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Development of a Downhole Piston Motor Power Section For Improved Directional Drilling: Part I – Design, Modeling & Analysis

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Keywords

directional drilling, motor, piston motor, axial piston, rotary piston, longitudinal motor, PDM, turbine, drilling, geothermal, high temperature, vibration mitigation

ABSTRACT

Directional drilling can be used to enable multi-lateral completions from a single well pad to improve well productivity and decrease environmental impact. Downhole rotation is typically developed with a motor in the Bottom Hole Assembly (BHA) that develops drilling power necessary to rotate the bit apart from the rotation developed by the surface rig. Historically, wellbore deviation has been introduced by a "bent-sub" that introduces a small angular deviation to allow the bit to drill off-axis with orientation of the BHA controlled via surface rotation. The geothermal drilling industry has not realized the benefit of Rotary Steerable Systems, and struggles with conventional downhole rotation systems that use bent-subs for directional control due to shortcomings with downhole motors. Commercially-available Positive Displacement Motors are generally limited to approximately 350 °F (177 °C) and introduce lateral vibration to the bottom hole assembly contributing to hardware failures and compromising directional drilling objectives. Mud turbines operate at higher temperatures but do not have the low speed, high torque performance envelope for use with conventional geothermal drill bits. Development of a fit-for purpose downhole motor would enable geothermal directional drilling.

Sandia National Laboratories is developing technology for a downhole piston motor to enable drilling high temperature, high strength rock. Application of conventional hydraulic piston motor power cycles using drilling fluids is detailed. Work is described comprising conceiving downhole piston motor power sections; modeling and analysis of potential solutions; and

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development and laboratory testing of prototype hardware. These developments will lead to more reliable access to geothermal resources and allow preferential wellbore trajectories resulting in improved resource recovery, decreased environmental impact and enhanced well construction economics.

1.0 Introduction

Positive Displacement Motors (PDM) are presently used as the workhorse of any directional drilling operation. However, these motors introduce significant lateral vibration to the drilling bottom hole assembly as the rotor follows an eccentric orbit within the motor housing to generate rotation. This lateral vibration is deleterious to the stability of drilling operations. The drilling industry is hampered by drilling dynamic dysfunctions that contribute to reduced performance, hardware failures, and increased drilling costs. Mud turbines operate at higher temperatures but do not have the low speed, high torque performance envelope for using conventional geothermal drill bits to penetrate a variety of rock types. An improved method of downhole rotation is needed that does not rely upon elastomers and performs as required for drilling high strength rock at elevated temperature. These limitations in operating temperatures and deleterious vibrations emphasize the need for new technology that maintains and/or improves drilling penetration rates, is capable of drilling to depth and can survive geothermal well soak temperatures if the bottom hole assembly must remain in the hole with no circulation. A viable downhole motor will enable multi-lateral completions for improved resource recovery, decreased environmental impact, and enhanced well construction economics. Development of a high temperature motor is also an EGS-enabling technology.

The motor concept is a Sandia-proprietary[2] downhole rotation solution proposed to the DOE Geothermal Technologies Office as a viable downhole rotation option. Fundamentally, the concept is a piston motor that relies upon a harmonic drive coupling to convert hydraulically-activated reciprocating piston motion into rotary motion in an output rotor. The concept is physically analogous to hydraulic swash-plate type piston motors that react piston generated forces against an angled swash plate to produce reactive torque in a barrel housing to generate rotary motion. Sandia has performed computational modeling, produced working prototypes, and developed test fixtures to validate the concept.

The objective of the work described herein is: 1) Develop technology for a new downhole motor, 2) Design a representative power section and demonstrate viability with a proof of concept demonstration, and 3) Enable intellectual property licensing and commercialization of a viable reduced-vibration downhole rotation solution for high temperature directional drilling. Part I of this paper describes motor requirements, the hydraulic power cycle, an axial piston solution, computational modeling to support the concept, and a rotary piston solution. Future publications will address supporting component and materials analysis, laboratory prototype testing, and potentially field drilling demonstrations.

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[2] U.S. Patent 9,447,798B1 from U.S. Patent Application No. 14/209,840, filed 3/13/2014; CIP of U.S. App. No. 14/298,377, filed 05/05/2014 and U.S. Provisional Patent Application No. 62/142,837, filed 4/3/2015 and U.S. Patent App. No. 15/090,282 - Modular Fluid Powered Linear Piston Motors with Harmonic Coupling and U.S. Patent App. No. 62569074 - Fluid-Powered Linear Motor with Rotary Pistons and Motion Rectifier.

2.0 Downhole Drill Motor Requirements

Drilling performance requirements are needed to quantify specifications for downhole motor power sections for drilling hard rock. Theoretical torque specifications can be derived based upon a widely-accepted rock-reduction model in the literature using representative properties for typical rock formations. The derived values correspond to optimum motor performance for rock reduction at minimum specific energy and form a set of minimum requirements on output torque and power for downhole motors. Actual values should be increased to account for factors such as increased hydrostatic pressure at depth, bit wear, heterogeneous rock, and non-ideal drilling conditions.

The approach uses a method to predict motor performance requirements for drilling at minimum specific energy. This method can be used to derive operational performance specifications for rotary speed, torque, and output power that must be delivered by candidate motor power sections to drill representative rock formations. While field drilling rarely proceeds at the ideal condition of minimum specific energy, this condition corresponds to maximum drilling efficiency and preferred operational loads. Requisite torque and power values are derived corresponding to minimum specific energy for various drilling conditions. These values are derived for a series of bit diameters drilling a variety of rock types across a range of operating conditions. Derived values may be subsequently used to compare performance envelopes for existing downhole motors. While downhole motors presently provided by the energy services industry are inadequate for drilling at elevated temperature or in high-strength rock characteristic of geothermal formations, the data derived herein will allow the suitability of existing motors to be assessed relative to minimum required performance metrics anticipated for geothermal drilling. The specific energy, E_s, is the energy required to drill a unit volume of rock where for rotary drilling the energy input is dominated by the rotary work component:

$$E_S = \frac{\text{Rotary Energy Expended Drilling}}{\textit{Unit Rock Volume Removed}} = \frac{\int \mathrm{T}\, d\theta}{V_r} = \frac{2\pi T}{\pi\, r_b^2 \delta} = \frac{2T}{r_b^2 \delta} = \frac{8T}{d_b^2 \delta} \tag{1}$$

where T is the torque required to drive a drill bit, θ is the rotation angle, V_r is the volume of rock removed, r_b is the bit radius, d_b is the bit diameter and δ is the depth of cut per revolution (ft/rev).

