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Selection and Integration of Positive Displacement Motors into Directional Drilling Systems

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SENIOR PROJECT - APPROVAL

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PROJECT TITLE: Selection and Integration of Positive Displacement Motors into Directional Drilling Systems

I have reviewed this completed senior honors thesis with this student and certify that it is a project commensurate with honors level undergraduate research in this field.

Signed: Dr. Don Dureing, Faculty Mentor

Date: May 6, 2003

Comments (Optional):

Kirk has done an excellent job here and we intend to produce a technical paper from his report.
Selection and Integration of Positive Displacement Motors into Directional Drilling Systems

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May 7, 2003
Motivation

The positive displacement motor (PDM) has been used in directional drilling now for several years and is a key component in bottom hole drilling assemblies. These downhole drilling motors, coupled with Measurements While Drilling (MWD) tools and proper stabilization allow directional wells to be drilled with one bottom hole assembly, eliminating costly multiple drillstring tripping, which in the past was necessary for surveying, hole corrections, and stabilized assemblies.

A key issue in designing bottom-hole assemblies for specific directional drilling situations is the selection and operation of the “best” positive displacement motor (PDM) for the application. A part of this decision is the integration into the total drilling system, paying close attention to the hydraulic horsepower at the bottom of the drillstring for powering the motor and for cleaning the drill bit. This decision is compounded somewhat by the many possible choices of PDM’s available on the market. PDM’s are available in various sizes, rotor/stator lobes, etc., each having their own set of performance characteristics.

This paper presents a logical methodology for predicting general performance of PDMs and shows how to use this performance information to come to a logical basis for choosing a best motor for a given set of operating conditions.

Background

In the early part of the twentieth century, directional drilling was becoming a main stay in offshore drilling. Economics did not allow for multiple platforms to be built in order to recover more oil and gas from the reservoirs. The industry had to come up with new ways to drill multiply holes off of one production platform.

To accomplish this task, operators found that if they could divert the bit at an angle, then they could stabilize the drillstring in this orientation and drill in another direction besides straight down. Early attempts at this method included Whipstock wedges that would be oriented in the bottom of the hole and direct the bit in a predetermined path. The drillstring would then be pulled out and stabilizers that held the bottom hole assembly at a fixed curvature would be attached. This process was the standard practice until bent subs became more widely used. The bent sub was a fixed bend installed near the bit that would offset the direction of the drilling when employed. The bent sub replaced the Whipstock wedge. Once the initial kick off angle was established, the operators would trip out the drillstring and replace the bent sub with stabilizers as they did with the wedges and continue drilling. These methods resulted in wells that had an average radius of curvature of about 1500 feet. Platforms today typically have around forty directional wells stemming off of them.

As one can imagine this process was slow and costly. By the 1960’s and 1970’s downhole motors started to make headway in the drilling market [7]. One advantage that these downhole motors possessed was the fact that it created one bottom hole assembly. The operators no longer had to trip the drillstring in and out to change out bottom hole...
assemblies. The downhole motors had bent subs and stabilizers built into them so that the entire process could be accomplished in one pass. In today’s competitive industry, downhole motors are allowing operators to develop directional wells quicker and with more precision. Though the average well curvatures are still around 1500 – 2000 ft, some downhole motors combined with other technologies can create directional wells with curvatures around 200 feet. This reduction in curvature allows for operators to economically create horizontal wells, which offer many advantages including faster drilling.

Several types of downhole power units exist. In the past, electric motors have been evaluated; today however, downhole power units are typically positive displacement or turbines. The earliest power unit in this class is the Turbodrill; the first patent credited to C. G. Cross in 1873 [7]. It operates like a generator in that there are turbines contained in a sleeve fixed to a central axis. When the drilling mud or other hydraulic fluid is pumped through the motor the turbines rotate causing axial motion for the bit. The turbine power units are generally characterized by their high speed and torque capabilities. Their addition to the bit power is in the range of 250 hp. This addition is considerable when the power supplied to the bit by traditional rotary drilling methods is only around 23 hp (1500 ft-lbs & 80 rpm).

