FACTORS AFFECTING PUMP SELECTION FOR SLURRY-TYPE FUELS

BY WILLIAM DONAHUE AND J. DAVID BOURKE

MOYNO
Always the Right Solution™
FACTORS AFFECTING PUMP SELECTION FOR TRANSFERRING AND METERING SLURRY-TYPE FUELS

Introduction

Several factors affect the selection of pumps for use in handling composite or slurry type fuels. While capacity and differential pressure are the primary determinants of pump selection, the rheology and abrasive nature of slurry fuels effectively modify capacity and pressure on an absolute basis. Viscosity is a main determinant of maximum speed, the available NPSH, volumetric efficiency, slip, and power consumption. While knowledge of these factors is important, it is also crucial to realize that a pump and pipe system subjects the rheology of slurry fuels to a set of dynamic variables. In the selection of a pump and pipe system for slurry fuels, data must be considered which are obtained through analysis that considers this dynamism.

Depending on the percentage, size, and condition of solids, pipe size, and many other factors, slurry fuels generally exhibit non-Newtonian, homogeneous characteristics. A variety of data shows the diversity of results expected from the wide range of possible variables. Standard equations may be used for friction loss calculations in piping, as long as the user recognizes the marginal value over a relevant range of pipeline velocities caused by process or material variations. Data points from a standard rotational viscometer can be used to extrapolate an approximate “true” viscosity related to the dynamic conditions that affect pump performance, piping friction loss, and equipment life.

Other factors affecting pump selection include abrasion, corrosion, and temperature. These factors can be controlled by proper selection of materials of construction. An attempt to classify the abrasive effect of coal slurries, from varying locales is difficult. Abrasives may look alike, feel alike, and seem to have many similar properties yet have entirely different wear characteristics. Some broad classifications and guidelines are possible, primarily related to pump speed and materials of construction. With most metals, there is an inverse relationship between corrosion resistance and abrasion resistance. The use of elastomer parts solves much of this dilemma. Elastomers, however, can be attacked by various chemicals and swell or deteriorate in the process. Temperature can cause similar problems. Physical properties stability is the most crucial factor in elastomer selection.

Viscosity

For proper pump selection, the estimation of pump and pipe system pressures or NPSH available usually is difficult when handling abrasive slurries, since they normally fall into the category of non-Newtonian fluids. Full scale tests are usually costly and time-consuming, and in many cases only a small sample of the proposed slurry fuel is available.

The NPSH required by any particular pump model at a given speed is usually provided by data in manufacturers' literature. In a positive displacement pump, until the rotor, piston, lobe, gear, or vane closes behind the fluid and applies positive pressure to it, the pump can only create a void. The amount of fluid to flow into the void will (much like any orifice) depend on the fluid viscosity, differential pressure across the opening, and an entrance loss or K factor that reduces the theoretical flow (due to turbulence, friction, vena contracta, etc.). Assuming a negligible fluid vapor pressure and negligible flooded head or friction losses at the pump suction, the maximum differential pressure the pump could create by opening a void would be approximately 14.7 psi at sea level. Under these specific conditions, as long as the pressure drop between the suction port and the pump element entrance does not exceed 14.7 psi, fluid will fill the void and the pump flow will be full displacement. If the pressure drop between the suction port and the pump element entrance requires a pressure greater than 14.7 psi, cavitation occurs as the fluid pressure drops below the vapor pressure. A portion of the void is filled with fluid vapor which is condensed in the pump after the positive pressure is applied. The result is a pulsating, noisy, erratic flow and deviation from the straight line “capacity versus speed” curve. Obviously, the more viscous the fluid, the higher the pressure drop (or the lower the flow rate) at which cavitation will occur. Therefore, for a given pump model using known Newtonian fluids of various viscosities at various NPSHs, it is possible for a manufacturer to develop curves which indicate the maximum speed the pump should operate at a given NPSH available. Pump manufacturers generally have performed tests to determine additional pump drive or horsepower requirements for Newtonian viscosities. The only portion of a “horsepower versus differential pressure” curve that changes with the addition of viscosity is a part of the
constant friction portion (Figure 1). Pump manufacturers generally make available tables indicating maximum pump speed compatible with viscosities at given NPSH requirements and a horsepower additive table. These data are based on Newtonian liquids and are not applicable to non-Newtonian fluids. The Brookfield rotational viscometer can be used with various spindles and spindle speeds to determine the degree of non-Newtonian properties (Figure 2). Slurries are almost always non-Newtonian and will either increase or decrease their viscosity as they are "worked" by the shearing action of a pump and pipe system. There is usually an inverse relationship between shear rate and viscosity in slurry rheology although this rate is rarely linear. Most homogeneous slurries tend to be thixotropic—the viscosity decreases as the rate of shear is increased and as the length of time they are sheared increases. In these instances, you would find that even though the velocity in the pipe was the same at two different points, the viscosity of the slurry at one point would be higher at that moment than at a point somewhat farther down the line, because it has been subjected to shearing stresses of flow for a lesser time interval. This time-dependence of slurry viscosity is obviously troublesome when it comes to predicting pipeline specifications, or for that matter laboratory testing. It rules out data obtained through the recirculation of slurries in test loops.