Following the method from Detournay (1999), the minimum specific energy will approach the *intrinsic specific energy* of the rock formation for "pure cutting" with a polycrystalline diamond (PDC) drag bit. Following Teale (1965), it has been observed that the minimum specific energy "is of the order of the compressive strength of the material drilled." Hence, a governing expression for the operational torque, T, for a drag bit to drill a formation of Unconfined Compressive Strength (UCS), at minimum specific energy may be derived from equation (1):

$$T = \left(\frac{d_b^2}{8}\varepsilon\right)\delta \tag{2}$$

where ε is the UCS of the rock (psi).

The depth of cut is

$$\delta = ROP/(60N) \quad \text{[ft/rev]} \quad = \quad ROP/(5N) \quad \text{[in/rev]} \tag{3}$$

where ROP is the rate of penetration (ft/hr), and N_m is the rotary speed (RPM).

This operational torque will increase in proportion to the hydrostatic pressure with increasing depth. Hence the method predicts the required torque at atmospheric operating conditions and must be increased in proportion to bottom-hole pressure.

The rotary power delivered by the motor, P_m, is

$$\mathrm{P}_m = \frac{2\pi \,\mathrm{T}\, N_m}{(60*550)} \quad [\mathrm{hp}] \tag{4}$$

Bit diameters of 4, 6, 8, 10 & 12 inches are evaluated. Strength (UCS) values for Berea Sandstone, Arizona Sandstone, Sierra White Granite, and Mississippi Limestone are used as representative values to span the range of UCS values that may be encountered. Rotary speeds are used comparable to what is generally practiced on PDM motors. Using these values, tabular data are generated to determine values of $d^2\varepsilon/8$; rate of penetration, ROP, corresponding to a specific depth of cut, doc, and rotary speed, N_m; operational torque, T, corresponding to $d^2\varepsilon/8$ values; and delivered power, P_m, corresponding to speed and operational torque. These values are graphically portrayed in Figure 1.

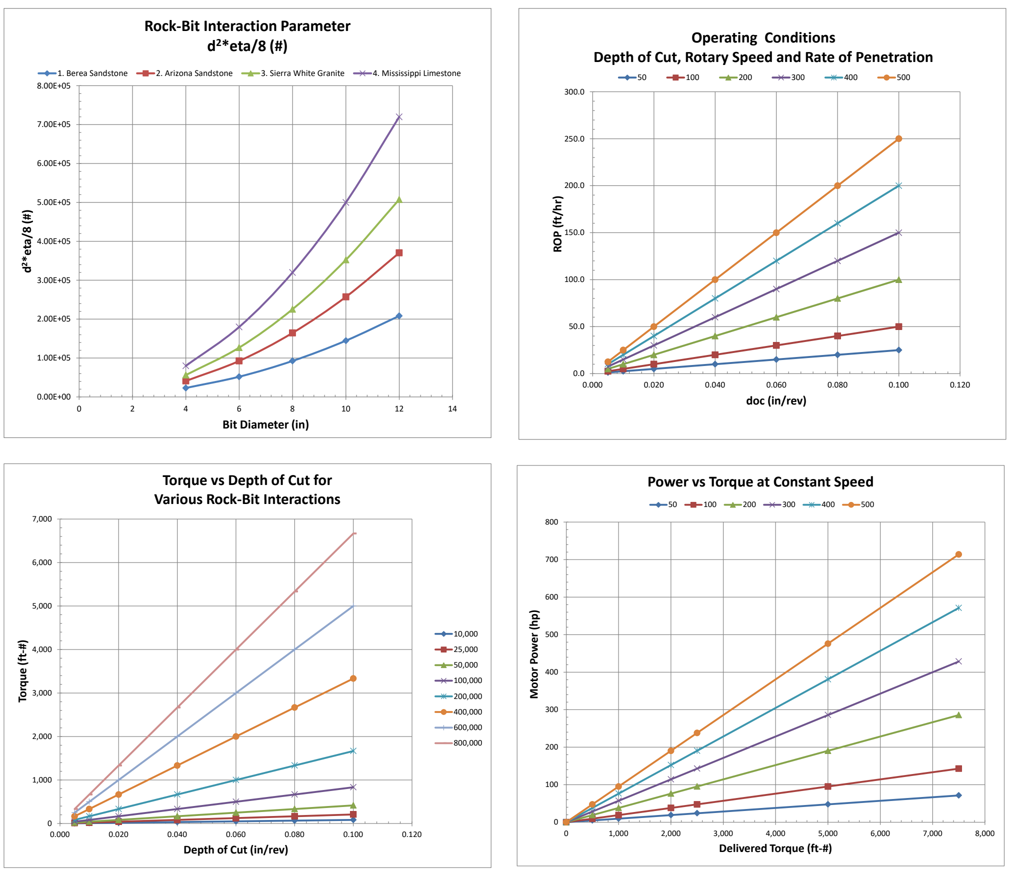

Figure 1. Nomograph to predict theoretical minimum torque and power to drill representative rock formations: a) Rock bit interaction parameter, b) Operating Conditions, c) Torque vs depth of cut, and d) Power vs Torque (sub-figures labeled across and then down).

The figures may be used as a "nomograph" to derive the operational torque and power, subject to known operating conditions, as follows: 1) Use Figure 1a to derive the $d^2\varepsilon/8$ term corresponding to the bit diameter and rock type; 2) Use Figure 1b to select the depth of cut, doc, corresponding to the target rate of penetration, ROP at the desired rotary speed, N_m. (A maximum depth of cut of 0.1 inch/rev is used as an upper limit as this is a large advance rate for a drag bit); 3) Use Figure 1c to derive the torque, T, corresponding to the depth of cut, doc, and $d^2\varepsilon/8$ value; and 4) Use Figure 1d to derive the power, P_m, corresponding to the torque, T, and rotary speed, N_m. For a given bit diameter, minimum torque requirements are predicted corresponding to the specified UCS and operating condition. Rotary power output corresponds to the predicted torque and rotary speed condition; hydraulic power input must be greater commensurate with the overall efficiency of the motor.