Another type of downhole power unit is the positive displacement motor (PDM). The first idea of this concept dates back to Rene Moineau, a French inventor who in the 1930’s designed pumps and compressors that used a powered rotor/stator device to move fluids. Later that decade T. Hudson and W. Gerber reversed Moineau’s concept and let the moving fluid turn the rotor thus converting hydraulic power into mechanical power. Throughout the mid 1900’s, this idea underwent several revisions until in 1957 Wallace Clark patented the first industrial PDM the “Dyna-Drill.” Since the original invention, many companies have invested in developing this tool. Eastman-Whipstock, Baker Hughes INTEQ, Dyna-Drill, Drilex, Schlumberger, and Sperry-Sun all have variations of a PDM with different rotor/stator relationships and performance characteristics [7].

PDMs come in a wide variety of classifications. Some are made for high torque applications and others for high speed. These characteristics are dependent on the rotor/stator relationship which will later be discussed. PDMs’ contribution to bit power is in the range of 150 hp. Even though this addition is not as much as the turbine power units, positive displacement motors offer several advantages. Turbine power units have to be operated at a specified rotational speed in order to achieve the level of power mentioned previously. This window of operation is much smaller than a PDM operating range. For the specified rotation to be achieved a downhole tachometer, a MWD tool, is necessary. PDMs do not require these MWD tools for operation. With the wide variety of performance abilities, a positive displacement motor exists for almost every application. This fact makes PDMs economical and ideal for retrofitting into existing production platform setups.
Components of Positive Displacement Motors (PDMs)

A positive displacement motor is made up of four basic components: a bent sub or adjustable housing, rotor/stator power section, bearing assembly, and a transmission unit. A surface pump on the drilling platform or ground moves the drilling mud through the drillstring to the PDM. This pump can provide the mud at varying flow rates and pressures. The fluid then moves into the power section of the motor.

**Figure 1** – Basic units of a PDM [1]

The rotor and stator act in a helical motion creating chambers in which the fluid pressure causes rotation in the rotor and thus rotation at the bit. More discussion on the kinematics of this relationship will follow. The bent sub can be located either before or after the power unit. Baker Hughes, manufacturer of the Navi-Drill, advocates the use of the bent sub after the power unit. In this geometry the offset between the bit and the axis of the drillstring is smaller which imposes less stress and wear on the motor as a whole. These bent subs allow for the PDM to change the direction of the drillstring.

The transmission unit in early models was a series of universal joints and rigid links to transfer the rotation and thus power to the bit. More recent motors employ the technology of flexible shafts in order to accomplish the same goal with less moving parts. Many PDMs also have a dump sub above the power unit. This dump sub allows for the drilling mud to bypass the motor when tripping the motor in and out of the hole. The PDM does not allow for drilling fluid to move through the motor under hydrostatic conditions. If the dump sub did not exist the PDM would trip out wet and trip in partially dry. Another way to let fluid move in and out of the bottom hole assemble during tripping is to install a jet nozzle in the rotor [1]. In this case the rotor has a hollow core so that fluid can pass not only through the rotor/stator power unit but also through the rotor center. Several restrictions apply to this technology, but it is useful when operating at high flow rates. The dump sub also can be a safety device. If the motor were to stall...
out, the dump sub would act as a pressure relief valve preventing serious damage to the PDM. Stall out will be discussed later in more detail.

The rotor and stator have a unique relationship in that the rotor has one less lobe than the stator. This geometric relationship is necessary to satisfy the fundamental gear tooth law. Manufacturers of the motors make sure that the design provides the smoothest rotation for the rotor around the stator. In this relationship it is important to note that the offset axial rotation creates vibration normal to the axial direction. This vibration can be minimized by proper gear design to aid in decreasing wear on the motor. Below are pictures of the original patent filed by Moineau for this concept and of common configurations that are currently being used in the industry today.

One main advantage of a PDM over the turbodrills is that their performance can be predicted and monitored without the use of measurement while drilling devices, MWD’s for short. Through motor theory and analysis of performance data provided by manufacturers, the affects of the addition of a positive displacement motor on the drilling system as a whole can be accurately estimated. Furthermore the parameters for idealization of the drilling system can be discovered.