![Figure 1. Horsepower vs. Differential Pressure](image1)

The non-Newtonian nature of the slurry fuels tested in our laboratory is shown in Figure 3. By plotting shear rate against shear stress and entering the data obtained with a Hercules viscometer onto a curve, the non-Newtonian nature of these fuels is evident. The "hysteresis loop" demonstrated by these fuels is true to the nature of thixotropic liquids. Once the material has been subjected to shearing, any additional shearing (even at lowered shear rates) will continue to reduce the viscosity of the material. Plotting a Newtonian liquid would give us a straight line and viscosity would be linearly related to shear rate. It is the time-dependence of thixotropic materials that causes this "looping" effect. In a practical sense, for pump selection we treat this material as a time-independent liquid. Where time-dependency does exist, an average value on the "hysteresis loop" is used.

![Figure 3. Time-Dependent Nature of Thixotropic Slurry Fuels](image2)

Probably one of the most convenient formulas for the prediction of non-Newtonian flow is the Power Law originally proposed by Ostwald. Where Newton's law for true fluids stated that the shear stress varied directly with shear rate, the Power Law states that for thixotropic, pseudoplastic, or dilatent fluids the shear stress varies directly as the shear rate to some power \( n \). That power factor is equal to 1 for Newtonian or true fluids, is greater than 1 for dilatent fluids, and less than 1 for pseudoplastic or thixotropic fluids (Figure 4).

Any function raised to a power will produce a straight line on a log-log graph. By plotting viscometer readings (either rotation or capillary or both) on such a graph, one can extrapolate to the nominal shear rate and pick off the apparent
NEWTON'S LAW OF FLUID MECHANICS

\[
\tau = \text{Shear Stress} \\
\gamma = \text{Shear Rate} \\
\mu = \text{Coefficient of Viscosity} \\
K = \text{Consistency constant} \\
(K = \text{viscosity when } \eta = 1)
\]

POWER LAW

\[
\eta = \text{Power Law Exponent} \\
\eta = 1 \text{ Newtonian} \\
\tau = K\gamma^n \\
\eta > 1 \text{ Dilatent} \\
\text{or Rheopectic} \\
\eta < 1 \text{ Pseudoplastic} \\
\text{or Thixotropic}
\]

**Figure 4. Formulas**

Viscosity that can be used in the Fanning equation to estimate pipe friction losses. If sufficient material is available for a pump test, this curve can be verified or altered by adding additional data points (Figure 5). These points are gathered by running what is termed a cavitation curve. Using Newtonian fluids of varying viscosities, such as silicone oil, it is simple to develop characteristic curves of point of deviation from the straight line capacity versus speed curve and the shape of the curve for each viscosity at a given NPSH.

**Figure 5. Viscometer Data**

When a non-Newtonian fluid is run under the same conditions, it is usually found that it matches none of the Newtonian curves, but it behaves like one viscosity at speed A, another viscosity at speed B, and a third viscosity at speed C (Figure 6). Knowing the average shear rate in the pump at those speeds, we then have additional points to affirm the fluid's adherence to the Power Law, and additional confidence that the pipe friction calculations will effect a practical degree of accuracy.

