had a response time of 1 msec. For the pressure signals shown in the following figures, the sample rate per signal was 1,000 samples per second.

Input, Outlet Drive Pressures

A two second long sample of the recorded pressure at the inlet is shown in Figure 26. Four shifts of the main valve were captured in the recorded pressure signal.

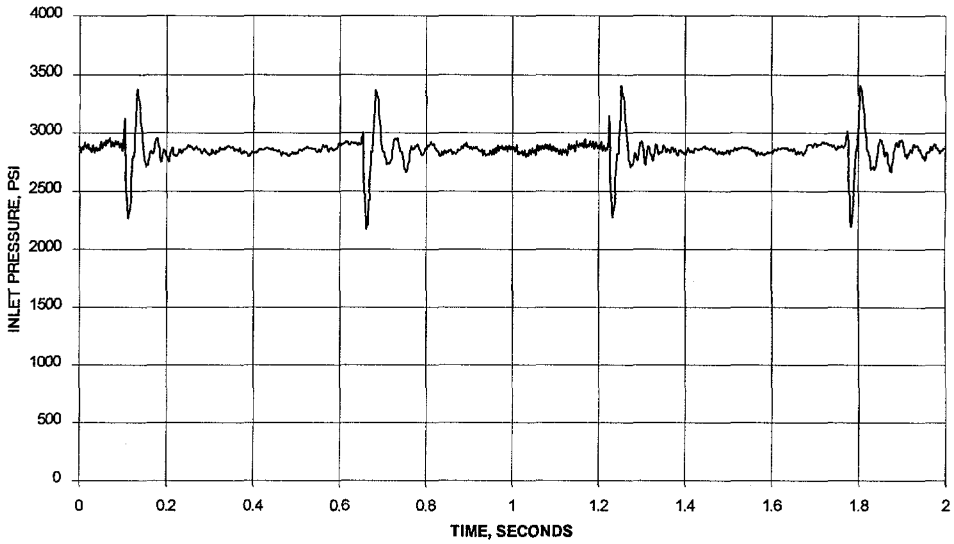


Figure 26. Inlet Low Pressure Flow Pressure Signal.

Each shift of the main valve corresponds to a change in direction of the drive piston/plunger assembly within the DHP. Time between stroke changes shown was approximately 545 msec, corresponding to a stroke rate of 110 SPM. The inlet pressure signal is characterized by an approximate 100 to 200 psi sharp rise, followed by a dip of about 620 psi, followed by a rise of about 520 psi. This pressure would propagate uphole toward the surface in a downhole configuration. As it would propagate toward the surface, it would be damped by the mud viscosity. At depths greater than about 4,000 feet, the pressure fluctuations were not readily detectable on the surface.

The low pressure flow pressure signal at the outlet of the DHP (exhaust flow), G2, is shown in Figure 27. It was recorded at the same time as the inlet pressure signal shown in Figure 26. In the downhole configuration, this would be the pressure signal just before the low pressure nozzles in the drill bit. Corresponding to the pressure dip apparent in the inlet pressure signal, there is a sharp pressure rise of about 1,200 to 1,400 psi in the low pressure flow outlet. This occurs because of the blowdown effect from a large accumulated volume upstream (the supply line) to a smaller volume (in the bit sub) with a fixed outlet orifice as the main valve shifts across center and momentarily connects the inlet directly to the outlet of the DHP through the main valve.

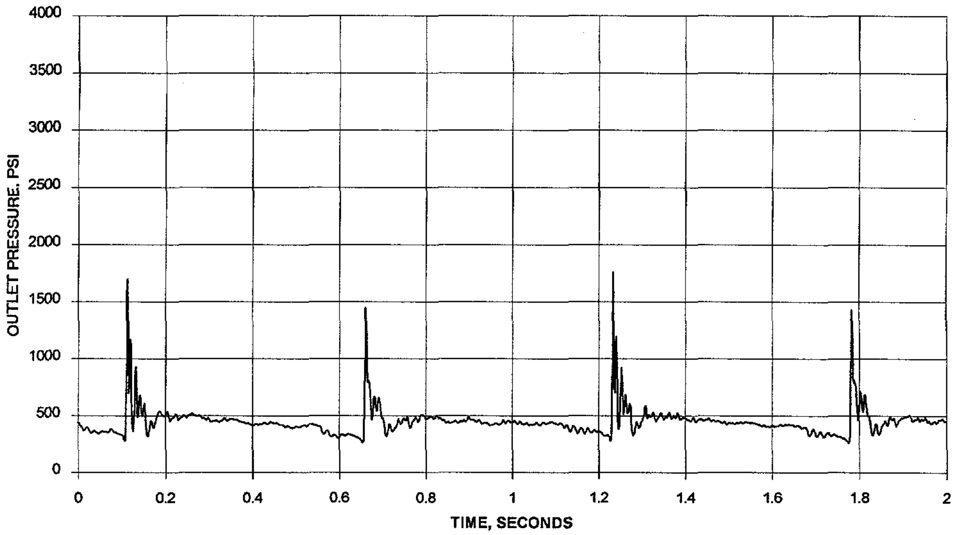


Figure 27. Outlet Pressure Signal.

The difference between inlet pressure and outlet (exhaust) pressure is the differential pressure across the DHP that drives it. This pressure signal is shown in Figure 28. Included in this pressure are the net drive pressure across the low pressure drive pistons that drive the UHP plungers and the pressure loss that occurs from the fluid flowing through the valves and flow passages of the pump. The differential pressure signal is typical of hydraulically driven intensifier type high pressure pumps. The large pressure dip is due to the necessity to re-pressurize and re-compress the inlet UHP fluid in the UHP cylinder before it is pumped out of the UHP cylinder through the outlet check valve. With the exception of the rapid pressure rise after the characteristic dip, it matches theoretically calculated differential pressure across the pump very well. Theoretical design calculations had estimated the characteristic pressure dip to be between 1,700 and 1,800 psi. The rapid pressure rise after the dip is thought to be a combination of a negative wave reflection of the pressure dip from the accumulator, A2, in the laboratory supply system.

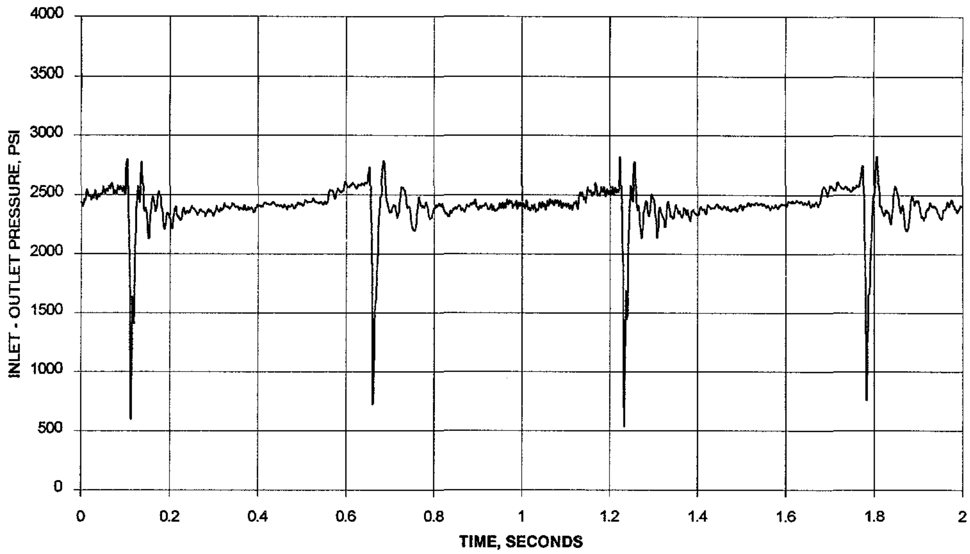


Figure 28. Inlet minus Outlet Pressure Signal.