**PDM Motor Theory**

Because of the geometric relationship between the rotor and stator, correlations can be drawn between pressure, flow rate, rotational speed, and torque. The following analysis combines fluid mechanics and kinematic relationships to form useful equations for PDMs. One note about the following analysis is that friction and leakage through the motor are ignored. These factors can be sizeable; however, if operation of the PDM is within defined limits, the following procedure gives practical predictions of the performance. Efficiencies can always be added to correct for the leakage if necessary.
In general the power produced by the drilling mud is proportional to the pressure and flow rate.

\[ P_{\text{hyd}} = pQ \]  

(1)

The pressure can be observed from the standpipe pressure and the flow rate is known. The mechanical power produced at the bit is:

\[ P_{\text{mech}} = TN \]  

(2)

Where the torque and rotation speed can be determined either by downhole devices or theoretically with other motor design parameters. As written these equations have no conversion factors however if

- \( p = \text{psi} \)
- \( Q = \text{gpm} \)
- \( T = \text{ft-lbf} \)
- \( N = \text{rpm} \)

Then \( P_{\text{hyd}} = \frac{pQ}{1714} \) hp

(1a)

And \( P_{\text{mech}} = \frac{TN}{5252} \) hp

(2a)

As the fluid moves through the motor, it experiences a pressure drop. This drop can be observed from the standpipe. When the BHA is just shy of the bottom of the hole and the pump is providing enough pressure to cause the PDM to rotate the observed pressure is called the off-bottom pressure. The pressure that occurs when weight or torque is applied to the BHA by making contact with the hole is the on-bottom pressure. The difference in on and off-bottom pressures is the pressure drop across the motor [7]. This differential pressure is proportional to the weight on the bit or applied torque. The more WOB the larger the differential pressure because the motor must create more torque to overcome the applied load. The hydraulic power consumed by the PDM then becomes;

\[ P_{\text{PDM}} = \frac{\Delta pQ}{1714} \]  

(3)

A point exists where the WOB exceeds the motor’s power producing capacity. This point is referred to as stall out. At this point the mud’s flow rate and pressure can no longer create enough torque to overcome the applied load and the rotor stops with mud being forced between it and the stator. Stall causes detrimental damage to the stator elastomer and the motor as 100% leakage occurs. Safety devices such as dump subs, bypass valves, and jet nozzles help to save the motors as discussed previously. It is best, though, to not operate near stall out conditions in order to avoid this situation. Many manufacturers list maximum operating conditions that already have a safety factor built in and are below stall out conditions.
Rotational Output of PDM

As mentioned the parameters T and N can be determined analytically. The rotational output speed has a direct relationship with the flow rate and can be written as:

\[ N = CQ \] (4)

In many cases, the output speed and flow rate are known from experimentation. Experimentally it is common to test PDM's on test stands and collect data for rotational speeds at varying flow rates. If N and Q are known to correlate to one another, the constant can be solved for directly. In the case of the Navi-Drills and also PDMs manufactured by Sperry-Sun, the constant can be found by correlating output speed to flow rates.

**Table 1 - Motor Performance Data [1 & 6]**

<table>
<thead>
<tr>
<th></th>
<th>Navi-Drill</th>
<th>Sperry-Sun</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor/stator</td>
<td>6 1/2&quot; M1X, 6 3/4&quot; M4XL</td>
<td>6 1/2&quot;, 6 3/4&quot;</td>
</tr>
<tr>
<td>flow-rate (GPM)</td>
<td>265-660, 265-530</td>
<td>300-600, 200-500</td>
</tr>
<tr>
<td>output speed (RPM)</td>
<td>90-220, 450-900</td>
<td>150-300, 200-500</td>
</tr>
<tr>
<td>C (rev/gal)</td>
<td>0.3365*, 1.698</td>
<td>0.5, 1</td>
</tr>
</tbody>
</table>

* some values C for min and max differ and recorded C is an average.

This calculated C would be valid for all performance data when an efficiency is added into it because as will be shown C depends solely on geometric parameters. Leakage in the motor can cause the constants to vary a little between the minimum and maximum operating conditions.

Analytically C can be calculated from equation 4 to:

\[ C = \frac{n}{q} \]

n = unit output shaft rotation
q = volume thru one stator stage of the motor

In order to further develop this equation, several schemes need to be considered. First, the output rotation needs to be related to known parameters. Second, the volume of the motor needs to be found.