**Figure 6. Actual Capacity vs. Theoretical Displacement**

If the application warrants it and sufficient fluid is available, the fluid analysis may be further verified by checking the pressure drop through one pipe size and calculating the apparent viscosity using the Poiseuille formula. In 1965, Penkala and Escarfail showed the relevance of rotational viscometer readings to pipe friction losses using the Poiseuille formula. Their data were plotted on a log-log analysis chart because not only were there two different types of rotational viscometers used in gathering the data for three different slurries, (60% solids, 51% solids, and 45% solids), but these tests were run through three different pipe sizes (Figure 7). The grouping of the data points tends to confirm the redundancy of tests in more than one pipe size for friction estimations (at least in the shear rate range of most piping systems). The results from three different types of viscosity measurement approximate each other to a degree that would lend confidence to pump selection, particularly in the shear rate range of less than 200 seconds⁻¹.

Similar tests to these were performed on various coal slurries in our laboratory using a Brookfield rotating disc viscometer, a Hercules high shear coaxial cylinder viscometer, and pipe data from the test facility outlined in Figure 8. Samples of these data are illustrated in Figure 9. Interpolation of data for exact solids loading is possible to within several hundred centipoise, adequate for the selection of positive displacement pumps. Given the sizing of electric motors, moderate viscosity variations are not critical in pump sizing. Friction loss and increased pressure from viscosity, however, have a significant effect on pump horsepower requirements.

Worth showing at this time is the velocity profile within a pipe for various values of the factor “n” (Figure 10). For a true fluid where “n” equals 1, the flow pattern for streamline flow is parabolic, with the maximum velocity at the center of the pipe double that of the average velocity. For a dilatent fluid (where “n” is greater than 1) the larger the number, the closer the velocity approaches that of the purely theoretical perfect dilatent where “n” equals infinity. That shape is conical rather than parabolic and
Figure 7. Paper Color Coating Viscosity

Figure 8. Pump and Pipe Test Facility Used to Measure Viscosity by Friction Loss and Cavitation
Figure 9. Viscosity of Various Coal Slurries

NOTE: Even with the use of dispersion agents, settling rates produce erratic readings with low shear rate viscometers at lower solids concentrations.

*Slurry provided by CoalLiquid, Inc.
the maximum velocity at the center of the pipe is three times the average. This dilatent or rheoplectic phenomena ("n" greater than 1) does not normally exist in slurries. On the other hand, the perfect pseudoplastic, where "n" = 0, would flow as a solid plug, with no difference in velocity from the pipe walls to the center of the pipe. Even though a perfect pseudoplastic could not possibly exist, note that for a pseudoplastic or thixotropic slurry with a power factor "n" equal to .33, there is no appreciable difference in velocity over the inner 1/3 of the pipe, so the center portion flows as a plug.

Throughout the 45 years of experience Moyno has with slurries, several interesting and applicable phenomena have been discovered. Particles of reasonable density, under 300 microns, tend to flow in suspension while larger particles tend to move downstream faster than the fluid (such as heavy particles flowing down in a vertical pipe). They also migrate and concentrate around the walls of the pipe. When the particles tend to move downstream slower than the carrier, they migrate to the center of the pipe. Buoyant particles with the same density as the carrier flow in a ring surrounding the axis, about halfway between the center and the wall of the pipe. Changing the amount of fine particles in suspension changes the fallout velocity of a coarse particle. A small percentage of fine particles in water actually reduces friction head loss. Turbid water, therefore, flows at a greater velocity under the same head than clear water. Apparently, in a solution with a very small percentage of flowing solids, the particles act as guidevanes and reduce the water turbulence. When one considers that particles are lagging or leading the fluid, migrating into more concentrated areas, etc., it is apparent that even the percentage of solids is not constant at every point in the pipeline.