UHP Output Pressure

The corresponding UHP output pressure signal is shown in Figure 29. The peak UHP output pressure shown is 29,660 psi. As can be seen, the UHP output pressure is not constant, but varies over time, fluctuating between a minimum of 27,450 psi and a maximum of 29,660 psi.

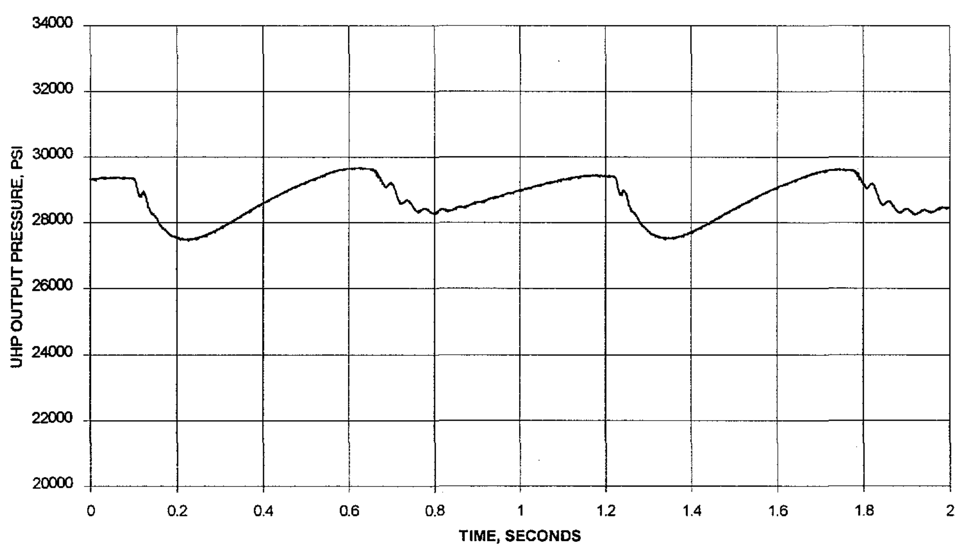


Figure 29. UHP Output Pressure Signal.

The average UHP output pressure calculated from the recorded pressure signal was 28,720 psi, or 0.968 of the peak (rated) UHP output pressure. Design of the UHP accumulator at full output flow rate called for the variation between maximum and minimum UHP output pressure at a rated output (maximum value) of 30,000 psi to be no more than 3,000 psi. It may be noted that adjacent peaks are not the same, but every other one is the same. This occurs because the UHP accumulator is located at the upper end of the pump and transmission losses from the upper end are larger than from the lower end where the outlet is located. Consequently, the higher UHP pressure peaks correspond to the pump stroking from the upper end to the lower end and pumping UHP fluid out of the lower end UHP cylinder.

Stroke Rate and UHP Output Flow Rate

During the laboratory run, the stroke rate of the DHP was counted against a stop watch. At an input flow rate of 270 gpm, the stroke rate was counted as 110 strokes per minute (spm), or 0.93 of the ideal stroke rate. From the displacement relationship, at this input flow rate, the ideal stroke rate would be 118 spm. The difference, 8 spm, is representative of volume loss as the main valve goes across center and UHP compressibility losses. It also indicates that almost the entire stroke length (displacement) available mechanically within the DHP was being utilized.

Rated UHP output flow (maximum achieved during latter part of stroke) was calculated using the measured pressure and known nozzle discharge coefficient as 18.0 gpm. As UHP pressure varies in time, so does UHP output flow rate according to flow through a fixed size orifice. The average UHP output flow was 17.3 gpm, or 0.98 of the peak flow. At 120 spm, the rated UHP output flow was estimated to be 19.6 gpm. This agrees with the performance specification at 120 spm within two percent.

Average inlet hydraulic horsepower (hhp) was 450 hhp. Average outlet (exhaust) horsepower was 65 hhp. Average net input horsepower to produce the UHP output horsepower was then 385 hhp. Average output UHP horsepower was 296 hhp. The average UHP output hhp to the average net input horsepower gives the average DHP efficiency as 77 percent. When this latter estimate is adjusted to 120 spm and 9.5 ppg mud weight, the projected efficiency was 75 percent.

3.3 Second Prototype (6-3/4" DHP) Laboratory Development Testing

The general approach to testing and development was to iterate between laboratory testing, downhole field experimentation and redesign efforts to evolve the design of the DHP. Over 50 laboratory test runs were conducted, accumulating 480 hours of laboratory test time on the second prototype DHP. A number of proprietary design improvements were introduced during laboratory testing to address mechanical strength and dynamic loading issues, low cycle fatigue, fabrication difficulties, and both drive fluid and UHP sealing issues. The second generation prototype was an improvement over the first prototype with better relative performance and improved hydraulic efficiency.

The hours for each of the laboratory test runs with the second prototype are shown in Figure 30. Most, but not all, of the test runs were to failure. The longest runs, 45 and 35 hours, occurred near the end of testing. All the testing was performed with laboratory drilling mud. Field experiments for the second prototype were FE-6 through FE-10, and are marked on the chart.

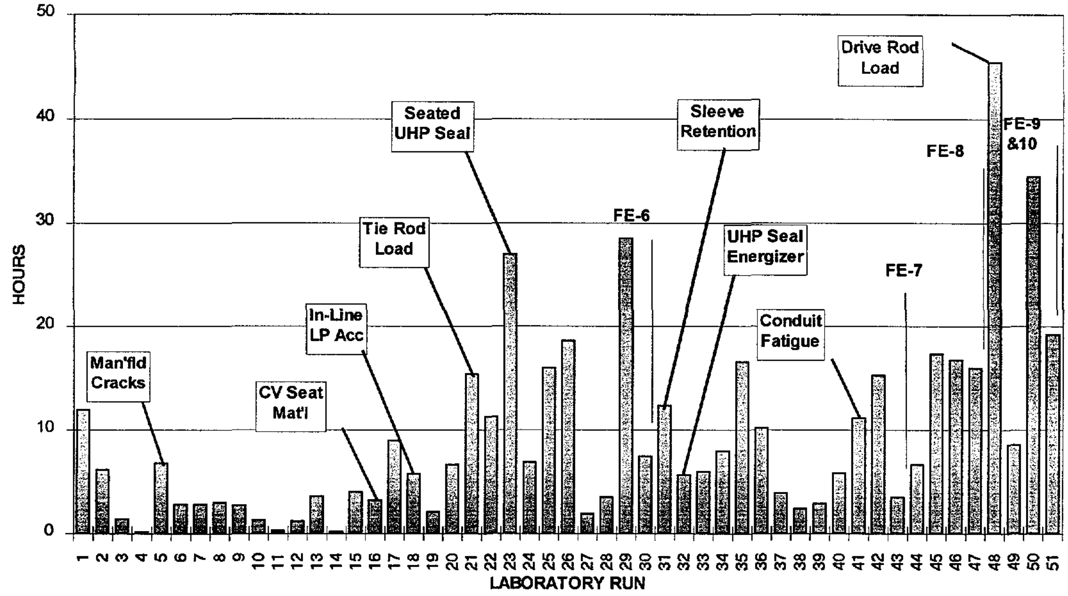


Figure 30. Laboratory Testing of 6-3/4" DHP.

