PUMPING ABRASIVE SLURRIES WITH PROGRESSING CAVITY PUMPS

BY J. DAVID BOURKE

MOYNO

Always the Right Solution™

Always the Right Solution™

PUMPING ABRASIVE SLURRIES WITH PROGRESSING CAVITY PUMPS

The progressing cavity pump has been well established in the field of abrasive fluid handling. The totally sealed geared universal joints in the drive train have effected a major increase in progressing cavity pump life expectancy on abrasive fluids. Staging, materials selection and optimum speed selection, are important factors for maximum pump element life. Proper pump speed selection and calculation of system pressures can be difficult when non-Newtonian fluids are involved. Extrapolation of viscometer data using the power law has proved to be a practical method for pump selection.

INTRODUCTION

THE progressing cavity or single-screw rotary pump is a single helical rotor rolling eccentrically in a double threaded helix of twice the pitch length *(Figure* 1). In so doing, it forms a series of sealed cavities, 180 degrees apart, that progress from suction to discharge as the single helix rotates. As one cavity diminishes, the opposing cavity is increasing at exactly the same rate so the sum of the two discharges is a constant. The result is a pulsationless positive displacement flow with no valves.

The displacement, in addition to being a function of the speed, is directly proportional to three design constants; the cross-sectional diameter of the rotor (D), its eccentricity (e) or radius of the helix, and the pitch of the helix (Ps). Its pressure capabilities are a function of the number of times the progressing seal lines are repeated. For example, a single-stage element may be capable of pumping efficiently against 75 psi, and by tripling the length and thereby tripling the number of seal lines, the pump is capable of operating as a three-stage unit just as efficiently at 225 psi *(Figure* 2).

Note that although it's a relatively complex configuration, the flow through the elements is not far removed from the straightest distance between suction and discharge. The result is relatively low velocity and shear for a given displacement and, therefore, excellent capabilities for handling highly viscous and sensitive slurries. Properly applied, the average shear rates in a progressing cavity pump can be less than 100 sec.^{-1} An excellent application for example would be the pumping oil and water mixtures without emulsification.

Another feature that gives this pumping principle its advantage for slurry handling is the use of elastomers as the outer gear in at least 95% of its applications. Through the use of a compression fit between the rotor and the stator (much like an "O" ring seal along the elements), the clearance between the elements required by gear or lobe pumps has been eliminated, thus making the pump capable of pumping low viscosity and gaseous fluids, as well as high-viscosity fluids. It can even be used as a compressor if a lubricant is added. The elastomeric outer gear also adds abrasion resistance beyond that of conventional rotary pumps. The particles tend to imbed rather than abrade. It also allows deformation to partially accommodate large solids such as rocks, rags, or tramp metal. There are, as previously mentioned, no valves to foul.

As a result, almost every conceivable fluid-like material can be pumped, including: sandy crude, coal slurries, waste sludges, polymers; drilling mud; tallow and grease; sealant; asphalt emulsions; paper stock; refractory mortar; gypsum roof deck, and caulking compound.

Rarely is a progressing cavity pump applied without some degree of abrasion. The abrasion-resistant feature of the progressing cavity pump has done more to effect its steady and diversified growth than any other single feature. After 40 years of successes and failures on what may be the most diverse applications of any pumping principle in the world, the real secrets to pump longevity or abrasion resistance are (in addition to basic design) materials selection and pump application.