Viscosity effects pump requirements by both increasing necessary horsepower at the pump and increasing pressure through additional head loss. As explained above, both the pump and the pipe exert shearing forces on the slurry, helping to reduce the viscosity. Since the shear rate caused by the pump is usually different from the shear rate imparted by the pipe, pump sizing usually requires working with two separate apparent viscosities.

Average shear rates within positive displacement pumps are a function of speed and design constants, and can be calculated. By calculating shear rate, viscosity relevant to the internal pump mechanism can be estimated. The shear rates for Moyno® progressing cavity pumps are included in Figure 11. This shear rate can be used with the viscosity curves in Figure 9 to select the apparent viscosity used for horsepower additives, NPSH requirements and volumetric efficiency for the specific pump operating conditions.

<table>
<thead>
<tr>
<th>Pump Element</th>
<th>Capacity/100 Rev. @ 0 PSI</th>
<th>Shear Rate Sec. –1/100 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>.056</td>
<td>140.73</td>
</tr>
<tr>
<td>2</td>
<td>.260</td>
<td>93.95</td>
</tr>
<tr>
<td>3</td>
<td>.860</td>
<td>93.18</td>
</tr>
<tr>
<td>4</td>
<td>2.02</td>
<td>93.39</td>
</tr>
<tr>
<td>6</td>
<td>5.20</td>
<td>78.34</td>
</tr>
<tr>
<td>8</td>
<td>11.7</td>
<td>77.36</td>
</tr>
<tr>
<td>E</td>
<td>12.0</td>
<td>64.21</td>
</tr>
<tr>
<td>F</td>
<td>22.0</td>
<td>64.23</td>
</tr>
<tr>
<td>G</td>
<td>36.0</td>
<td>57.17</td>
</tr>
<tr>
<td>H</td>
<td>65.0</td>
<td>57.85</td>
</tr>
<tr>
<td>J</td>
<td>115.0</td>
<td>64.68</td>
</tr>
<tr>
<td>K</td>
<td>175.0</td>
<td>63.87</td>
</tr>
<tr>
<td>23</td>
<td>335.0</td>
<td>105.51</td>
</tr>
</tbody>
</table>

Figure 11. Shear Ratios for Moyno Progressing Cavity Pumps

Average shear rates within the pipe are a function of flow rate and diameter. This relationship is shown...
in Figure 12. The shear rate in the pipe determines the apparent viscosity of the slurry in the pipe; the shear rate from Figure 12 can be used with the viscosity curves in Figure 9 to determine the apparent viscosity within the pipe under the specific conditions of flow and pipe size. This apparent viscosity is used to determine pipe friction losses. Figure 13 allows determination of friction loss, in PSI per 100 feet, if flow rate, pipe size and the apparent viscosity are known. It is necessary to estimate viscosity both in the pump and the pipe to accurately estimate pump horsepower.

Once the pump and pipe system has been installed, it is possible to accurately measure flow without the aid of a flow meter. By measuring the system discharge at two or more pump speeds on a given liquid, the system can be calibrated and RPM can be linearly transposed into GPM. In positive displacement pumps, once the slip at any given pressure is overcome by a constant RPM setting, flow is directly proportional to speed. Calibration data can be transposed on dual-scale tachometers or hand-held programmable calculators, which are available for less than $500. Non-impeding, probe-less flow meters commonly cost several thousands of dollars.

**Abrasion**

It is extremely difficult to categorize the abrasive nature of slurries. Abrasion is affected by particle size, shape, hardness, and the equipment which is being used along with the solids percent loading of the slurry. While there have been attempts to classify the abrasive nature of slurries, such as measuring the wear on a platinum wire after it has rotated at a constant speed for a given time period, they usually have very little relevance, however, to what occurs inside positive displacement pumps which are required to operate at varying speeds over ranges as wide as 0 rpm to 3450 rpm. In an effort to reduce maintenance, as the abrasiveness of the pumped fluid increases the pump speed should be decreased. The amount of wear in an abrasive application is more closely proportional to the speed squared than it is to a linear relationship. Since coal is predominately a soft mineral, coal slurries are not found to be as abrasive as the harder silicas but speed should be reduced for maximum life. For progressing cavity pumps, for example, maximum speeds in the range of 400 to 450 rpm are recommended.