Listed in Table 3 are the most frequent failures during laboratory testing of the second prototype with the number of times that they occurred. Several of the runs in Figure 30 are flagged where significant changes and/or improvements were introduced to the DHP design to address the failures. The first two flags correspond to failures that were irritating and time consuming, but not technically significant. The first was establishing quality welds of plugs in cross ports of the valve manifolds that cracked and eroded during the first several runs. The second flag, through run 16, corresponds to obtaining UHP check valve seats made from the correct material specification.

Table 3. Second Prototype DHP Laboratory Failures.

UHP dynamic seal (7)
CV lower seat (7)
LP dynamic seals (6)
Drive rod parted (4), nut backoff (1)
Tie rod parted (5)
Sleeve, retention (4)
Piston Dynamic Seals (4)
Manifold, lower, weld eroded (3)
UHP Conduit Fatigue (3)
Manifold, upper, eroded (1)

Although introduction of the modified proprietary UHP seal geometry during testing of the first prototype improved UHP seal reliability, it was still not sufficient. Testing with the second prototype indicated that the UHP seal was experiencing significant motion with each stroke of the DHP causing a deterioration of the UHP seal. To stop motion of the UHP seal required the introduction of a low pressure attenuator to the lower UHP end of the DHP, an increase of the pre-load on DHP components within the housing, and the introduction of a seal seating scheme to secure the UHP seals in position. A later modification, introduced for test run 32, that further improved UHP seal performance was a stronger seal energizing member.

A significant conceptual change in the DHP design between the first and second prototypes was the change from the tie-rod tube to the concentric connecting tube design (see Section 2.2). This resulted in higher than expected loads on the plunger tie rods (see Section 2.4) and caused several low cycle fatigue failures of the plunger tie rods. A subsequent port modification to the upper and lower out-board drive pistons reduced this load to acceptable levels.

The method of retaining the main valve sleeves (see Section 2.7) in the upper and lower valve manifolds was a problem carried from the first prototype. Methods of retaining the main valve sleeves were experimented with during the laboratory testing, including welded tabs, brazing, and mechanical retention schemes. An alternative method of retaining the sleeves in the valve manifolds was eventually developed through separate laboratory testing and implemented in the DHP.

As a result of the processing technique employed to reduce abrasive wear of the UHP conduits that conduct UHP fluid from the upper end to the lower end of the DHP, the conduits experienced hydrogen embrittlement and became subject to low cycle fatigue failure. This occurred several times before the processing problem was corrected.

The last item flagged in Figure 30 is the main valve drive rod loading. To allow the 6-3/4 inch second prototype DHP to perform at 120 SPM and full specified output flow rate again required speeding up the travel speed of the main valve. This increased the loading on the drive rod and resulted in low cycle fatigue failures of the ends of the drive rod where the spools attach (see Section 2.7). Subsequently the main valve drive rod was redesigned with a larger cross section rod to lower the maximum stress level.

A number of other design improvements were also introduced into the second prototype during laboratory and field testing to improve reliability. These included resolving low cycle fatigue issues where the inner and outer connecting tubes were welded together and the connecting tube connection to the low pressure drive pistons. Low pressure dynamic seals on the drive pistons and on the connecting tubes were improved with better backup rings and improved seal cross section geometries; however, the abrasive wear on these seals remains a concern.

A longer travel pilot valve was introduced with the second prototype that resolved the spring fatigue issue experienced with the first prototype. A higher speed main valve was also implemented to achieve full 120 SPM operation and was successful once the drive rod redesign had been completed. Although the pressure losses through the DHP increased because of several of the design improvements introduced during laboratory testing, the overall DHP efficiency was still 70 percent at the specification design point.

Design and reliability issues listed in Table 3 remaining to be addressed are abrasion and wear of the low pressure (LP) dynamic seals on the connecting tubes and of the piston dynamic seals; and erosion of the valve manifolds, particularly the upper manifold.

4. DOWNHOLE FIELD EXPERIMENTS

Five field experiments were conducted with the 6-3/4 inch second prototype, all in 1996. They were designated FE-6 through FE-10. Field experiments with the 7-5/8 inch first prototype were designated FE-1 through FE-5. A total of 75 downhole test hours was accumulated with the second prototype over eight DHP runs, with the longest run of 17 hours occurring on drilling mud in a well in East Texas. Results of the field experimentation with the second prototype DHP were also reported in Veenhuizen (1997b) and Veenhuizen et al (1997c).

4.1 Second Prototype DHP Field Experiments

General information as to when, where, and with whom the downhole field experiments were conducted are given in Table 4. Downhole experiments FE-6, FE-7 and FE-8 were conducted in East Texas in the Travis Peak formation at depths between about 6,200 and 7,300 feet. As described by Watson et al (1996), this formation generally starts at depths between 6,300 and 6,500 feet and is composed of laminated sections of shale, some limestone, sandy mudstones, muddy sandstones, and sections of very abrasive fine-grained sandstone. Downhole experiments FE-9 and FE-10 were supported by Statoil of Norway and conducted in a test well operated by RF-Rogaland Research in Stavanger, Norway. The DHP tests were at a depth of about 4,100 feet (1,250 meters) in a very abrasive granite formation.

Table 4. Field Experiments with Second Prototype.

No.	Date	Location	Operator	Contractor
FE-6	Mar-96	Carthage, TX	UPRC	Nabors Drilling
FE-7	June-96	Carthage, TX	UPRC	Nabors Drilling
FE-8	July-96	Carthage, TX	UPRC	Nabors Drilling
FE-9	Oct-96	Stavanger, Norway	Statoil	RF-Rogaland
FE-10	Dec-96	Stavanger, Norway	Statoil	RF-Rogaland

The DHP, as it was tested downhole in the field experiments, was shown in Figure 24 of Section 2. Overall length, not including the drill bit, was 33 feet. The top 5 feet 10 inches was a fishing neck 6-1/4 inch in diameter. The remainder of the pump housing is 6-3/4 inch in diameter. A short bit sub, 1 foot 6 inches in length, just above the bit, was 6-7/8 inches in diameter. A photograph of the DHP being picked up on the rig during FE-9 in Norway is shown in Figure 31.