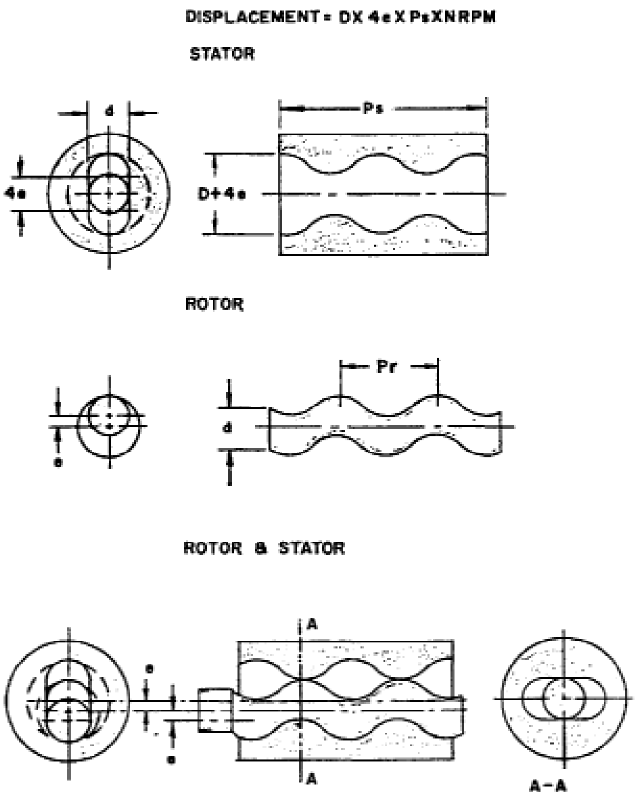

Figure 1—Progressing cavity pump is a single helical rotor rolling eccentrically in a double threaded helix of twice the pitch length

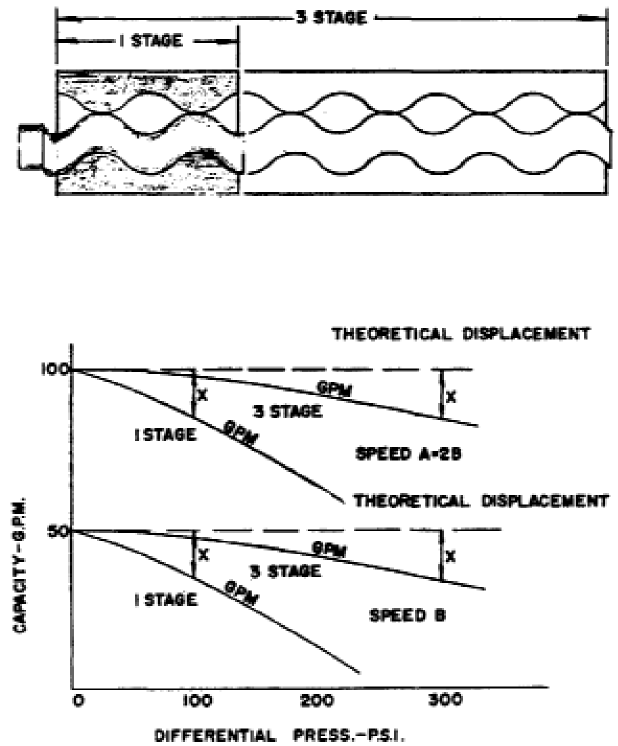

Figure 2—By tripling length and number of seal lines of a single-stage element, a three-stage unit is developed which is capable of operating at triple the pressure

DESIGN AND DEVELOPMENT

An excellent case history is when progressing cavity pumps first entered the waste treatment field approximately 20 years ago, the design and application was such that 2000 hrs between repairs was a normal expected pump life on settled sewage sludge. Now that the progressing cavity pump has become almost the standard positive displacement pump in waste treatment plants, users are documenting life in excess of