In looking at the output rotation consider the example of a planetary gear system. In this case the outside shell is the stator and remains stationary. The rotor is represented by the inside gear and is connected to an input shaft with a linkage. The linkage and input shaft are taking the place of the fluid which actual turns the rotor. Universal joints connect the rotor to the output shaft. In more modern PDM's the universal joint setup is replaced by a flexible shaft.
Figure 5 – Scheme for calculating output rotation of motor

![Scheme for calculating output rotation of motor](image)

For this case consider counterclockwise (CCW) motion as negative and clockwise (CW) motion as positive. If the input shaft makes one complete rotation CCW, then the linkage arm would have one complete rotation CCW. The arms rotation causes the rotor gear to rotate CW while the rotor center moves with the linkage in a CCW motion. The output shaft is attached to the rotor and consequently rotates CW.

Table 2 – Rotation table for Figure 5

<table>
<thead>
<tr>
<th>Arm</th>
<th>Stator</th>
<th>Rotor</th>
<th>Output</th>
<th>Arm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total</td>
<td>0</td>
<td>a/b-1</td>
<td>a/b-1</td>
<td>-1</td>
</tr>
<tr>
<td>Rel to Arm</td>
<td>1</td>
<td>a/b</td>
<td>a/b</td>
<td>0</td>
</tr>
</tbody>
</table>

The relative motion of the rotor to the arm is the gear ratio a/b where a and b also represent the number of lobes on the stator and rotor, respectively. The total output of the shaft is the same sign and magnitude as the total rotation of the rotor which coincides with earlier statements. The output rotation will in this set up always be clockwise because the ratio of a/b > 0.

Applying the results of this example to \( C = n/q \) produces

\[
C = \frac{a/b - 1}{q}
\]

To evaluate the volume of the stator stage, consider the second scheme. If the top of the rotor and stator are held rigid and the output shaft turns a/b-1 times, the stator would “untwist”. The rotor would be reduced to one pitch. In the power unit, the rotor has two pitches per stator stage. The volume would be conserved as the “untwisting” occurred.
Figure 6 – Scheme for calculating volume of motor stage

Let $A = \text{the difference in cross-sectional area of the stator and rotor}$

$L_s = \text{the length of one stator stage}$

Then the volume of one stage of the motor $q = AL_s$. Substituting this fact into $C$ results in:

$$C = \frac{a - b}{bAL_s}$$

Recognizing that $a-b=1$ for all cases in Moineau motor geometry:

$$C = \frac{1}{bAL_s} \quad (5)$$

Combining 4 and 5

$$N = \frac{Q}{bAL_s} \quad (6)$$

Under the same flow rate conditions, changing the volume of the PDM or the rotor/stator ratio can alter the rotational speed. If the volume is also held constant a motor with 1:2 ratio would operate at a higher speed than a 5:6 or a 9:10 motor. Generally it can be seen from the motor data in Table 1 that the 1:2 motors operate between 200 and 900 rpm, which at the bottom end of their performance is still rotating at the top end of the 4:5 and 5:6 motors' performance.

From equation 6 the volume per stage of the motor can also be determined. The Sperry-Sun 6 ¾” motor has a 1:2 rotor/stator relationship. The volume of a stage length of this particular motor would be:

$$500(2\pi) = 500/(1*AL_s)$$

$$AL_s = .159 \text{ gals}$$

This value could also be found from the constants in table 1 by applying equation 5.
Table 3 – Stage volumes of sample motors

<table>
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</tr>
<tr>
<td>C (rev/gal)</td>
<td>0.3365</td>
<td>1.698</td>
</tr>
<tr>
<td>(A_{Ls}) (gal)</td>
<td>0.0945</td>
<td>0.0937</td>
</tr>
</tbody>
</table>

In general, as the flow rate increases the rotational speed increases. If fitting a motor to an existing drilling system, equation 6 is useful to find out how fast the PDM will rotate with the given surface pump. On the design side, equation 6 is useful in determining the volumes and rotor/stator relationships for the given design parameters.

Another essential relationship for determining the performance of PDMs is the way torque is related to pressure.