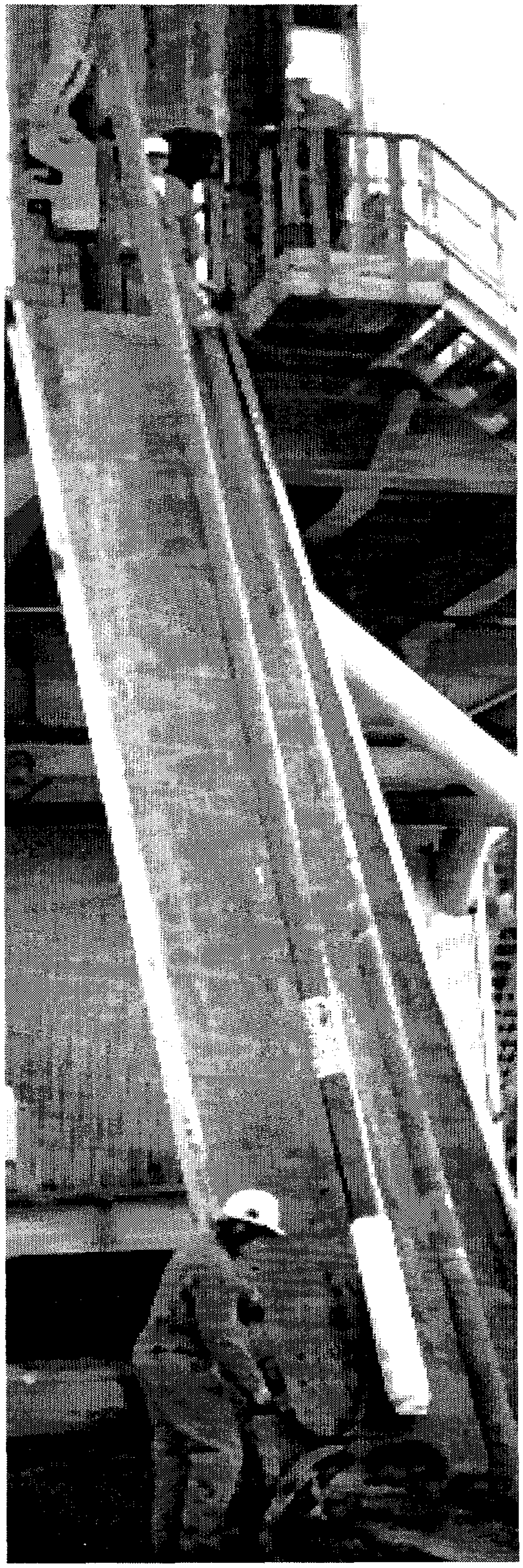


Figure 31. Second Prototype DHP Being Picked Up on the Drill Rig in Norway.

4.2 Field Experiment Parameters

Values of the drilling parameters for each of the field experiments with the second prototype are given in Table 5. FE-6 through FE-8 were conducted in UPRC gas wells being drilled in East Texas in the Carthage field and were all with the same Nabors drilling rig. Two DHPs were run in each of the last three field experiments, FE-8 through FE-10. Rig surface mud pumps used during these experiments were a Gardner Denver PZ-9 with 5-1/2 inch liners with Nabors Drilling, and a Continental Emsco F-1000 equipped with 4-1/2 liners for FE-9, and with 5 inch liners for FE-10. Maximum surface standpipe pressures during DHP operation were all between 3,400 and 3,500 psi. Mud weights were also generally higher than wells in which the first prototype was tested. Because of DHP problems, maximum UHP output pressures in FE-8 never exceeded 22,000 on the first DHP, or 25,000 psi on the second DHP except for the first few minutes of operation on each pump. The bit selection, weight on bit and bit rpm during each field experiment were generally the same as normally would have been used conventionally without the DHP and jet-assist.

Table 5. Second Prototype Field Experiment Parameters.

No.	Bit Size	In ft.	Footage ft.	WOB 1000 lb	RPM	Pump psi	Mud ppg	DHP psi
FE-6	7-7/8	6,242	240	45	48	3,400	9.9	29,000
FE-7	7-7/8	6,352	254	45	45	3,400	9.8	27,400
FE-8 #1	7-7/8	7,200	94	50	55-65	3,400	10.1	22,000
FE-8 #2	7-7/8	7,294	36	50	55-72	3,400	10.1	25,000
FE-9 #1	7-7/8	4,070	72	13-44	60	3,500	9.8	29,000
FE-9 #2	7-7/8	4,142	0	0	0	3,500	9.8	29,000
FE-10 #1	7-7/8	4,147	67	13-44	60	3,500	9.8	29,000
FE-10 #2	7-7/8	4,221	51	35	60	3,500	9.8	29,000

In East Texas, where FE-6, FE-7 and FE-8 were conducted, rig standpipe pressures average about 1,950 psi for conventional drilling with carbide insert roller cone bits. Table 6 shows a comparison between conventional and UHP jet-assisted drilling hydraulics as the DHP was operated during FE-7. The conventional bit has a pressure drop of 1,180 psi at 300 gpm, or 4.2 HSI. With the DHP, most of the hydraulic horsepower is directed to the UHP jet on the drill bit, resulting in a power density of about 5.2 HSI. To accomplish this, some of the conventional pressure drop is taken away from the bit, leaving only about 1.2 HSI for the conventional bit nozzles. The total across the UHP jet-assisted bit is about 6.4 HSI.

Table 6. DHP Jet-Assisted and Conventional Hydraulics.

	Conventional	Jet-Assisted
Mud Pump Power (hhp)	343	497
Stand Pipe Pressure (psi)	1,960	3,420
Circulation Rate (gpm)	300	249
Pressure Losses (psi)	780	550
DHP Pressure Drop (psi)	-	2,458
UHP Pressure (psi)	-	27,400
UHP Flowrate (gpm)	-	16
Bit Pressure Drop (psi)	1,180	412
Bit Flowrate (gpm)	300	233
Bit HSI		
Conventional	4.2	1.2
UHP	-	5.2
Total	4.2	6.4

4.3 Downhole DHP Field Experiment Failures

Hours achieved in each of the downhole field experiments are shown in Figure 32. The DHP failures experienced downhole with the second prototype are listed in Table 7. Failures in FE-6 and FE-7 were due to an unreliable sleeve retention method in the main valve in one case and overstressing the main valve drive rod in the latter case. These failures were resolved with design changes that were introduced and confirmed through laboratory testing, concluding in the 45- and 35-hour laboratory test runs mentioned in Section 3.

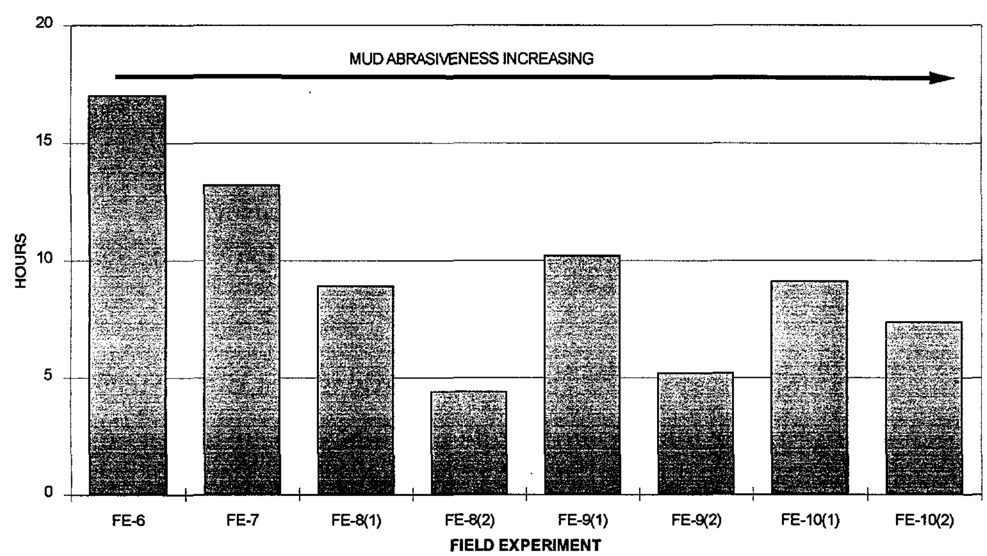


Figure 32. Second Prototype DHP Downhole Hours at Pressure.