One detrimental effect that speed reduction may have on pump life may best be shown by drawing the effect of a fixed amount of wear on the performance curve for a speed “A” and a speed “B” that is one-half of speed “A” (Figure 14). It would take the pump at speed “B” almost four times as long to reach the wear curve. As long as the differential pressure is zero or very low, this equates to almost four times the life. Under pressure, however, it is obvious that the same amount of wear has a greater effect on volumetric efficiency at the lower speed than at the higher speed. At 100 psi, for example, the same amount of wear would have caused the flow rate to drop to zero at speed “B”, while at the higher speed “A”, the flow rate at 100 psi would still be in excess of 50% of the rate before the wear occurred. Then, although it would take almost four times as long to reach the same amount of wear at half the speed, the effect of wear on flow rate under pressure is more apparent at the slower speed as the pump elements could not be allowed to wear at the same degree without replacement. This effect partially negates the longer life expected by speed reduction. To compensate for this, pump pressure ratings should be derated and additional stages should be used on abrasive applications. This helps maintain high volumetric efficiencies under pressure at even the lower speed, as shown on the curve, reducing the effect of the wear on flow rate and thereby increasing the time between pump or parts replacement. Experience of recent years on coal slurries indicates that pump pressure capabilities should be double or triple that as required by the application as calculated by head, friction loss, and nozzle discharge re-
Figure 13. Pipe Friction Loss
quirements. For progressing cavity pumps this corresponds to ratings of 30 to 40 psi per stage on coal slurry fuel applications.

Material Selection

Most metals used in the transfer of coal based slurry fuels can be checked for corrosion with standard corrosion handbooks. Immersion tests can be performed but adequate results cannot be determined until a period of two to four weeks has elapsed. The tests performed in our laboratory indicate that for coal oil, coal water, and coal alcohol slurries, cast iron and carbon steel pump internals are adequate for corrosion resistance.

For best abrasion resistance, metallic parts should be constructed of the hardest material available. For positive displacement pumps which have the capability of trapping abrasive particles between a moving member and a stationary member (this includes progressing cavity, lobe, gear, vane, flexible impeller, piston, diaphragm, and many other positive displacement pumps) one or more parts should be constructed of a resilient elastomer and the others should be constructed of a very hard material. For best abrasion resistance, the most cost effective and convenient material is tool steel, air hardened to 55-58 Rockwell “C”, with a heavy layer of hard chrome plate. Carbide or ceramic coatings generally cannot match the abrasion resistance of chrome plate in rubbing friction applications. Also, steels that have been chrome plated can be stripped and replated if the wear is not too severe. Most other coatings require that the entire part be scrapped after it is worn.