Output Torque of PDM

The output torque will be discussed in terms of a 1:2 motor configuration. The simplest derivation of output torque is through the equation of energy balance between output mechanical energy per rotation of the output shaft and hydraulic energy taken from the fluid over the length of a motor stage. Consider a motor that is 100% efficient.

\[ TN = \Delta p Q \]

The unit of time cancels between flow rate and speed. For one rotation of \(2\pi\) and a volume of \(A_{Ls}\) for one revolution of the motor, the result would be:

\[ 2\pi T = A_{Ls} \Delta p \quad (7) \]

This gives an expression for output torque [2 & 3].

It is also useful to derive the output torque expression from a force balance consideration. The analytical derivation for the output torque stems from the analysis of the cross-sections in figures 7 and 8. Figure 7 is a diagram of two rotor stages and one stator stage. Fluid enters the top section at \(P_1\) and moves through the motor. The fluid exits at \(P_2\). The pressure drop across the motor can be observed from the standpipe, i.e. the difference in on and off-bottom pressures. \(P_1\) is greater than \(P_2\); otherwise the drilling mud would flow backwards through the PDM.

From figure 7, the source of the work can be inferred. Let the stator stage be considered as the control volume. Even though the rotor in the stator is moving, the actual volume between the two is constant. Refer to the scheme presented for deriving the output rotation to understand this concept more clearly. The work for a control volume is

\[ W = \int v \cdot dp \]

From the figure, it shows no pressure change over the control volume making \(dp = 0\) and thus the work within an entire stator stage is zero. If the control volume is changed to the last rotor stage, it is apparent that work is being done because a
change in pressure is present. The magnitude of this work is not as apparent. To obtain an expression for the work refer to figure 8.

Figure 7 – Rotor and Stage Lengths

Figure 8 – Cross-section of rotor and stator rotation

Figure 8 is useful to visualize the relative motion between the rotor and stator by fixing the centers $O_s$ and $O_r$ of each. The relative motion between rotor and stator remain the same as in the planetary motion arrangement. The circular cross-section of the rotor is captured in a typical angular position defined by $\phi$ while the angular position of the stator is defined by $\theta$. Note that in this case $\phi = 20$. In figure 8, $P$ represents the force produced by the pressure difference across the rotor cross-section. $F$ is the reaction of the rotor against the side of the stator cavity.

The figure cross-section can be understood in two ways. The first is as cross-sections of the rotor and stator taken at different distances $Z$ from the top of the power unit. Position 1 would represent $Z = 0$ and $Z = L_r$. Position 2 would represent $Z = L_r/4$ and $5L_r/4$, and position 3 would represent $Z = L_r/2$ and $3L_r/2$. Remember that the stator and rotor both have a helical shape.

The other way to interpret the figure is to think of it as the position of the top of the rotor as the stator rotates and the bottom of the rotor is fixed. Position 1 would be $\theta = 0^\circ$, $180^\circ$, and $360^\circ$. Position 2 would be where $\theta = 45^\circ$ and $225^\circ$, and position 3 would be where $\theta = 90^\circ$ and $270^\circ$. 
The motion of the rotor can be modeled as a sliding mass that is attached to a two bar linkage. One link is fixed between the centers of the stator and rotor circles. The distance between the centers is \(e\). The distance \(d\) that the rotor center is from the center of the stator circle is

\[
d = 2e \cos \theta
\]

**Figure 9** – dimensions of linkage from figure 8

The free-body diagram shows the torque about the center of the stator circle is

\[
T = 2F \cos \theta \quad (8a)
\]

**Figure 10** – Free-body of linkage assemble

Because equation 8a would hold true for all differential elements, it is better to express equation 8a as:

\[
dT = 2 \, dF \, e \cos \theta \quad (8b)
\]

From the geometry of the diagram:

\[
\tan \theta = \frac{F}{P}
\]

\[
F = P \, \tan \theta \quad (9a)
\]

Again it is more appropriate to express this equation in differential form:

\[
dF = dp \, \tan \theta \quad (9b)
\]

Combining equations 8b and 9b results in

\[
dT = 2e \, dP \, \sin \theta
\]

Where \(dP\), the force due to pressure, is \(\Delta p \, dZ\). The sum of all reactionary torques then will be equal to the torque on the output shaft. Thus the differential unit of torque is
In order to get the equation in terms of $Z$ use the fact that theta is by ratio $2\pi Z/L_s$. This relationship is coherent with the fundamental gear tooth law in that the ratio of the rotation of the rotor and the stator is a constant.

\[
dT = 2\pi e\Delta p (\sin \theta) dZ
\]

The limit $L_s/2$ is the length of one rotor stage. Note that if the integrand were from 0 to $L_s$ then the result would be 0. This derivation supports the previous discussion about the work done in the control volumes. The result implies that no torque or work is achieved over a whole stator stage. Thus just the first and last half stages or one whole stage of the rotor produce the work to rotate the bit. Despite this fact, most motors are made up of several stages. The extra stages act as a dynamic seal against leakage. The more stages the better the seal, but this number is limited by the resulting increase in friction due to the extra stages. Most motors range from 3-5 stages.