The failures limiting downhole pump life in the last three downhole experiments, FE-8, FE-9 and FE-10, were all due to fluid erosion or abrasion within the DHP. Beginning with FE-8, it became apparent that the drilling mud in the field was more abrasive than in the laboratory. As indicated in Figure 32, the field drilling mud became generally more abrasive with each successive field experiment, resulting in more severe erosion and more apparent abrasion.

Table 7. Second Prototype Failures During Downhole Experiments.

Experiment	DHP Failure
FE-6	Main valve sleeve, lower, C-clad
FE-7	Drive rod parted, upper end
FE-8 #1	Manifold, upper, eroded
FE-8 #2	Manifold, upper, eroded
FE-9 #1	Pilot Valve, excessive clearance
FE-9 #2	Manifold, lower, thin wall blowout
FE-10 #1	Manifold, upper, eroded
FE-10 #2	Manifold, upper, eroded

The approaches being evaluated to address the fluid erosion and abrasion issues are more erosion-resistant materials and erosion-resistant coatings. Specific areas that need to be addressed are erosion of the valve manifolds, upper and lower, erosion of the main valve drive rod, erosion within the low pressure drive pistons; abrasion between the connecting tubes and low pressure dynamic seals, and abrasion between low pressure liners and the drive piston dynamic seals. It is believed, however, that to properly resolve the erosion and abrasion issues, several areas of the design of the DHP will require significant revision. As a result, it is planned to design, build and test a third prototype that is expected to be a pre-production prototype DHP.

4.4 Jet-Assisted Drill Bits

Including the first and second prototype DHPs, there have been 10 downhole field experiments during which 15 UHP jet-assisted drill bits have been run a total of 3,550 feet in 147 hours. Consequently, the jet-assisted drilling data base with the DHP is rather small and limited in scope in comparison to previous experience with the FlowDril system (see Section 1.1). A more complete discussion of UHP jet-assisted of mechanical drilling was given by Veenhuizen, et al (1997a).

The drill bits used during the downhole field experiments with the DHPs have all been conventional, off-the-shelf, tri-cone insert bits to which a single UHP jet has been adapted. They have been either Hughes Christensen or Smith International bits ranging from IADC codes 417 through 737. Two bits to which ultra-high pressure has been adapted are shown in Figure 33. The bit on the left has a high pressure tube extended through the conventional mud jet port. This arrangement is referred to as the "RT" nozzle and high pressure tower arrangement. The ultra-high pressure nozzle, not installed on the bit in the photograph, is at the extended end of the high pressure tube. The bit on the right, referred to as the "ST" arrangement, has a ported leg welded in place of the conventional mud jet port to support the ultra-high pressure nozzle. The ultra-high pressure nozzle is threaded into a port on the bottom of the leg.

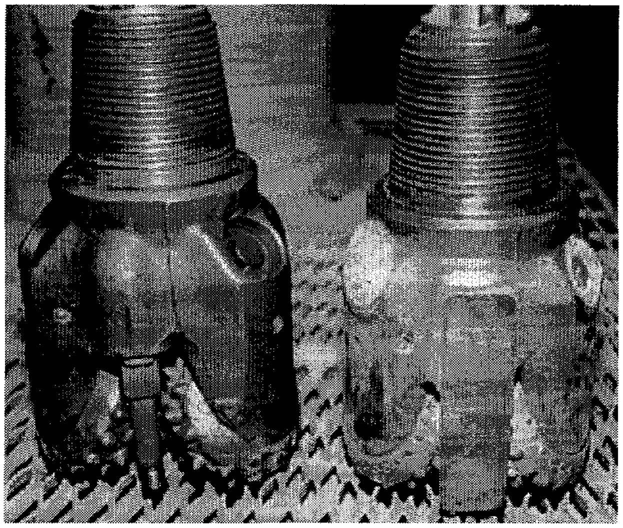


Figure 33. Jet-Assisted Drill Bits; Left is "RT" arrangement and Right is "ST" arrangement of Ultra-High Pressure Tower and Nozzle.

The ultra-high pressure jets produced on the jet-assisted bits have all been continuous, fully submerged, non-cavitating jets generally directed at the hole bottom near the gage of the hole. The UHP nozzles used have all been FlowDril proprietary nozzles. Nozzle diameters used downhole have ranged from 0.072 inches to as small as 0.059 inches in diameter. The nozzle discharge coefficient is typically 0.95.

If the jet pressure developed at the rock surface is sufficiently above some threshold pressure, it will erode, or cut the rock, creating a well defined slot or kerf in some cases, or create a series of connected craters or spalled regions. Decay of the center line pressure with distance from the nozzle exit is shown in Figure 34. Typically, the jets maintain full nozzle pressure out to about five nozzle diameters, after which the center line pressure starts to decay and the UHP jet starts to lose effectiveness. To make jets effective for cutting rocks, sufficient nozzle pressures are required and decay of jet pressure with distance from nozzle requires placing the nozzles as close as is practical to the hole bottom.

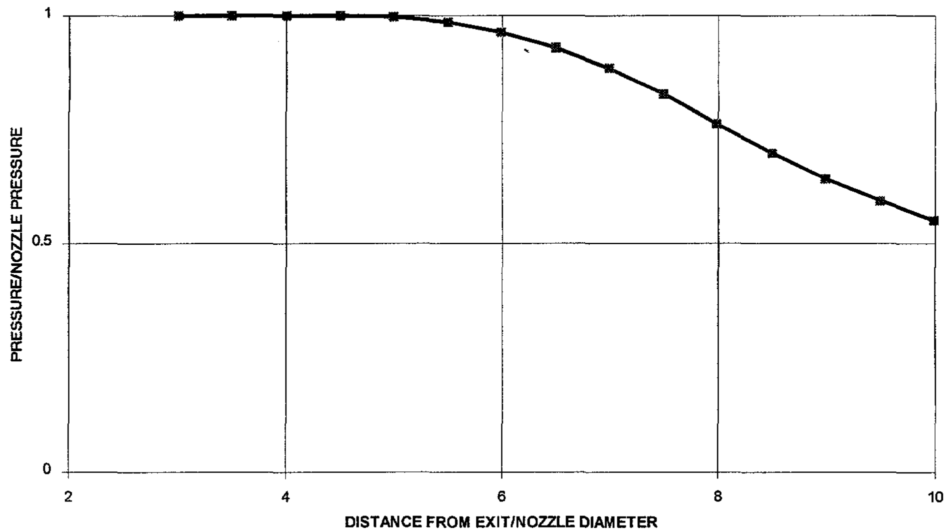


Figure 34. Center Line Pressure Decay of Ultra-High Pressure Nozzle.