If two intermeshing elements within a positive displacement pump are both constructed of inflexible metallics or ceramics, abrasive particles will become caught between the intermeshing elements and abrade them over time, or on non-lubricating water or alcohol based slurries, pumps may actually seize. When one of the elements within the pump is constructed of an elastomer, an abrasive particle which is caught between the elements will deform the elastomer rather than abrade either the elastomer or metal pumping elements. Once the crown of the sealing element (lobe, gear, screw, vane, etc.) has passed over the imbedded particle, the elastomer will resume its original shape and the abrasive particle will be flushed clear of the pump by the following liquid flow. If the elastomer is too soft, however, the volumetric efficiency under higher pressure suffers due to deformation. If too hard, it abrades rapidly. Optimum hardness range for a maximum life appears to be from 50 to 70 durometer (Shore A). The most abrasion resistant rubber compounds are based on natural rubber. Its lack of oil resistance eliminates its use on all COM applications and possibly some CWM applications that may use oil base stabilizing or viscosity reducing additives. Compatibility of any particular rubber compound can be checked by immersing sample disks of the various rubber compounds in the fluid for several days and checking for volume increase and/or durometer change. Assuming fluid compatibility with one of the better grades of available elastomers, and assuming maximum speeds and maximum pressure per stage have been properly selected, the results should be a maintenance free rotary positive displacement pump. Figure 15 shows the results of elastomer immersion tests on several different coal based slurry fuels. Natural rubber is indicated to be the best choice for coal alcohol mixtures. Experience indicates that Buna N is the best elastomeric compound to handle coal-oil mixtures. The durometer should be within ±10% of its original value and the volume of the rubber slug tested should be within ±5% of the original value after 14 days of immersion. Swelling is also common when the elastomer components are subjected to increased temperatures. Frequently, the first choice elastomer for abrasion and chemical resistance will be eliminated by its limited temperature capabilities. Figure 16 lists the maximum temperature capabilities of most commonly used elastomers in pumps. When these temperatures are exceeded, elastomers tend to break down and become brittle or revert. They lose their abrasion resistance and tensile strength. One way to counteract the effect of increasing horsepower as temperature or chemical attack causes rubber to swell is to decrease the size of the matching metal parts.
<table>
<thead>
<tr>
<th>Elastomer</th>
<th>Max. Temp. (°F)</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural Rubber</td>
<td>200°</td>
<td>Best for alcohol fuels and abrasion at moderate pressures.</td>
</tr>
<tr>
<td>Buna N</td>
<td>250°</td>
<td>Best choice for oils and moderate to high pressures for abrasion resistance.</td>
</tr>
<tr>
<td>EPDM</td>
<td>300°</td>
<td>Best for moderate to high temperatures and abrasion with wide resistance to chemical attack.</td>
</tr>
<tr>
<td>Fluoroelastomers</td>
<td>400°</td>
<td>Expensive, moderate to poor abrasion resistance. Good chemical and temperature resistance.</td>
</tr>
<tr>
<td>Urethane</td>
<td>180°</td>
<td>Excellent abrasion resistance. Water soluble above 150°F. Useful for COM.</td>
</tr>
<tr>
<td>Epichlorohydrin</td>
<td>275°</td>
<td>Moderate to fair abrasion resistance. Excellent choice for oil service. Could cause problems in high temperature water slurries.</td>
</tr>
</tbody>
</table>

Figure 16. General Elastomer Guidelines
Conclusion

Rotational viscometers can be used to practically extrapolate viscosity for pipe friction and pump performance calculations within applicable shear rate ranges. Friction losses in pipe can be calculated from these data using the Fanning equation for Newtonian fluids. These data can also be used to determine pump efficiencies, NPSH available, and NPSH required. Coal based slurry fuels should be considered moderately abrasive, and pump speeds should be 25 to 40% of the maximum speeds recommended on water. Pumps selected for coal slurries should also have pressure ratings two to three times higher than what will be encountered in the pump and pipe system. Cast iron housings and carbon steel internals are compatible with all of the slurry based fuels tested in terms of resistance to corrosion. Any internal rotating parts which are subjected to close tolerances or intermeshing with other parts should be constructed of hardened steels with a protective coating. Chrome plating is the most cost effective coating. It is commonly available, not easily damaged, inexpensive, hard, and can be reapplied with no damage to the structural or dimensional stability of the part's base metal. When two or more rotating parts intermesh, one part should be constructed of an elastomer. The elastomers will deform to absorb the abrasive particles rather than being abraded by them. Immersion tests can be performed on any elastomer if rubber compatibility is not known. Maximum temperatures must also be considered for elastomer selection and may affect the horsepower required for the pumping unit. Once these factors are identified and applied, it is possible to construct a pump and pipe system that will be relatively maintenance free for a long period of time. Flow is proportional to speed for positive displacement pumps and speed measurement can be used to indicate flow rates.

Adequate knowledge and application of the factors affecting pump selection for transferring and metering slurry type fuels can save the user considerable time and money. Initial equipment costs will usually be lower because the actual viscosity in the dynamic system is much less than what is normally measured in static laboratory tests, which indicate higher NPSH requirements and pressure losses than will actually occur. Maintenance costs can be reduced dramatically by applying units with adequate pressure capabilities that run at moderate speeds, thereby increasing pump life dramatically. Also knowledge of the factors occurring within the pump and pipe system and the knowledge of how a particular positive displacement pump performs can eliminate the need for other costly equipment.

References