Another means to identify performance requirements is to conduct a survey of PDMs that are commercially available – although existing PDMs may not have torque enhancements for drilling the typically harder geothermal formations. A cursory survey of PDMs presently available in industry is shown in Figure 2. Comparison of these values to the above indicates that conventional PDMs are capable of driving PDC bits to drill representative 44 ksi UCS rocks at large penetration rates (0.1 inch/rev).

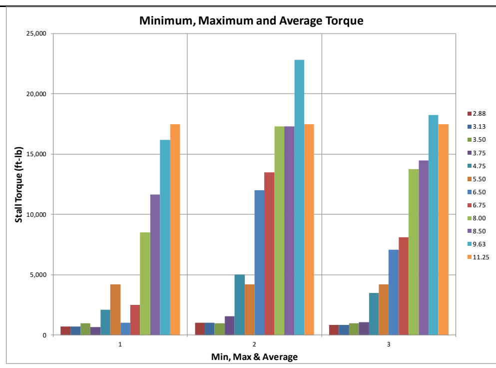

Figure 2. Survey of representative commercially-available PDM motors.

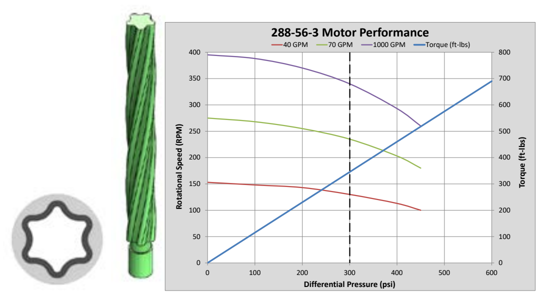

Figure 3. Positive Motor Specification for a 288-56-3 (Toro, 2009) with illustration of section of PDM rotor and a cross section through the stator comprising the power section.

Of particular interest to this development is a representative 288-56-3 PDM with a 2-7/8 inch motor housing, 5-lobed rotor, 6-lobed stator, and three stages. A prototype motor piston will be developed for side by side comparison with this PDM and presented in Part II & III of this paper series. Its flowrate- speed relationship and pressure-torque characteristic are shown in figure 3. At the recommended operating line (dashed vertical line), this motor produces 345 ft-lb at 300 psi pressure differential; the stall torque is 690 ft-lbs at 600 psi differential pressure. The speed at various flowrates are plotted on the left axis and are subject to reduced speed delivery as the fluid leaks past the elastomer at increasing pressure differentials. This motor will be used as a standard comparison in the eventual development of a longitudinal piston motor.

3. Piston Motors

3.1 Conventional Hydraulic Piston Motors

Piston motors are the widely used in the mobile hydraulics industry. The general configuration is shown in figure 4. It consists of an array of pistons in individual cylinders symmetrically disposed around the central axis of a cylindrical barrel. The piston motor generates rotary motion and output torque in response to pressurized fluid conveyed across the valve plate to the individual pistons. As the cylindrical barrel rotates, the individual cylinders periodically pass over the intake and discharge ports on the valve plate. As the individual pistons experience inlet pressure, the piston slipper reacts against a front swash plate. Since the swash plate is inclined at an angle, α_b, with respect to the central shaft the pistons will experience harmonic motion as the cylindrical barrel rotates. Since the slipper reacts against the swash plate at an angle, a side load is produced that acts on the cylindrical barrel inducing rotation. The motion of the cylindrical barrel is coupled to the output shaft of the machine to deliver rotary speed and motive torque for the application. Figure 4b below describes rotor torque generation in the piston motor.

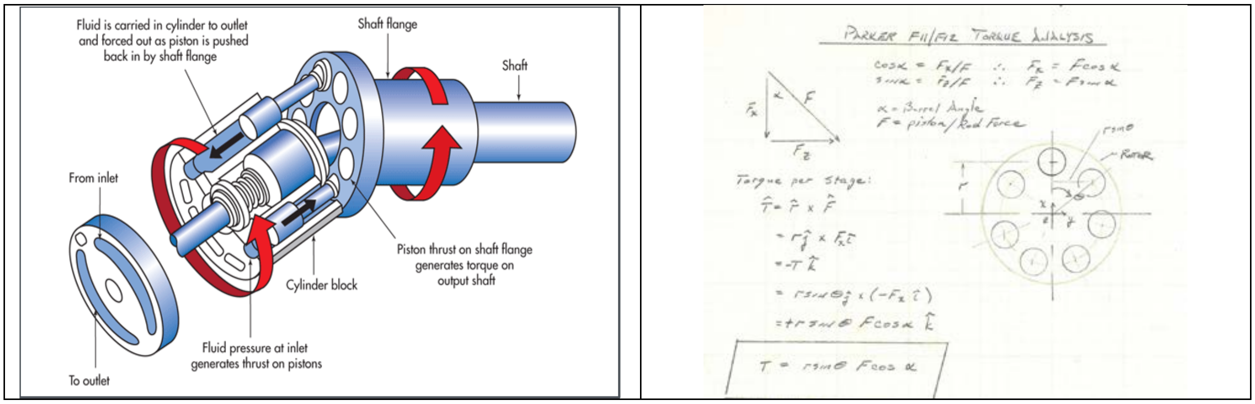

Figure 4: a) Conventional Hydraulic Piston Motor (per Hydraulics & Pnematics.com) and b) Method of Torque Generation.

3.2 Longitudinal Hydraulic Piston Motor - Requirements and Considerations

The objective of this work is to develop a prototype power section that enables a longitudinal form factor of the typical hydraulic piston motor shown in Figure 4. Following the analog of the hydraulic piston motor, it should include the following power section design features.