The bits used in each of the field experiments with the second prototype are given in Table 8. Also included in the table are the offset well bits for which ROP comparison data was obtained, the footage drilled with UHP, and the UHP nozzle size used on each UHP jet-assisted bit. The second DHP with the ATJ66D bit for FE-9 was not run on bottom and no actual drilling occurred because of rotary drive problems with the rig.

Table 8. Jet Assisted Drill Bits Run with First and Second Prototype DHPs.

No.	Bit Diameter In.	UHP Bit	Offset Bit	UHP Footage, ft.	UHP Nozzle Dia. In.
FE-6	7-7/8	F3H-RT	F3	240	0.059
FE-7	7-7/8	F3OD-RT	ATJ35	182	0.062
	7-7/8	F3H-ST	ATJ35	72	0.062
FE-8 #1	7-7/8	F7-RT	F8	110	0.062
FE-8 #2	7-7/8	F7OD-ST	F8	20	0.062
FE-9 #1	7-7/8	F57-RT	F57	72	0.062
FE-9 #2	7-7/8	ATJ66D-RT	F57	0	0.062
FE-10 #1	7-7/8	ATJ66D-RT	F57	67	0.062
FE-10 #2	7-7/8	F57-RT	F57	51	0.062

4.5 ROP During Field Experiments

For each of the DHP field experiments, ROP information was gathered from either the operators, or from the rigs. These data varied from very detailed digital data from elaborate data acquisition systems for FE-9 and FE-10, to the simple rig geolograph charts and the mud loggers minutes per foot reports (also from the rig geolograph) for FE-6, 7 and 8. In each case, an effort was made to acquire conventional ROP data from either a nearby offset well, or from either immediately higher or lower in the same well bore and formation. In all cases, the offset bits were either the same manufacturer and type, or at least the same type and IADC code. A comparison was then made between the UHP jet-assisted ROP and the conventionally drilled ROP data at the corresponding depths to determine the ratio of UHP jet-assisted ROP to the conventional ROP.

Shown in Figure 35 for each field experiment, including those conducted with the 7-5/8 inch first prototype DHP, is the ratio of UHP jet-assisted average ROP to the average conventional ROP, where the average ROP was determined over the range of depth drilled with UHP. The ROP ratio for FE-1 was not included in the chart as it was very shallow, 240 to 500 feet, such that the UHP jet naturally cavitated and was much more effective than normal. The UHP ROP ratio for FE-1 was between 3.5 and 4.5.

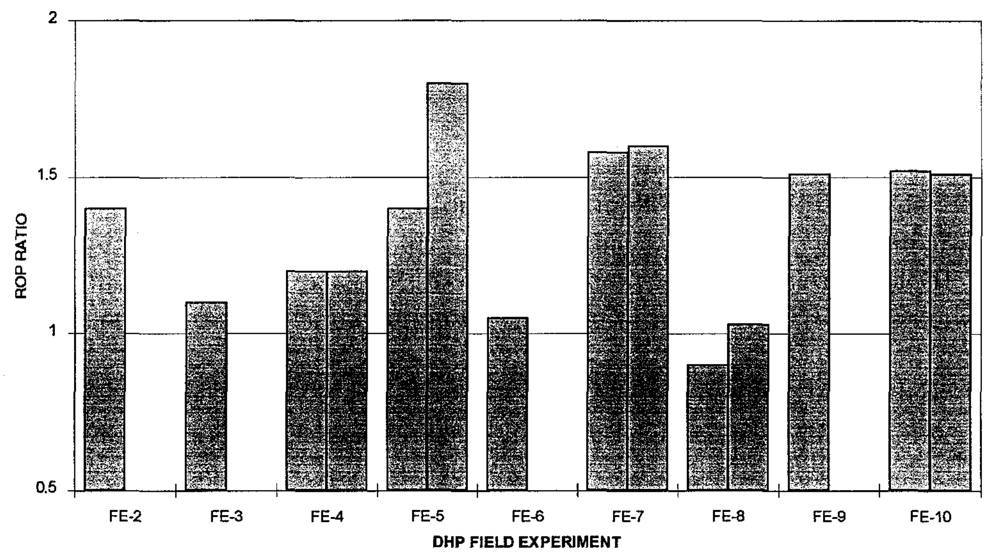


Figure 35. Jet-Assisted ROP Ratios during Field Experiments.

The ROP ratio for FE-2 through FE-4 varied between 1.1 and 1.4. In each of the field experiments FE-1 through FE-3, and FE-6, the UHP nozzle was conservatively set away from the hole bottom. This was to minimize the risk of losing the nozzle against the hole bottom and causing a premature conclusion to the DHP experiment. It was set the same distance from the hole bottom as had been the practice with the previous FlowDril system even though nozzles with the DHP were smaller in diameter. The typical UHP nozzle diameter with the previous FlowDril system was 0.080 to 0.084 inches. Even in FE-4 where the "ST" nozzle tower arrangement was introduced to allow locating the UHP nozzle closer to the hole bottom, the nozzle was still not close enough to be effective.

Beginning with FE-5 testing of the bits in FlowDril's laboratory drilling test facility was begun. As a result, ROP ratios for FE-5, FE-7 and FE-9 and 10 were a more respectable 1.5 to 1.6.

Laboratory drilling test results for the Smith International F3H and F3OD bits prior to the FE-7 field experiment is shown in Figure 36. The F3H bit had been run previously in field experiment FE-6 and had not exhibited any significant ROP enhancement with UHP jet-assist. FE-6 was the first field experiment with the second DHP prototype and the UHP nozzle, as mentioned above, was conservatively set away from the hole bottom. For FE-7, both bits had the "ST" UHP nozzle tower arrangement located between the same cones; however, one had the nozzle closer to the hole bottom by 1.3 nozzle diameters than the other. The UHP jet-assisted ROP data for the two bits were essentially indistinguishable, one having an ROP ratio of 1.58, the other 1.55 at the highest WOB, although the higher value was for the bit with the nozzle closer to the hole bottom.

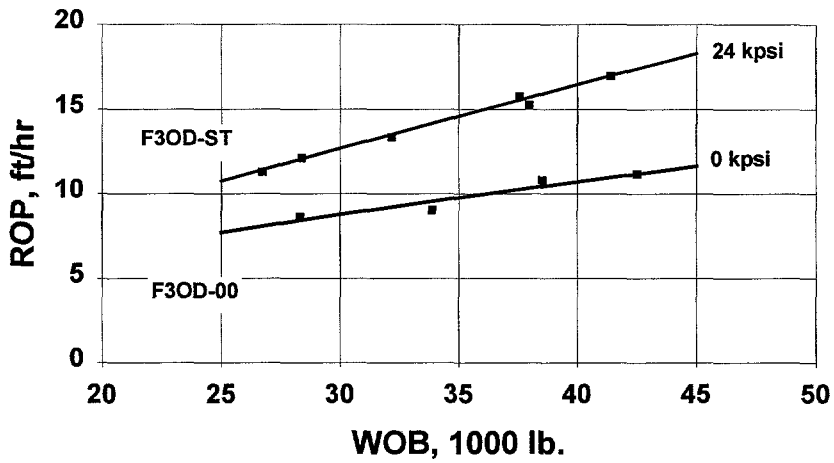


Figure 36. Jet-Assisted ROP of 7-7/8" F3 Bits, Carthage Marble, OBP 2,100 psi, do 0.062".