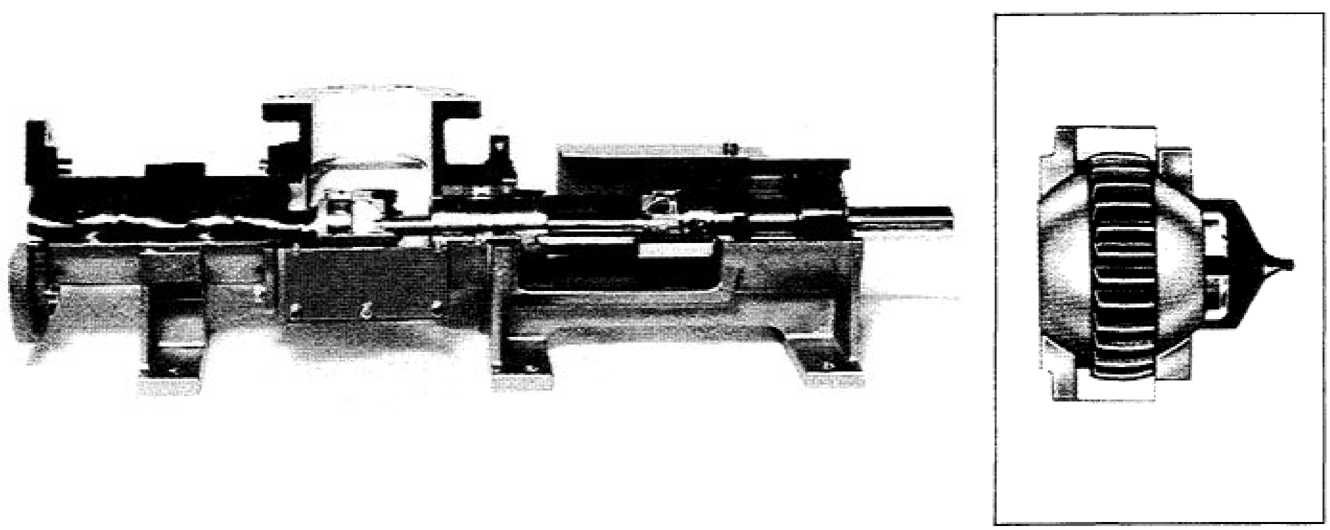

Figure 3—Cutaway illustration of a progressing cavity pump designed for abrasive sludge and slurry applications. At right is enlarged cutaway of gear-type universal joint used in progressing cavity pumps to combat excessive wear that was destroying ball and pin-type designs

27,000 hrs before the first repair. The benefits of such improvements produced sales in excess of 100,000,000 for waste sludge pump applications from 1973-1979.

There is little doubt that if the improved longevity had not been accomplished, the rewarding increase in market penetration would not have been realized. This market is probably the best example of success for progressing cavity pumps, but the same principles of continuing improvement in life expectancy are applied on abrasive fluids in all areas of endeavor from sandy crude to oil muds—that each time a pump is applied, a little more knowledge about the possible pitfalls and methods to avoid them is collected.

Pump design or redesign and product improvement, of course, are contributing factors to improved life on abrasive applications. One basic change in design in the waste treatment field, for example, was the introduction of a gear-type universal joint *(Figure* 3). During operation, the center of the rotor or inner gear of the progressing cavity pump moves in an eccentric path opposite to the direction of shaft rotation. This necessitates the incorporation of two universal joints operating in the slurry between the drive shaft and rotor. One manufacturer used a hardened pin and slotted ball joint design from the date of the original license (over 40 years). In abrasive service, although the ball and pin joints are initially packed in grease, the slurry eventually works through the lip-type seal washer, and with time can contribute to maintenance expense. The gear drive incorporates a crowned gear driving mechanism with spherical thrust plates that distribute both torque and thrust loads over a surface, as opposed to the line contact of conventional ball and pin-type universal joints. Besides reducing the stresses, the joint is positively sealed from slurry entrance. A rubber bellows that is flexed only one degree close to the fulcrum point of the joint is clamped on both I.D. and O.D. Universal joint maintenance was virtually eliminated by use of this design.

PUMP APPLICATION

Equally significant in overall reduction of maintenance over the years on abrasive applications has been improved pump selection—selecting the proper pump, proper number of stages, proper speeds, proper materials of construction. The starting point is speed. The more abrasive the slurry, the slower the speed. The amount of wear on an abrasive application is more closely proportional to the speed squared than it is to a direct relationship. Whereas 40 years ago pumps were sold to apply at 1800 rpm for clay slurries, catalogs now recommend maximum speeds of 600 rpm for abrasive slurries and, where pump longevity is desired, speeds are more commonly in the 300 to 350-rpm range. The resulting life increase has astounded many.