- The fluid power cycle should generate piston oscillation by hydraulic flow through the power section. This will require alternating pressure on the piston lands for reciprocation and will require the valves to be synchronized to the piston motion.
- A harmonic drive coupling must convert axial piston force and motion to rotor torque and rotation.
- It will require multiple pistons for continuous rotation to overcome the dwell points, or motion reversals of the pistons. Multiple pistons will additionally be required for adequate torque generation.
- The overall design intent must accommodate: 1) fluid leakage around the piston section (no seals), 2) Low friction surfaces at the piston interfaces to reduce parasitic losses, 3) A removable rotor assembly for servicing, 4) case/rotor design integration, 5) pressure/exhaust manifold integration, and 6) piston motion / valve port integration.

Both an axial piston and rotary piston configuration have been conceived, designed and developed for a longitudinal piston motor and are the subject of the balance of this paper.

4. Axial Piston Motor Concept

4.1 Overview

The form factor of a downhole piston motor excludes use of the general design configuration used in a conventional piston motor as the piston forces would be inadequate to develop sufficient torque to be comparable to the requirements presented in section 2. Accordingly, the hydraulic piston motor design arrangement must be linearized by sequential arrangement of piston motor modules that work collectively to generate rotation and torque analogous to hydraulic piston motors except the pistons are not symmetrically distributed around a common output rotor but rather linearly distributed along the axis of the power section. This allows the major dimension of the piston to approach the diameter of the motor housing giving rise to higher net torque. Such a linear piston motor module in shown in Figure 5 and performs the speed and torque generation function of a single piston within a conventional hydraulic motor. Other configurations can be conceived as described in the referenced patents. A linear array of these modules may be serially connected to a common output shaft to produce an entire power section for a full motor as shown in Figure 6. The axial piston motor module concept in Figure 5 includes several major assemblies: Housing, Liner, Rotor and Exhaust Manifold, Valves and Valve Block, Piston, Ball Transfer and Harmonic Drive. These assemblies comprise the longitudinal piston motor concept with the following attributes.

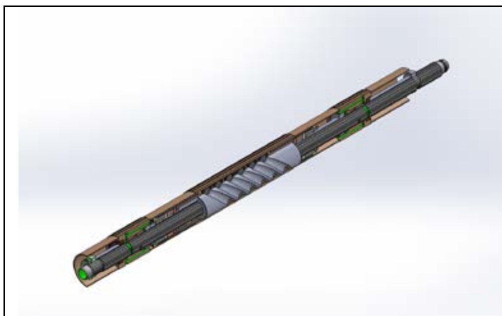

Figure 5. Linear piston motor module.[3]

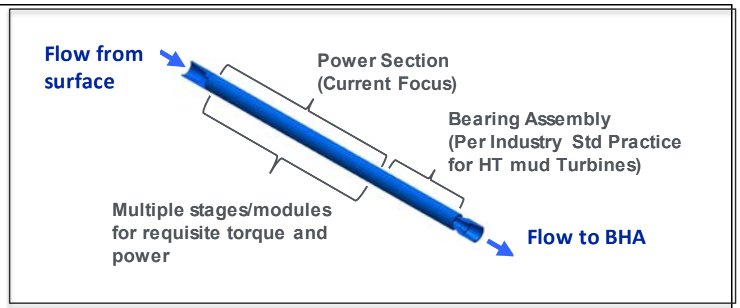

Figure 6. Linear array of modules for full power section.

- Housing - The housing is the body of the motor and protects the inner components from the return flow in the annulus.
- Liner - A liner is the motor interface to the housing. It allows reactive torque in the liner to be transferred to the housing. It also provides features for fluid conveyance as needed for fluid power conversion.
- Rotor/Exhaust Manifold – The rotor is the main fluid conduit through the motor and must deliver pressurized flow to the piston and collect exhaust when discharged. It is also the member that carries the shaft power from the individual modules comprising the power section to the balance of the motor. Figure 7 shows the rotor with an exhaust manifold

[3] Axial Piston Motor Module. U.S. Patent 9,447,798B1 from U.S. Patent Application No. 14/209,840, filed 3/13/2014; CIP of U.S. App. No. 14/298,377, filed 05/05/2014 and U.S. Provisional Patent Application No. 62/142,837, filed 4/3/2015 and. U.S. Patent App. No. 15/090,282 – Modular Fluid Powered Linear Piston Motors with Harmonic Coupling.

within it to collect and transfer exhaust flow to the end of the power section. The integration of the exhaust manifold with the rotor must keep the flows separated until the power fluid has been delivered to all chambers comprising the power section.

- Valve Block & Valves - Like conventional hydraulic motors, the longitudinal motor concept employs rotary valves that selectively pressurize and exhaust piston chambers during rotor rotation. The function of the valve block is to convey power fluid to the piston chamber to act on the piston. The valves are keyed to the rotor with pressurization and discharge ports that are synchronized with piston reciprocation. The valve ports and passages are specified to convey required flowrates to support piston extension and retraction.