The cross-sectional area of the stator is $4e\delta$. Substituting into equation 10 yields:

\[
2\pi T = A L_s \Delta p
\]
Analytical Prediction of PDM Performance

The results of stand tests provide data on PDM performance. In typical testing a PDM is attached to a dynamometer. Pressure differentials for various torque settings are recorded at constant flow rates. Corresponding applied torques and rotational speeds are also recorded. This information is included in motor handbooks that are published by the manufacturer. One way the information is presented is in graphical form. Figure 11 shows the general relationship that torque and rotational speed have for a constant flow rate.

**Figure 11 – General performance curve of PDM**

When no torque from the dynamometer is applied the rotational speed is at its maximum. At this point the differential pressure over the motor is just large enough to overcome the friction with in the power unit. This point is the smallest differential pressure or no-load pressure for the motor. As the torque is increased with the dynamometer, the rotational speed begins to drop off because of leakage through the motor. The increase in torque also causes the differential pressure to increase. The point at which the maximum torque and minimum speed occur is the stall out point where 100% leakage occurs through the motor. The conditions and consequences for stall have already been discussed. In order to avoid stall conditions, a general rule of thumb is to operate around two-thirds of the maximum.

A similar performance curve for the 6 ½ " M1X Navi-Drill is shown in figure 12. The ratings and information that the manufacturers provide generally correspond to about ten percent leakage through the motor. Although this fact means that PDMs do not truly behave according to theory, the analysis presented in the paper is accurate enough for practical field applications as long as the motors are being operated within the recommended pressure and flow rate ranges.
The maximum operating differential pressure creates a torque that is roughly two-thirds of the stall torque. Calculated from the information provided by the manufacturer, the constant from equation 4 is 0.3365 for this specific motor. At the minimum flow rate, the power unit rotates at 90 RPM; at its maximum flow rate, the unit rotates at 220 RPM. These values are obtained from equation 4 or by correlating the manufacturer's performance data in table 1. From equation 2a, the maximum mechanical power produced by this motor would then be:

\[ P_{\text{mech}} = \frac{220 \cdot 2690}{5252} = 113 \text{ hp} \]

From equation 3, the power consumed by the motor under the maximum operating conditions above would be:

\[ P_{\text{hyd}} = \frac{465 \cdot 660}{1714} = 179 \text{ hp} \]

The overall efficiency of this motor would be:

\[ \eta = \frac{P_{\text{mech}}}{P_{\text{hyd}}} = 63\% \]

The same process can be applied to the other motors presented in the paper given their operating differential pressures. Efficiencies cover a large spectrum depending on the geometries and the age of the motor. As motors age the elastomer stator wears down which can allow more leakage than a new motor would permit. This leakage would reduce the overall efficiency of the motor. Yet for new motors, efficiencies can reach 80-85%.
System Analysis

In order to optimize the operating parameters of a PDM, an understanding of available power must be understood. At the surface of the drilling operation the surface pump supplies drilling mud through the drillstring to the PDM. The mud pump operates on a curve that balances flow rate for pressure. The pumps can operate at high pressures but low flow rates or at high flow rates and low pressures. Replacing the liners in the pumps can alter the pump's performance. Smaller liners would produce higher pressures and larger ones would produce larger flow rates. For example, a 1500 hp pump would have a performance curve similar to the one in figure 13. Even though the pump might max out at in the 10000 psi range most operators will run the pump at half or less in order to expand the life of the pump.

Figure 13 – Surface Pump Performance

![Graph showing Surface Pump Performance]

For example, a 1500 hp pump operating at 3000 psi can deliver the fluid at this pressure up to a maximum flow rate of 857 gpm.