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

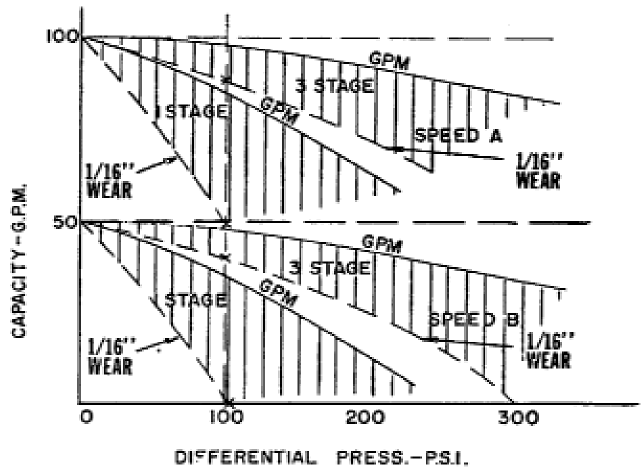

Figure 4—Effect of wear on progressive cavity pump performance

curves shown previously for a speed A and a speed B that is half of A *(Figure* 4). It would take the pump at speed B almost four times as long to reach the curves. As long as the differential pressure is zero (or very low), this equates to almost four times the life. However, under pressure it is obvious that the same amount of wear has a greater effect on the 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 the wear on flow rate under pressure is more apparent at the slower speed and the pump elements could not be allowed to wear to the same degree without replacement. This effect partially, or even (in extreme cases) totally, negates the longer life expected by speed reduction. To compensate for this, more stages are used on abrasive applications. This helps maintain high volumetric efficiencies under pressure at even the lower speeds, as shown on the curve reducing the effect of the wear on flow rate and thereby increasing the time between element replacement. This is, in effect, reducing the pressure per stage. The earliest literature rated these pumps at 100 psi per stage, while today manufacturers recommend a maximum of 75 psi/stage on nonabrasive fluids and as low as 10 to 20 psi maximum per stage on abrasive applications when long life is desired.

MATERIALS SELECTION

Another area of great importance to pump life on abrasive slurries is the proper selection of materials of element construction. For best abrasion resistance the rotor or inner gear is fabricated from an air hardened tool steel, 55-58 Rockwell “C”, with a heavy layer of hard chrome plate. This represents the best abrasion resistance available. Over the years, thousands of dollars have been spent investigating various coatings, including tungsten and silicon carbides, stellites, aluminum, titanium and chromium oxides, and many others. A coating has yet to be found for the rotor that will match a hard chrome plate for abrasion resistance, when properly applied over a base metal of the proper hardness. This also affords a relatively inexpensive method of resurfacing for repair. Although the rotor is available also in 316 stainless steel, the softness greatly reduces abrasion resistance in spite of the layer of hard chrome.

Compatibility with the fluid pumped most normally dictates the materials of construction for the stator or outer gear. Unless the hardness of the abrading particle is softer than hardened tool steel, metal or rigid stators are not recommended. As mentioned previously, the most abrasion resistant combination includes an elastomeric stator. If the elastomer is too soft, the volumetric efficiency under pressure suffers due to the ease of deformation. If it is too hard it abrades too rapidly. Optimum hardness range for maximum life is from 50 to 70 durometer (Shore A). By far, the most abrasion-resistant rubber compound is a high grade of natural rubber. Its lack of oil resistance, etc., eliminates its use in many applications because of an incompatibility with the fluid pumped. Second in abrasion resistance and first in overall compatibility and use is a grade of 65 durometer Buna N, a medium high acrylonitrile-butadiene rubber with excellent grease, oil, and chemical resistance. Unfortunately, the rubber industry has yet to come up with an elastomer resistant to all of the solvents. Fluorocarbon rubbers show fair resistance to some of the hydrocarbons such as toluene or benzene, but have abrasion resistance below that of the previously mentioned compounds and are extremely expensive. It also has no resistance to ketones. Ethylene Propylene Diamine rubber, on the other hand, has good ketone resistance, fair mechanical properties, reasonable cost, but has no resistance to petroleum hydrocarbons. Obviously, the all-purpose stator for the paint industry does not exist. Compatibility can be checked by immersing sample discs of the various rubber compounds in the fluid for several days and checking for size and/or durometer change.