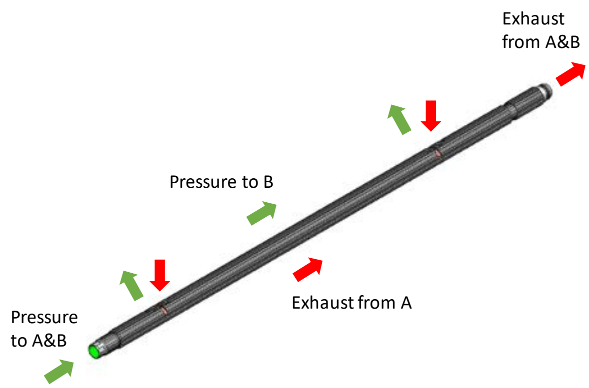

Figure 7. Rotor with centralized exhaust manifold

- Piston - The hydraulic energy in the pressurized drilling fluid is converted to kinetic energy in the piston. The piston reacts against a barrel cam during its stroke to convert piston kinetic energy to output rotor rotational energy. The interface between piston and rotor requires high strength ball transfers to facilitate conversion of piston force into rotor torque. The piston seals against the liner and reciprocates in response to differential pressure on its lands. The piston reacts against the harmonic drive to develop torque in the output rotor. Like in the hydraulic analog, the piston must react against an inclined plane to develop a lateral force component to produce torque in the output rotor.
- Ball transfer – The ball transfer takes the place of the slipper foot in the hydraulic piston motor. However, rather than sliding across the inclined plane as the slipper foot does on the swash plate, the ball transfer rolls on the harmonic drive. At the piston interface, the ball rides in a special seat to reduce rolling friction at the piston interface.
- Harmonic Drive - The hydraulic energy is converted to kinetic energy in the piston that must be converted to motive torque in the rotor. This is accomplished in the axial piston motor module shown in Figure 5 using a barrel cam – the central feature of the Harmonic Drive. The barrel cam is preferentially specified to generate torque in the output rotor. The harmonic drive is a cylindrical cam, sometimes referred to a drum or barrel cam, manufactured to produce the desired displacement response in the piston with rotor turning angle. As described in Figure 8, the barrel cam is characterized by a cut in the circumferential surface of the harmonic drive. This feature is mathematically specified so that a cam follower will produce desired motion. The cam follower can be specified using a harmonic, a cycloid, or polynomial functions (Mabie & Ocvirk, 1978).

4.2 Kinematic Response

Assuming a harmonic function, the follower (piston) displacement is

$$z = \frac{S}{2}\left(1 \;- \cos\left(\frac{\pi\theta}{\beta}\right)\right) = \frac{S}{2}(1 \;- \cos(m\theta)) \tag{5}$$

where z is the piston displacement, S is the Stroke, and m is the number of harmonics per revolution with $m = \frac{\pi}{\beta}$ where β is the angular rotation to full stroke S. For a single harmonic per revolution, m= 1 & β=π. Likewise, for two harmonics per revolution, m=2 and β=π/2, etc.

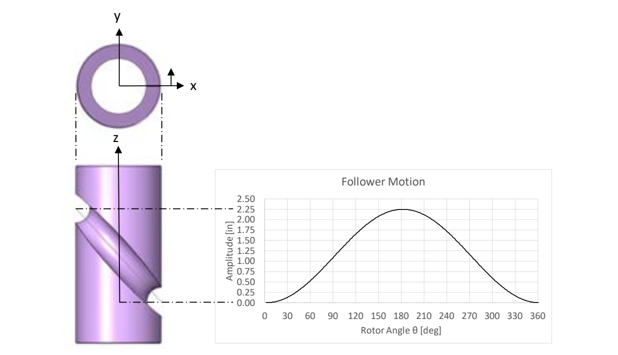

Figure 8. Barrel cam construction imparting harmonic motion to a follower.

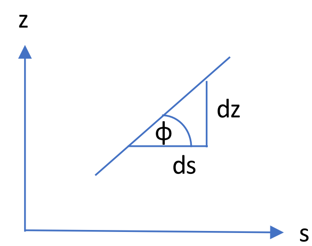

Figure 9. Pressure angle for barrel cam.

The piston velocity, v, and acceleration, a, are known from the cam follower motion:

$$v = \frac{dz}{d\theta} = \frac{\pi S}{2\beta}\left(sin\frac{\pi\theta}{\beta}\right) = \frac{mS}{2}(\sin(m\theta\,)) \tag{6}$$

$$a = \frac{dv}{d\theta} = \frac{\pi^2 S}{2\beta^2}(cos\frac{\pi\theta}{\beta}) = \frac{m^2 S}{2}(cos(m\theta\,)) \tag{7}$$

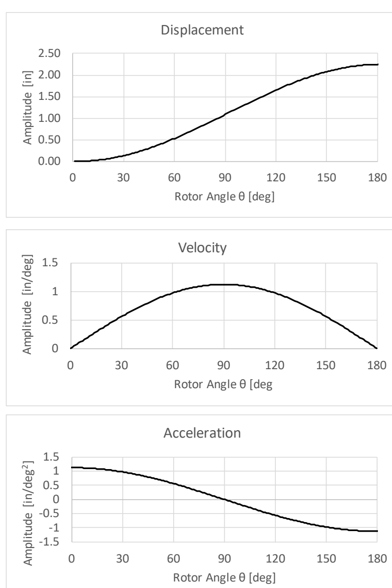

Figure 10. Piston displacement, velocity and acceleration from barrel cam harmonic motion.

The inclination or pressure angle, ϕ, throughout the cycle is also of interest as this will influence the conversion of piston thrust to torque. As shown in Figure 9, the inclination angle is $\tan^{-1}(\frac{dz}{ds})$. Using s = rθ for the circular distance on the barrel at the barrel cam radius, r, then

$$\frac{dz}{ds} = \frac{dz}{d\theta}\frac{d\theta}{ds} \tag{8}$$

$$\emptyset = \tan^{-1}[\frac{\pi S}{2\beta r}\sin(\frac{\pi\theta}{\beta})] = \tan^{-1}[\frac{mS}{2r}\sin(m\theta)] \tag{9}$$

Note there is a preferred ratio of stroke to radius for a given harmonic drive to produce favorable pressure angles in the harmonic motion response.

4.3 Piston Force Balance

The force balance on the piston in Figure 11 is

$$\sum F_z = p_1A_p - p_2A_p - F_R - m_p\ddot{z} = 0 \tag{10}$$

where p_1 is the forward motion piston pressure, p_2 is the backpressure on the piston, F_R is the load reaction at the harmonic drive, m_p is the mass of the piston, A_p is the piston area, and piston friction is neglected, then

$$F_R = A_p(p_1 - p_2) - m_p\ddot{z} \tag{11}$$

Figure 11. Piston Free Body Diagram.

4.4 Rotor Torque Generation

Following the example of Shigley & Mitchell for the Mechanics of Power Screws, imagine an infinitesimal length of the single harmonic track that is unrolled, as shown in Figure 12. One edge of the track will form a hypotenuse of a right angle with a corresponding pressure angle, ϕ. The axial force, F_A, applied to the harmonic drive is transmitted from the ball transfer (i.e., equal and opposite to F_R). A load F_T acts on the contact point to resist lowering the load F_A along the track. This load is produced by the resistive torque load on the rotor assembly.