The power that this fluid possesses is dictated by equation 1. In an average drilling situation, a surface pump maintains around 3000 psi in the standpipe. The fluid at this pressure can be delivered at different flow rates. As the fluid progresses through the drilling system, pressure is lost. This lost is referred to as parasitic pressure loss. The parasitic pressure is a summation of the pressure loss due to friction in the drill stem, friction in the annulus, and debris in the annulus. Experimentally it has been found that this loss is proportional to $Q^{1.86}$. The difference in the surface pump power and the parasitic power loss is the power available for driving the PDM and cleaning the bit (see Figure 14). The pressure and power loss here is based on a 5 ½" drillstring and a 9 7/8" whole/bit diameter for drilling mud with a density of 12 lbm/gal [5]. In general at the point of maximum available power the parasitic pressure losses will be approximately one-third of the supplied pressure from the surface pump [8].
\[ P_b = P_s - P_p \quad \text{where} \quad P_p = p_p Q = \text{const} Q^{2.86} \]

\[ P_b = p_s Q - \text{const} Q^{1.86} Q \]

Setting the derivative to zero gives the \( Q \) at which \( P_{\text{max}} \) occurs.

\[ 0 = p_s - 2.86 \cdot \text{const} Q^{1.86} \quad (11) \]

or substituting the definition of \( p_p \) back in results in:

\[ p_p = \frac{P_s}{2.86} \quad (12) \]

**Figure 14 – Hydraulic losses in system**

In general drilling conditions without a PDM, most operators try to maintain a flow rate at the point where maximum power is available at the bit for cleaning. By inspection this is around 600 GPM. Analytically this flow rate can be determined from equation 12.

\[ Q = \left[ \frac{P_s}{2.86 \cdot \text{const}} \right]^{1/86} \quad (13) \]

An appropriate PDM for this available power curve would be one that operates in the range of 400 to 700 GPM. The Navi-Drill M1X operates from 265-660 GPM; a good fit for the available power.
By adding the PDM performance curve to this chart, the operating range becomes obvious. The PDM curve is determined by the pressure drop data provided by the PDM manufacturers. The manufacturers also supply the minimum and maximum operating flow rates.

**Figure 15 – PDM Hydraulic Power usage**

![Graph showing hydraulic power usage](image)

From this point the other restraints become guess and check. The motor must operate in a range where a minimum annular velocity can be maintained. There must also be enough power left to allow an adequate pressure drop across the bit for hole cleaning. This parameter can be checked and results in a total flow area for the bit.

Next, the flow rate at which a minimum annular flow can be maintained needs to be found. Annular flow rates general run between 50 ft/min and 200 ft/min. A common maintainable annular velocity is around 120 ft/min. This rate provides adequate flow to move the cuttings away from the bit but not so much flow that it would erode the hole wall and enlarge the annulus.

\[
V_a = \frac{4Q}{\pi(D_h^2 - D_s^2)}
\]

\(D_h = \text{hole diameter}\) \hspace{1cm} \(V_a = \text{annular velocity}\)
\(D_s = \text{drillstring diameter}\) \hspace{1cm} \(Q = \text{flow rate}\)

The bit size range and thus hole size range for the motors under consideration is 7-7/8" to 9-7/8". From equation 11, the annular flow rates for a 9-7/8" hole would be:

<table>
<thead>
<tr>
<th>(V_a)</th>
<th>min</th>
<th>120</th>
<th>max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q(gpm)</td>
<td>137</td>
<td>329</td>
<td>549</td>
</tr>
</tbody>
</table>

By comparing these values to the flow rate range of the motors in chart 1, the annular velocity constraints on the PDM become evident. With the PDM usage shown, operating
at maximum hydraulic horsepower would provide ample flow to maintain the annular velocity.