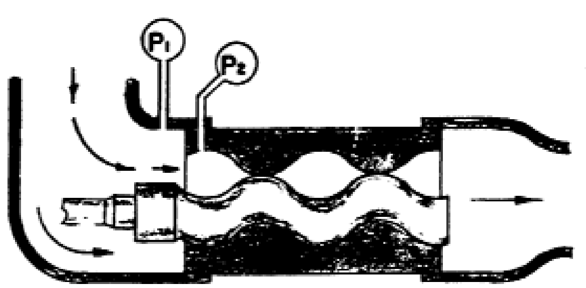

Figure 5 — Amount of fluid to flow into void depends on pump size, pump speed, fluid viscosity, differential pressure across opening, and an entrance loss or K factor that reduces the theoretical flow

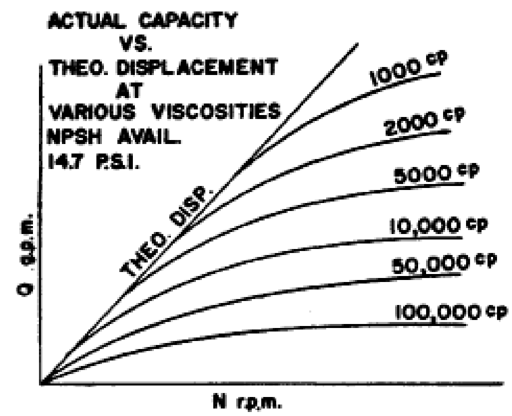

Figure 6 — Curves indicating pump cavitation speeds for Newtonian fluids of varying viscosities at a given NPSH available

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 result should be maintenance-free progressing cavity pump life exceeding that of other types of positive displacement pumps.

At times, however, the estimation of the system pressures or net positive suction head available for proper pump selection in itself presents a problem. This is especially true in the handling of abrasive slurries, since they normally fall into the category of non-Newtonian fluids. Obviously, not only would full-scale tests be costly and time-consuming on a “one-time” application, but in many instances only a small sample is available. A fast, easy, empirical method of fluid evaluation was developed that would enable proper selection of pump speeds, drives, etc., without an actual pumping test.

For Newtonian or true fluid samples, the problem was simple. Viscometer readings at various shear rates will prove its Newtonian nature, and hydraulic handbooks abound with pipe friction data for viscous fluids. The problem became simply to find the effect of viscosity on the Net Positive Suction Head required and power requirements for the various models. In a positive displacement pump, until the rotor or piston closes behind the fluid and applies positive pressure to it, the pump can only open a void. The amount of fluid to flow into the void will (much like any orifice) depend on the fluid viscosity, the differential pressure across the opening, and an entrance loss or K factor that reduces the theoretical flow (due to turbulence, friction, vena contracta, etc.) as shown in *Figure* 5. 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 then, as long as the pressure drop between the suction port and the pumping element entrance does not ex-

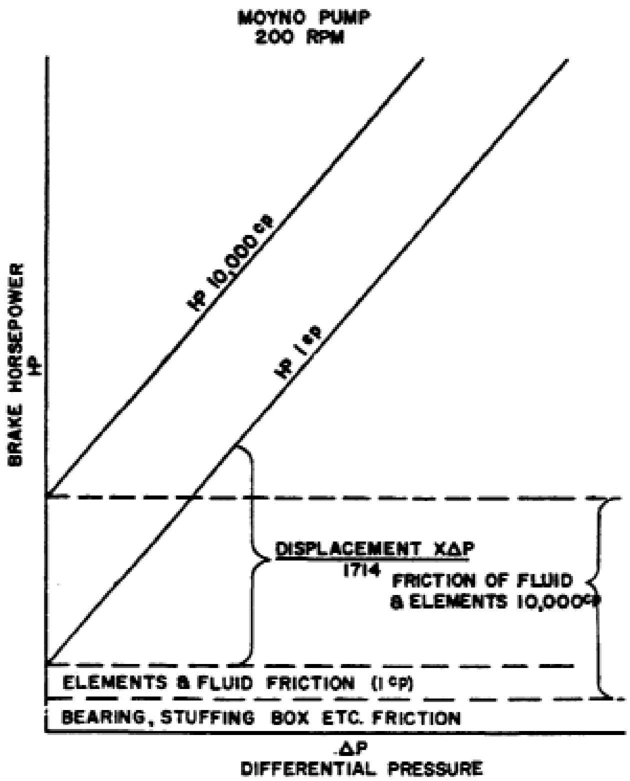

Figure 7—Only portion of "horsepower vs. differential presure" curve that changes with viscosity change is part of the constant friction portion

ceed 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 pump element entrance requires a greater pressure than the 14.7 psi available for full displacement flow to enter, 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 rotor applies positive pressure. The result is a pulsating, noisy, erratic flow and a deviation from a straight line, "Capacity vs. 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 (such as silicone fluids) of various viscosities at various Net Positive Suction Heads, it was possible to develop curves such as in *Figure* 6 which indicate the maximum speed the pump should operate at a given NPSH available.