Figure 12. Force Balance at the Harmonic Drive/Ball Transfer Interface (change F in above to F_A and P to F_T).

Summing forces in the horizontal direction:

$$\sum F_H = F_A - N\cos\phi - \mu N\sin\phi = 0 \quad => \quad N = \frac{F_A}{\cos\phi + \mu\sin\phi} \tag{12}$$

And in the vertical direction:

$$\sum F_V = F_T - N\sin\phi + \mu N\cos\phi = 0 \quad => \quad F_T = N(\sin\phi - \mu\cos\phi) \tag{13}$$

Combining to eliminate the normal force N and dividing top and bottom by cosϕ:

$$F_T = F_A\frac{(\sin\phi - \mu\cos\phi)}{(\cos\phi + \mu\sin\phi)} \quad => \quad F_T = F_A\frac{(\tan\phi - \mu)}{(1 + \mu\tan\phi)} \tag{14}$$

The torque on the harmonic drive due to the ball reaction F_T is T= F_T d/2 = F_T r where d & r are the corresponding diameter and radius at the ball contact. The torque on the harmonic drive is:

$$T = F_T r = F_A r\frac{(\tan\phi - \mu)}{(1 + \mu\tan\phi)} \tag{15}$$

5. Computational Modeling & Analysis

A control volume of the piston bore is used to address the coupling between the power fluid mechanics and the reciprocating pistons, and investigate influence of valve geometry and timing on motor performance. This will prescribe the pressure variation with rotor rotation within the piston chamber and is used needed to address a force balance on the piston and determine its influence on rotor torque generation. The governing equations are developed and then used in a computational code to predict results.

5.1 Control Volume Analysis

A control volume is defined for the piston bore as shown in Figure 13. The instantaneous mass within the piston bore is

$$M = \rho V \tag{16}$$

where M, ρ & V are the fluid mass, density and volume of the control volume, respectively.

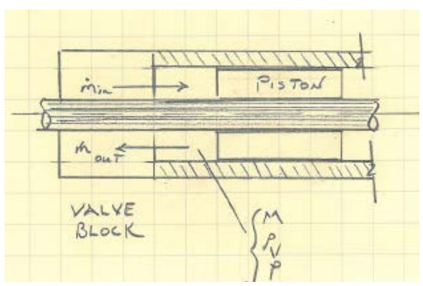

Figure 13. Control Volume of the Piston Bore.

The fluid mass time rate of change is

$$\frac{dM}{dt} = \frac{d\rho}{dt}V + \rho\frac{dV}{dt} \tag{17}$$

For a mass flow rate $\dot{m}$, and defining Q as the flow rate, then for conservation of mass

$$\frac{dM}{dt} = \rho Q = \dot{M}_{in} - \dot{M}_{out} = 0 \tag{18}$$

From the definition of Bulk Modulus,

$$\beta_f = \rho \frac{dP}{d\rho} \tag{19}$$

where β_f is the bulk modulus of the fluid and P is the fluid pressure, then,

$$\frac{dM}{dt} = \rho Q = \frac{\rho}{\beta_f}\frac{dP}{dt}V + \rho\frac{dV}{dt} \tag{20}$$

Solving for the pressure change,

$$\frac{\rho}{\beta_f}\frac{dP}{dt}V = \rho Q - \rho\frac{dV}{dt} = \rho(Q - \frac{dV}{dt}) \tag{21}$$

$$\frac{dP}{dt} = \frac{\beta_f}{V}\left(Q - \frac{dV}{dt}\right) \tag{22}$$

With θ as the rotor angle, and ω as the rotor speed, then using $\omega = \frac{d\theta}{dt}$ and applying the chain rule as $\frac{dV}{dt} = \frac{dV}{d\theta}\frac{d\theta}{dt}$, then the pressure in the chamber as a function of rotary angle is

$$\frac{dP}{dt} = \frac{dP}{d\theta}\frac{d\theta}{dt} = \frac{\beta_f}{V}\left(Q - \frac{dV}{d\theta}\frac{d\theta}{dt}\right) \tag{23}$$

and

$$\frac{dP}{d\theta} = \frac{\beta_f}{V}\left(\frac{Q}{\omega} - \frac{dV}{d\theta}\right) \tag{24}$$

This formulation shows the flowrate in and out of the control volume must be proportional to rotary speed as the piston extends and retracts to change the size of the control volume. If proportional flow is maintained, then the difference function may be approximately zero and the pressure variation will not change significantly with rotation. It also shows the flowrate generally must follow the velocity distribution as the dV/dθ term depends upon the piston speed. Finally, this function shows the ratio of the bulk modulus of the fluid to the fluid volume will act as a gain on this difference magnifying the pressure gradient during the piston reciprocation cycle. This gain is particularly influential at minimum chamber volume.

5.2 Flowrate

The flowrate is determined using the orifice equation governing flow through the valve block assembly based upon the pressure in the control volume:

$$Q = C_d A_0 \sqrt{2\left|\frac{P_b - P}{\rho}\right|} \tag{25}$$

Where C_d is an orifice discharge coefficient, P_b is the boundary pressure outside the control volume, A_o is the orifice flow area and as before P is the pressure in the control volume and Q is the flowrate.

5.3 Computational Modeling Results

The physics coupling hydraulic fluid power, piston oscillation, and rotor shaft power are engineered using a fluid-structure interaction computational model. An integration scheme is used to account for the dP/dθ term. Assuming a three-module motor at constant rotor speed (add other input information), the overall response is shown in Figure 14. Of interest is the pressure profile as a function of rotor angle (14g). This shows for proper design, the pressure in the control volume may be fairly constant during piston extension and retraction. While the oscillations at the end point are dependent upon the accuracy of the integration scheme, the valves must be properly designed to manage pressure pulsations at the ends of the stroke. This result of near constant pressure allows for high torque production as the piston/harmonic drive combination passes through maximum pressure angle. Figures 14i & j show the flowrate for this speed and the resulting output torque at this pressure differential.