The bits run in FE-8 were F7's, one with the "RT" configuration and the faster one with the "ST" arrangement with the nozzle repositioned and the nozzle closer to the hole bottom. Unfortunately, difficulties with the DHPs precluded full output pressure during most of this field experiment, and a new experimental UHP nozzle on the "ST" bit failed after the first hour of drilling before the jet-assisted ROP could be fully established, so it was not possible to accurately assess the jet-assisted performance of these bits. In FE-9 and 10, bits with the "RT" nozzle tower configuration were used; however, the risk was taken to place the nozzles closer to the hole bottom and they were optimally located between cones on the bits as determined by testing in the laboratory.

Shown in Figure 37 is the effect of locating a single nozzle tower through different mud jet ports of a tri-cone bit on jet-assisted ROP ratio. This affects the location and spacing of the UHP jet with respect to rows of cutters on different cones and which cutters lead or follow the UHP jet. The bit was a Smith International F57 with the "RT" UHP nozzle arrangement. This bit was run in the laboratory in Sierra White Granite in preparation for sending to a field experiment, FE-9, at Rogaland Research in Stavanger, Norway. The nozzle tower was tried between cones one and two (1/2), between cones two and three (2/3), and between cones three and one (3/1). It is apparent that a significant difference exists between the several locations. The best result, an ROP ratio of 1.25, was obtained for case (1/2). A further improvement to a ratio of 1.32, case (1/2IB), was achieved by moving the nozzle inboard slightly from the gage once it had been determined to locate the nozzle tower between cones one and two. Similar testing to determine the optimum location for the UHP was performed in the laboratory with the ATJ66D-RT bit used in FE-10.

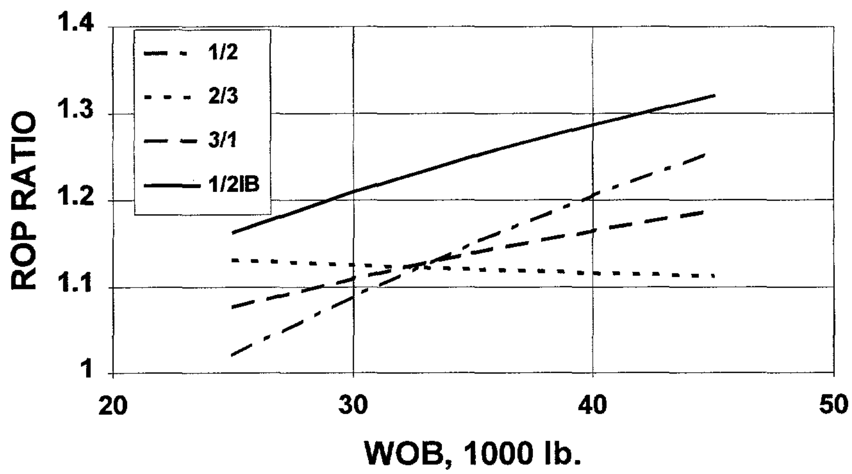


Figure 37. Effect of Jet Placement on ROP Ratio for 7-7/8" F57 Bit, Sierra White Granite, OBP 2,000 psi, do 0.062", Po 25,000 psi.

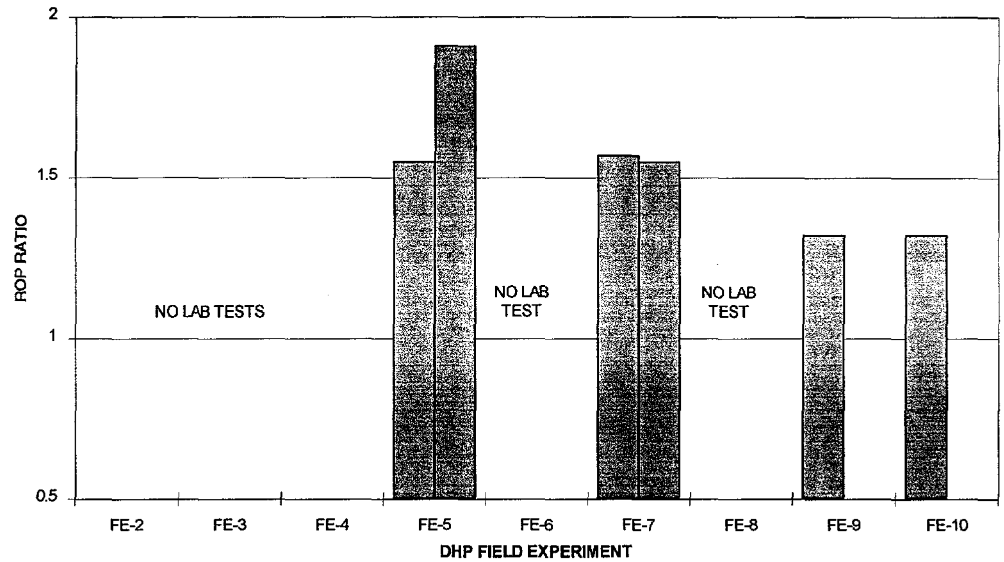


Figure 38. Laboratory ROP Ratios for Field Bits.

The laboratory ROP ratio data determined for the bits used in the field experiments with the first and second prototype DHPs are summarized in Figure 38. When comparing Figure 38 with Figure 36, it is apparent that the laboratory and field correlate reasonably well, even though field conditions and rocks were different. It is also apparent that better ROP ratios were achieved in the field when the bits were first tested in the laboratory prior to the field experiment. When the bits were tested prior to the field experiment, and the DHP was able to provide expected output pressures as in FE-5, 7, 9, and 10, the ROP ratios were between 1.5 and 1.6.

5. RESULTS, CONCLUSIONS AND FUTURE PLANS

The most significant results and conclusions from this work are:

- DHP Concept - Downhole field experiments have demonstrated that ultra-high pressure, up to 30,000 psi, can be generated downhole using a reciprocating, intensifier style, ultra-high pressure downhole pump (DHP), and that the resulting ultra-high pressure jet-assist of the drill bit using the DHP can increase bit rate of penetration (ROP) by 1.5 to 1.6 times conventional rates. It was also shown that packaging the DHP into the envelope of a standard drill collar was a viable concept. The field experiments verified the handling and operation on the drilling rig, as well as verifying the DHP performance parameters.

- DHP Reliability - It was recognized that reliability of the DHP has to be improved; however, the 41-hour downhole run during the third field experiment showed that commercially viable life expectancy was achievable under certain conditions. The failures limiting downhole pump reliability to about 10 hours in the last several downhole experiments were all due to fluid erosion or abrasion. In particular, the upper valve manifold failed as a result of fluid erosion four times. Specific design issues that remain to be addressed are erosion of the valve manifolds, upper and lower, erosion of the main valve drive rod, erosion within the low pressure drive pistons; abrasion between the connecting tubes and low pressure dynamic seals, and abrasion between the low pressure liners and the drive piston dynamic seals.