A series of tests to determine additional pump driver power requirement for Newtonian viscosity was also performed. The only portion of the "horsepower vs. differential pressure" curve that changes is part of the constant friction portion *(Figure* 7). It is then possible to make available to the selector two more tables indicating maximum pump speed compatible with viscosity or NPSH and a horsepower additive table to go with the tables for maximum speed and pressure per stage for abrasion. With a good pipe friction table for viscous fluids, the pump selector's tools are complete.

The first thing a pump selector finds after he's been thoroughly educated in the use of these tables is that anyone pumping Newtonian fluids is using a less expensive gear pump. A survey of the current literature of homogeneous and heterogeneous flow of non-Newtonian fluids leaves little doubt that any friction loss calculations without extensive testing on the fluid in question would have to be risky. Fortunately, most of our areas of concern have to do with viscous fluids, and flow is laminar or streamline.

Zandi[1] states that "a literature survey clearly indicates that a universally acceptable technique for predicting the head loss of turbulent flow of non-Newtonian suspensions and fluids in pipe does not exist." A literature survey showing the validity of extrapolation of tests run on a pint sample into results in a pipe and pump system is also quite limited (in spite of its predicted laminar flow).

In 1957, attempts were made to predict from viscometer readings the actual performance in the pipe system. Originally, there was no attempt at correlation between the nominal or apparent viscosity of the fluid at viscometer shear rates and pump system shear rates. The Brookfield viscometer was used with various spindles and spindle speeds to determine for the most part the degree of non-Newtonian properties *(Figure* 8). Friction losses were calculated using Poiseuille's formula for laminar flow, since (as mentioned previously) flow was almost entirely laminar or viscous on applications where friction losses were of concern to us and an apparent viscosity "somewhat" lower than the apparent viscosity of highest viscometer shear rate (in the case of pseudoplastic or thixotropic fluids) was used. Since shear rates in pumps and piping were considerably greater than those in the Brookfield viscometer, maximum pump speeds recommended were usually lower than need be, and piping pressure drop calculations were considerably higher than would be expected to exist in the system. It was noted of data from the great number of fluids plotted on log-log

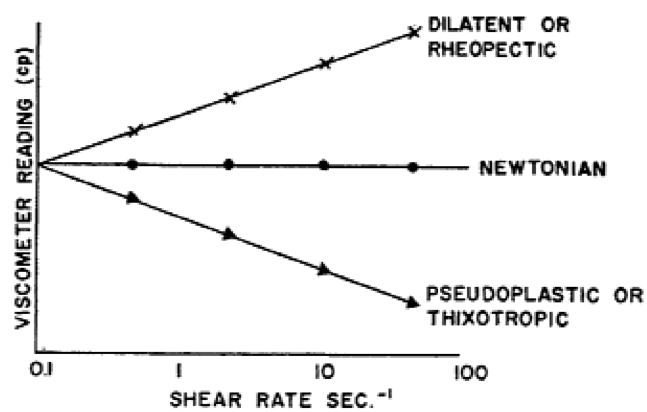

Figure 8—Brookfleld viscometer was used with various spindles and spindle speeds to determine degree of non-Newtonian properties