The last restraint of fitting a PDM into an existing drilling system is the nozzle selection. The available hydraulic power for the bit is the difference between the PDM curve and available hydraulic power curve. If the pump operates at 600 GPM, the PDM consumes 163 hp out of the 683 hp available. This leaves 520 hp for the bit. This power correlates to a pressure drop across the bit of 1485 psi per equation (1a). The total flow area for the bit can be found from:

\[ TFA = \frac{\pi}{4} \left[ \frac{\rho \cdot Q^2}{6700 \cdot \Delta p_b} \right]^{1/2} \]  \hspace{1cm} (12)[5]

where \( \rho = 12 \text{ lbm/gal} \)
\( Q = 600 \text{ GPM} \)
\( p_b = 1485 \text{ psi} \)

\[ TFA = .518 \text{ in}^2 \]

This same procedure can be repeated in a more general manner by using graphs that plot TFA against varying flow rates. The chart would also show the annular velocity as a function of flow rate. By setting limits on the chart, the operating \( Q, V_a, \) and TFA can be found.

For the M1X to operate in a 9 7/8" hole with 5 ½" drill pipe with a surface pump capacity of 3000 psi and 600 gpm, the bit would need to have a TFA of .518 in² in order to use the most power that the pump is providing. This process could be repeated for any system and any PDM configuration. The total process is a balancing act between flow rate, system losses, bit size, and nozzle size.

Conclusions

Downhole motors and more specifically positive displacement motors are essential tools in bottom hole assemblies in drilling today. PDMs add a power boost or act as the sole means of rotation for the bit. They generally contribute 100-150 extra horsepower compared to the 23 hp generated by tradition drillstring rotation.

PDMs come in a variety of configurations making retrofitting one into existing situations confusing without proper knowledge of how to evaluate a PDM’s performance. Performance in general terms:

- Rotational speed is proportional to its flow rate. It can be increased by reducing the number of lobes on the rotor, increasing flow rate, or decreasing the volume per stage of the motor.
• Torque is proportional to the pressure drop across the motor. Torque can be increased by increasing the differential pressure, increasing the volume of the stage, or increasing the number of lobes on the rotor.

• The mechanical power produced is proportional to rotational speed and torque. It can be increased by increasing either rotational speed or torque.

• PDMs with a greater number of lobes generally produce more torque but operated a slower speeds than motors with just 1 or 2 lobes on the rotor. The power produced by the motors cannot be generalized in terms of rotor/stator configuration and is specific to each case.

• Efficiency drops with larger lobe configurations because of friction and fluid slippage.

• Before a motor can be chosen, the hydraulic power available must be calculated. The maximum available power is approximately two-thirds of the power initially supplied by the surface pump. The best motor makes optimal use of the available hydraulic horsepower.

• As a rule of thumb when deciding the flow rate at which to operate the PDM, it ought to consume near one-third of the hydraulic power available. This flow rate needs to be checked against the annular velocity to make sure it is large enough to provide adequate bit cleaning but not so large that it will erode the hole wall.

• The last step in fitting the PDM into the existing system is to chose an appropriate size bit nozzle. If the size nozzle turns out to be too large based on available equipment, try increasing the operational flow rate. If the size is still too large, the hole size may need to be decreased or a different PDM may need to be considered.

• Be careful not to operate the PDM outside of the manufacturer recommended operating conditions. Operating beyond these suggested limits puts the BHA at risk for stall out and thus detrimental damage. Also operating outside these limits decreases the accuracy of the equations presented.

• All equations assume 100% efficiency and no leakage. Although this is not realistic, it gives a good prediction and analysis of a PDM’s performance in normal operating conditions.

The processes presented in this paper are general and can be applied to any drilling situation and any PDM. It is essential to understand the performance possibilities and system limitations in order to truly incorporate the best PDM into an existing drilling system.
Nomenclature

- $A$ = cross-sectional area of void in power unit
- $a$ = number of lobes on stator
- $b$ = number of lobes on rotor
- $BHA$ = bottom hole assembly
- $D_h$ = hole diameter
- $D_s$ = drillstring diameter
- $e$ = distance between the center of the stator and rotor circles
- $L_s$ = length of stator stage
- $MWD$ = measurement while drilling
- $N$ = rotational speed
- $n$ = unit output shaft rotation
- $PDM$ = positive displacement motor
- $P_b$ = power at bit
- $P_p$ = power parasitic (power loss)
- $P_s$ = power from surface pump
- $p$ = pressure
- $p_b$ = pressure across bit
- $\rho$ = mud density
- $Q$ = flow rate
- $q$ = volume thru one stator stage of the motor
- $T$ = torque
- $TFA$ = Total flow area across bit
- $V_a$ = annular velocity
- $WOB$ = weight on bit
References