- DHP Future Plan - The approaches being evaluated to address the fluid erosion and abrasion issues are more erosion-resistant materials and erosion-resistant coatings. To properly resolve the erosion and abrasion issues, several areas of the design of the DHP will require significant revision. As a result, it is planned to design, build and test a third prototype that is expected to be a pre-production prototype DHP.

Results and conclusions from Phase II of this project are summarized below. Phase II included laboratory testing and field experimentation with a second prototype DHP for drilling in 7-7/8 inch holes. Also included in Phase II was some laboratory ROP testing of the ultra-high pressure jet-assisted drill bits used with the DHP during field experiments.

5.1 Phase II Results and Conclusions

Over 50 laboratory test runs were conducted, accumulating 480 hours of laboratory test time on the second prototype DHP. A number of proprietary design improvements were introduced during laboratory testing to address mechanical strength and dynamic loading issues, low cycle fatigue, fabrication difficulties, and both drive fluid and UHP sealing issues. At the conclusion of laboratory testing, the second prototype had achieved run times averaging 40 hours with laboratory drilling mud. Five field experiments were also conducted with the 6-3/4 inch second prototype, all in 1996, designated FE-6 through FE-10. A total of 75 downhole test hours was accumulated over eight DHP runs, with the longest DHP run of 17 hours occurring on field drilling mud in a well in East Texas. The last two field experiments, FE-9 and FE-10, were conducted in RF-Rogaland's test well in Stavanger, Norway. From the laboratory and field experimentation with the second prototype, the following results and conclusions were obtained:

1. Performance - Introduction of an even faster shifting main valve and increased pilot valve travel length allowed the second prototype to achieve the rated stroke rate of 120 SPM. As initially tested, the hydraulic efficiency had been increased to between 72 and 75 percent; however, later design modifications reduced the efficiency to 70 percent.
2. DHP Envelope - The field experiments again verified the handling and operation of the DHP on the drilling rig. Making it shorter as well as smaller in diameter, and improving the fishability of the tool by including a fishing neck on the upper end of the housing to accommodate an overshot made the tool easier to handle and the downhole risk more acceptable.
3. Surface Mud Pumps - The rig mud pumps available during the field experiments with the second prototype could reliably operate the DHP at surface pressures at least as high as 3,550 psi.
4. Laboratory Failures - The most frequent DHP failures during laboratory testing were again the UHP dynamic seals, the lower check valve seat (due to an improper material), and the low pressure dynamic seals. Improvements were introduced during the course of laboratory testing to improve the reliability of each of these components
5. Field Failures - The failures limiting downhole pump life in the last three downhole experiments, FE-8, FE-9 and FE-10, were all due to fluid erosion or abrasion within the DHP. In particular, the upper valve manifold failed as a result of fluid erosion four times. It was apparent that the drilling mud in the field was more erosive and more abrasive than in the laboratory. The field drilling mud became generally more abrasive with each successive field experiment, resulting in more severe erosion and more apparent abrasion, limiting DHP life to about 10 hours for the last six of the eight downhole runs.
6. Design Issues - Specific areas that need to be addressed are erosion of the valve manifolds, upper and lower, erosion of the main valve drive rod, erosion within the low pressure drive pistons; abrasion between the connecting tubes and low pressure dynamic seals, and abrasion between the low pressure liners and the drive piston dynamic seals. The approaches being evaluated to address the fluid erosion and abrasion issues are more erosion-resistant materials and erosion-resistant coatings. To properly resolve the erosion and abrasion issues, several areas of the design of the DHP will require significant revision.

Testing and development of the DHP was the focus of the downhole experiments with the DHP; however, there was always an interest in the jet-assisted ROP. Including the first and second prototype DHPs, there have been 10 downhole field experiments during which 15 UHP jet-assisted drill bits have been run a total of 3,550 feet in 147 hours. Consequently, the jet-assisted drilling data base with the DHP is rather small and limited in scope. For each of the DHP field experiments, ROP information was gathered from the either the operators, or from the rigs. In each case an effort was made to acquire conventional ROP data from either a nearby offset well, or from either higher or lower in the same formation. A comparison was then made between the UHP jet-assisted ROP and the conventionally drilled ROP data at the corresponding depths to determine the ratio of UHP jet-assisted ROP to the conventional ROP. During the course of the DHP field experiments, it became evident that a more thorough understanding of the mechanism and interaction of UHP jet-assisted of roller cone bits would be beneficial. As a consequence, some testing of the UHP jet-assisted drill bits was conducted in FlowDril's drill bit testing facility prior to several of the downhole field experiments. From the laboratory bit testing and the field experiments, the following UHP jet-assisted ROP results and conclusions were obtained:

1. ROP Ratio - When the UHP jet-assisted bits were tested prior to field experiment, and the DHP was able to provide expected output pressures as in FE-7, 9, and 10, the UHP ROP ratios were between 1.5 and 1.6.
2. UHP Jets - The UHP nozzles used on the jet-assisted drill bits have all been FlowDril proprietary nozzles. To make jets effective for cutting rocks, sufficient nozzle pressures are required and decay of jet pressure with distance from nozzle requires placing the nozzles as close as is practical to the hole bottom.
3. UHP Jet Position - Locating a single nozzle tower through different mud jet ports of a tri-cone bit affects the location and spacing of the UHP jet with respect to rows of cutters on different cones and which cutters lead or follow the UHP jet. It was apparent that a significant difference in jet-assisted ROP occurred when locating the UHP nozzle between different cones on a tri-cone bit.
4. Laboratory and Field ROP Correlation - Laboratory and field ROP ratios correlated reasonably well even though field conditions and rocks were different. It was also apparent that better ROP ratios were achieved in the field when the bits were first tested in the laboratory prior to the field experiment. Although the UHP jet-assisted ROP has been typically between 1.5 and 1.6 times the conventional ROP, an improvement in the UHP jet-assisted ROP ratio would make the DHP jet-assisted drilling economics more favorable. Therefore, it is planned to increase the effort with the UHP jet-assisted bits to provide consistent ROP results and improve the ROP ratio.

5.2 Future Plans

The approaches being evaluated to address the fluid erosion and abrasion issues are more erosion-resistant materials and erosion-resistant coatings. Specific areas that need to be addressed are erosion of the valve manifolds, upper and lower, erosion of the main valve drive rod, erosion within the low pressure drive pistons; abrasion between the connecting tubes and low pressure dynamic seals, and abrasion between and low pressure liners and the drive piston dynamic seals. It is believed, however, that to properly resolve the erosion and abrasion issues, several areas of the design of the DHP will require significant revision. As a result, it is planned to design, build and test a third prototype that is expected to be a pre-production prototype DHP.

Although the UHP jet-assisted ROP has been typically between 1.5 and 1.6 times the conventional ROP, an improvement in the UHP jet-assisted ROP ratio would make the DHP jet-assisted drilling economics more favorable. Therefore, it is planned to increase the effort with the UHP jet-assisted bits to provide consistent ROP results and improve the ROP ratio.

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