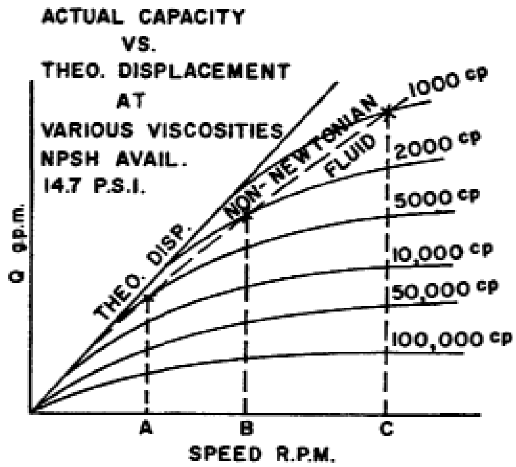

Figure 9—Comparison of cavitation test curve for non-Newtonian fluid with Newtonian fluid curves

paper that lines were, in the majority of cases, relatively straight, indicating that at least over the shear rate range of the viscometer the majority of fluids submitted for analysis followed the power law originally proposed by Ostwald[2]:

$$\tau = K\gamma^{n}$$

$$\text{since } \mu_N = \frac{\tau}{\gamma}$$

$$\text{then } \mu_N = K\gamma^{n-1}$$

where $n = 1$ for Newtonian fluids

$n > 1$ for Dilatent fluids

$n < 1$ for Pseudoplastic fluids

τ = shear stress

γ = shear rate

μ_N = nominal viscosity

The natural progression, then, in 1963, led to an attempted extrapolation of viscometer figures to the average

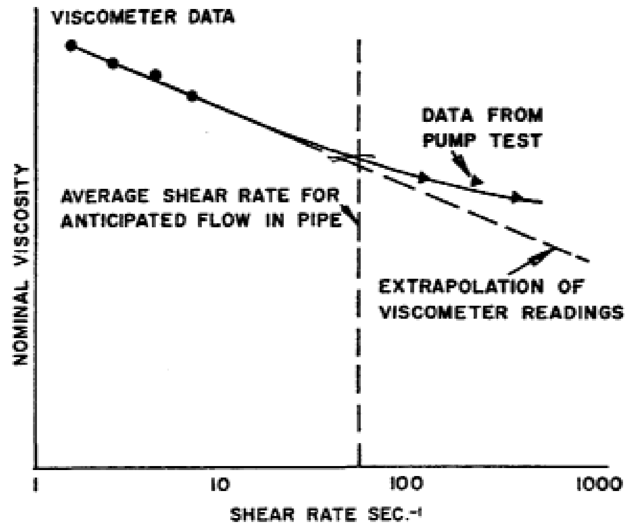

Figure 10—Intercept points of Newtonian and non - Newtonian fluid curves provide additional points of apparent viscosity at various pump shear rates for comparison with viscometer curve extrapolation

shear rate value in the pump and piping, fully aware of the possible pitfalls. It is known that: (1) very few fluids follow the power law over a wide range of shear rates; (2) the correlation between rotational viscometer readings and apparent viscosity in pumps and piping is highly questionable. Nevertheless, in those instances where we have been able to run pumping tests or verify data on the subsequent installation, we have found the results surprisingly close in the greater majority of the cases. In those cases where the quantity of material is available and