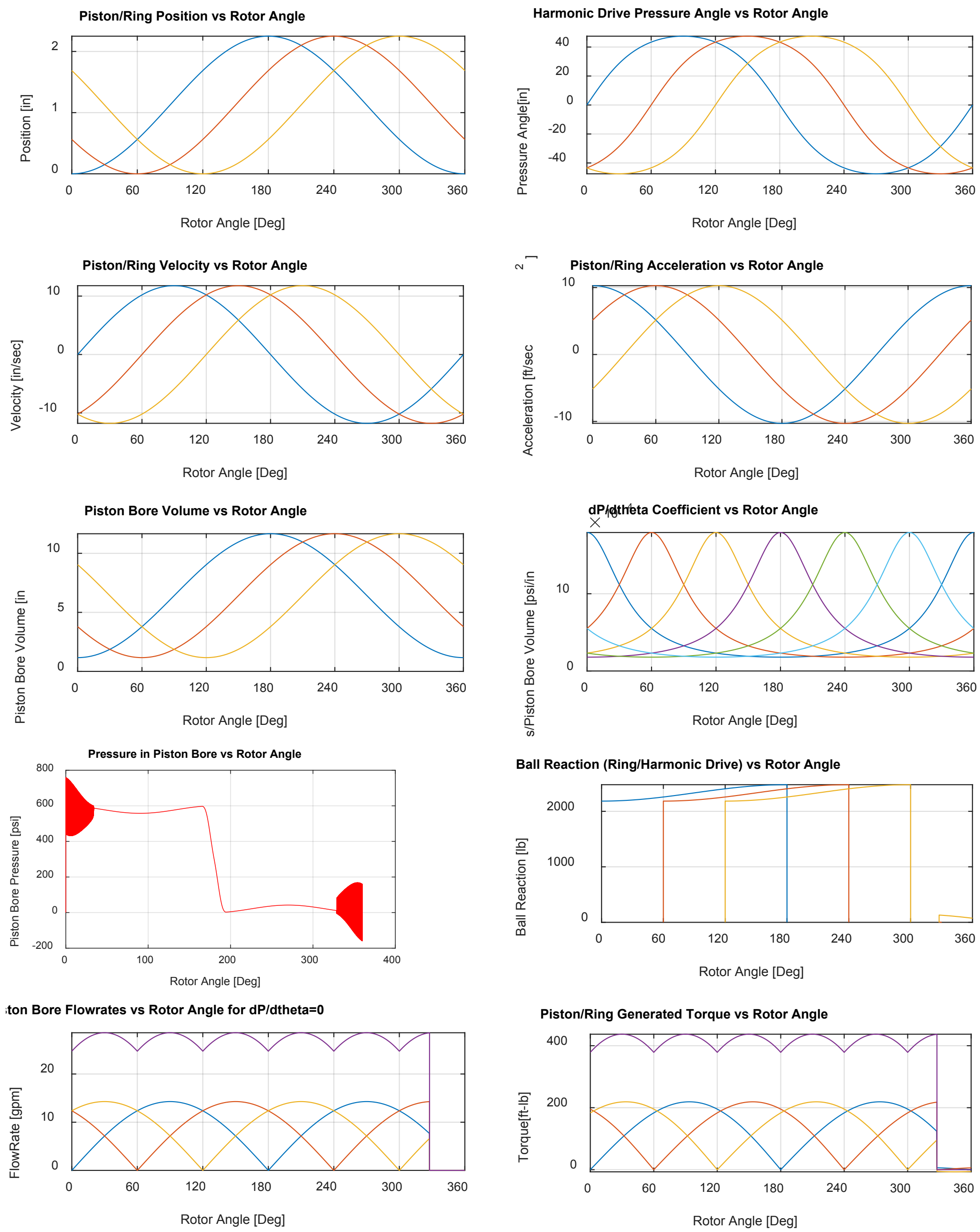

Figure 14. Computational model results for the axial piston motor with three modules.

6. Rotary Piston Motor Module Concept

6.1 Overview

Although the axial piston module meets the stated objective of linearizing the hydraulic piston motor concept, the output torque is linearly proportional to the number of modules. Additionally, the linear piston module configuration is incompatible with liquid drilling fluids as particulates in the drilling fluid will settle-out on the piston lands. While the linear module has applications, a rotary module is additionally considered.

The rotary piston module retains the fluid cycle features, the reciprocating ring, and the harmonic drive features already presented yet replaces the reciprocating axial piston as the power fluid interface with a reciprocating rotary bladed piston. Pressure acts on the piston lands of the rotary piston to create an offset force that generates torque on the rotor. The bladed rotary piston causes rotary reciprocation of a power screw, comparable to the harmonic drive, yet with a non-reversing track section. The power screw is coupled to the harmonic drive via a ring assembly and causes axial reciprocation of the ring assembly in response to rotational reciprocation of the bladed piston assembly. The reciprocating bladed piston can be fitted with multiple blades to produce the desired torque. Interconnection of these major components is shown in Figure 15.

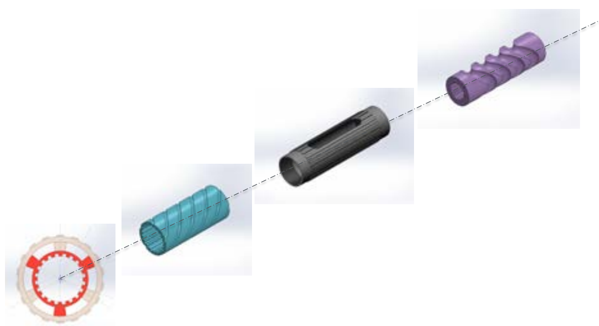

Figure 15. Rotary Piston Module Design with rotary piston, power screw, ring, & harmonic drive.[4]

The number of rotary piston blades increases in proportion to use with an increased number of harmonics to generate greater torque and less ring displacement at a given rotary speed. This results in the several available configurations addressed below; other options may be considered.

6.2 Block Diagram Formulation

A complete characterization is needed, as for the axial piston module, to interpret operation of the rotary piston module. However, the rotary piston module retains the positive displacement nature of operation whereby a fluid volume produces a rotational displacement and a pressure difference produces an output torque. Hence a linear block diagram analysis may be used to predict trends and maximum values that may result for various configurations.