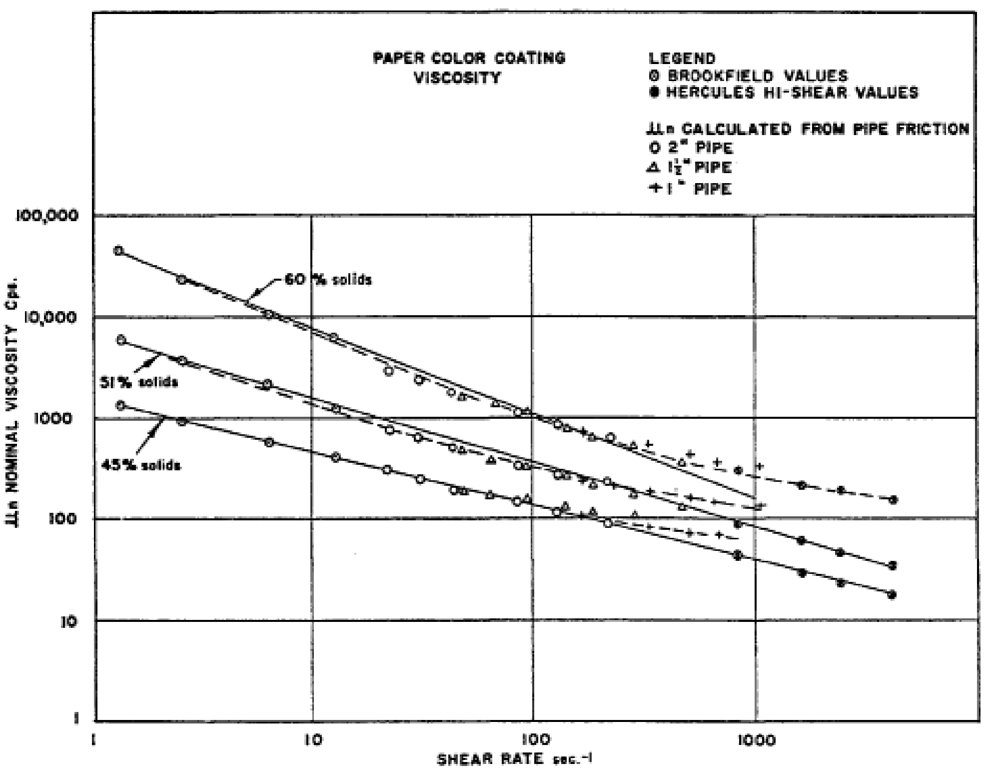

Figure 11—Plot showing relevance of rotational viscometer readings to pipe friction losses

the application justifies running tests, a cavitation or speed vs. capacity test at a given Net Positive Suction Head is run and compared to the previously mentioned Newtonian fluid curves *(Figure* 9).

The average shear rate within progressing cavity pump elements is a function of the speed and design constants and can be calculated. The points of intercept of the non-Newtonian fluid curve with the curves for Newtonian fluids then gives us additional points of apparent viscosity at various pump shear rates to reinforce or alter our viscometer curve extrapolation *(Figure* 10).

If the application warrants it (and sufficient fluid is available), the fluid analysis may be further solidified by checking pressure drop through one pipe size and calculating the apparent viscosity using Poiseuille formula. In 1965, Penkala and Escarfail[3] showed the relevance of rotational viscometer readings to pipe friction losses using the Poiseuille formula. Their data was plotted on our log-log analysis chart, because not only have they used two different types of rotational viscometers in gathering their data for three different slurries (60% solids, 51% solids, and 45% solids), but they have run their tests through three different pipe sizes *(Figure* 11). As mentioned previously, our tests are seldom run in more than one pipe size. The grouping of their 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). Also, note that results from three different types of viscosity measurement, rotating disc, coaxial cylinder, and pipe flow approximate each other to a degree that would lend confidence to the pump selector, particularly in the shear rate range of less than 200 sec^{-1}. In addition to two types of rotational viscometer in the testing lab, a special high-pressure (nitrogen gas) extrusion type viscometer with capillaries up to 1 in. diameter is helpful. The analogies between pipe flow and capillary viscometry is not as strained as with rotational viscometers and the large capillary tubes allow measurement of heterogeneous fluids with larger particle size than rotational viscometers will allow. Consistency. in the data obtained from the two methods of viscosity measurement of relatively small samples assures that time-consuming pump running tests are unnecessary for proper selection. Log-log fluid analysis curves for materials ranging from Tootsie Rolls to toothpaste now number in excess of 1000, and most small samples of fluids or slurries are handled routinely and with confidence of reasonable accuracy of pump and piping selection.

SUMMARY

In summation, then, technologies have been developed to predict with reasonable accuracy the reaction of abrasive fluids and slurries in pipe lines and pumping systems. Using this data to properly select the progressing cavity pump with special attention to the conservative selections of maximum speeds, adequate number of stages for good volumetric efficiencies under even conditions of wear and proper materials of construction can change what has always been a device noted for its abrasion resistance, the progressing cavity pump, into an almost maintenance-free system. ◆

References

(1) Zandi, I. “Hydraulic Transport of Bulky Materials,” Pergamon Press, New York, 1971.

(2) Ostwald, W. “Kollardzschr,” *38,* 261 (1926).

(3) Penkala, J. E. and Escarfail, J. P., “Flow of Coating Color in Pipes, Fittings, and Pumps,” *T.A.P.P.I., 48,* (1965).