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[4] U.S. Patent 9,447,798B1 from U.S. Patent Application No. 14/209,840, filed 3/13/2014; CIP of U.S. App. No. 14/298,377, filed 05/05/2014 and U.S. Provisional Patent Application No. 62/142,837, filed 4/3/2015 and U.S. Patent App. No. 15/090,282 - Modular Fluid Powered Linear Piston Motors with Harmonic Coupling and U.S. Patent App. No. 62569074 - Fluid-Powered Linear Motor with Rotary Pistons and Motion Rectifier.

Conversion of flowrate to rotor output speed proceeds according to the following block diagram:

Figure 16. Block diagram for conversion of flowrate to rotary speed.

Conversion of pressure differential across the rotary piston blades to output torque proceeds in likewise manner.

Figure 17. Block diagram for conversion of pressure differential to rotor torque.

6. 3 Rotary Piston Motor Performance

The rotary piston motor module allows the benefit of a higher number of harmonics on the harmonic drive resulting in a reduced displacement in exchange for a greater number of blades on the rotary piston that allows generation of greater overall torque. This is enabled by reducing the harmonic drive amplitude as the number of harmonics increases allowing large pressure angles to be maintained. Rotary speed and torque performance is summarized for a single module in Figures 18 & 19 as a function of input flowrate and pressure differential across the rotary pistons, respectively. These motor configurations meet the torque requirements of Section 2 and show how speed and torque can be tailored based upon requirements to produce a desired response. A full rotary piston motor may be fabricated with a minimum of two rotary piston modules with overall rms torque increasing in proportion to the number of modules. This analysis was performed for a 3 inch diameter motor; the concept can be applied to a variety of motor sizes with output torque scaling proportionally.

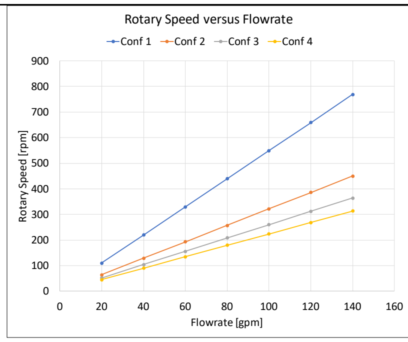

Figure 18. Rotary speed from flowrate for various rotary piston module configurations.

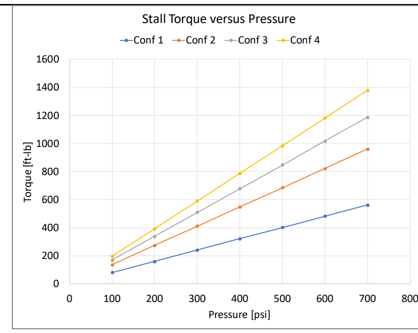

Figure 19. Output torque from pressure differential for various rotary piston module configurations.

7. Conclusions

A linear analog to a conventional hydraulic piston motor has been described and detailed. This motor concept can produce speed and torque amenable to driving standard drag drill bits for conventional drilling operations. Both an Axial and Rotary configuration have been described. The axial piston configuration may be more amenable to high speed lower torque applications. The rotary piston configuration is amenable to use with conventional drilling fluids as it accommodates flow-through the power section and is not susceptible to drilling fluid additives settling out on the piston lands. Computational modeling using the governing conservation of mass has demonstrated the concept can be configured to deliver rotary speed and torque required for the application. The concept does not employ elastomers for seals, will work at elevated temperatures, and does not introduce deleterious vibrations to the bottom hole assembly. This published work is a preliminary overview of the technology and is not meant to be exhaustive.

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Nomenclature

a	piston acceleration	r	radius of barrel cam
A_0	orifice flow area	r	radius of harmonic drive for torque computation
A_b	rotary piston blade area	r_1	inner radius of the rotary piston chamber
A_c	Area of a rotary piston chamber	r_2	outer radius of the rotary piston chamber
A_p	Axial piston pressure area	r_b	radial centroid of pressure distribution rotary piston blade
A_s	Area of a sector of a circle	ROP	Rate Of Penetration [ft/hr]
A_t	Area of all rotary piston chambers	s	Circular distance parameter
C_d	orifice discharge coefficient	S	Piston Stroke [in]
E_s	Drilling Specific Energy [psi]	t	Time
d_b	bit diameter	T	Torque
f_{rp}	rotary piston oscillation frequency	t_p	Pressure cycle period
F	load reaction at the harmonic drive	t_t	total cycle period
F_A	axial load applied to the harmonic drive	T_{RP}	Rotary Piston Torque
F_R	piston/ring load reaction at the harmonic drive	v	Piston Velocity
F_T	Tangential force on Harmonic Drive	V	fluid volume in control volume
$F_{T,HD}$	Tangential force on Harmonic Drive	V_r	Rock Volume Drilled
$F_{T,PS}$	Tangential force on Power Screw	V_T	Total rotary piston chamber volume
m	number of harmonics per revolution	z	Piston displacement [in]

Mdot	time rate of change of mass in control volume
n_b	number of piston blades
n_c	number of rotary piston chambers
M	fluid mass in control volume
m_p	mass of the piston
N_m	Rotary Speed [RPM]
N	Normal force on the harmonic drive and/or ball transfer
n	number of harmonics on harmonic drive
Δp	pressure difference across the blade
p_1	forward motion piston pressure
p_2	backpressure on the piston
P_m	Motor Power [hp]
P	Fluid Pressure in control volume
P_b	boundary pressure outside the control volume
Q	Flowrate [gal/min]
r_b	bit radius
r_{ps}	radius of power screw

<u>Symbols</u>

α	rotary piston stroke [deg]
α_b	swash block angle
β	Rotary angle for piston rise to full stroke
β_f	Bulk Modulus of the fluid
δ	depth of cut [in/rev]
ε	intrinsic specific energy of rock
θ	rotation angle [deg]
μ	friction coefficient
ϕ	pressure angle [deg]
ρ	fluid density in control volume
ω	rotor speed [rad/sec]
ω_1	rotor speed [rad/sec]
ω_2	secondary rotor speed [rad/sec]
ω_{RP}	rotary piston angular speed [rad/sec]

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