

HANDLING HIGH SOLIDS CONTENT NON-NEWTONIAN FLUIDS

BY J. DAVID BOURKE



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HANDLING HIGH SOLIDS CONTENT NON-NEWTONIAN FLUIDS

There are many ways to "handle" high solids content non-Newtonian fluids - pumps, conveyors of various types, even buckets. Simply, our methods of pump and speed selection, estimation of pipe friction losses, etc., for non-Newtonian fluids, and most specifically, our methods for handling drum filter cake or centrifuge discharge are what will be discussed here.

Hopefully, few are unfamiliar with the Moyno® or single screw rotary pump and there is no need

to spend much time describing its method of operation. By definition, it is a single helical rotor rolling eccentrically in a double threaded helix of twice the pitch length (Fig. 1). In so doing, it forms a series of sealed cavities, 180° 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.

placement flow.

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 we repeat these progressing seal lines. For example: a single stage element may be capable of pumping efficiently against a pressure of 75 psi. By tripling the length and thereby tripling the number of seal lines, the pump is capable of operating just as efficiently at 225 psi, as a three stage unit (Fig. 2).

You will note that although this is 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, therefore excellent capabilities for handling highly viscous and sensitive slurries.

Another feature that gives the principle added advantages for slurry handling is the use of elastomers for the outer gear in at least 95% of our applications. Through the use of a compression fit between the rotor and the stator (much like an "O" ring seal along the elements), we have eliminated the clearance between the elements required by gear or lobe pumps; therefore, the pump is capable of pumping low viscosity and gaseous fluids, as well as highly viscous fluids. The pump can even be used as an air 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. The elastomeric gear also allows deform-

FIGURE 1

$$\text{DISPLACEMENT} = D \times 4e \times P_s \times \text{NRPM}$$

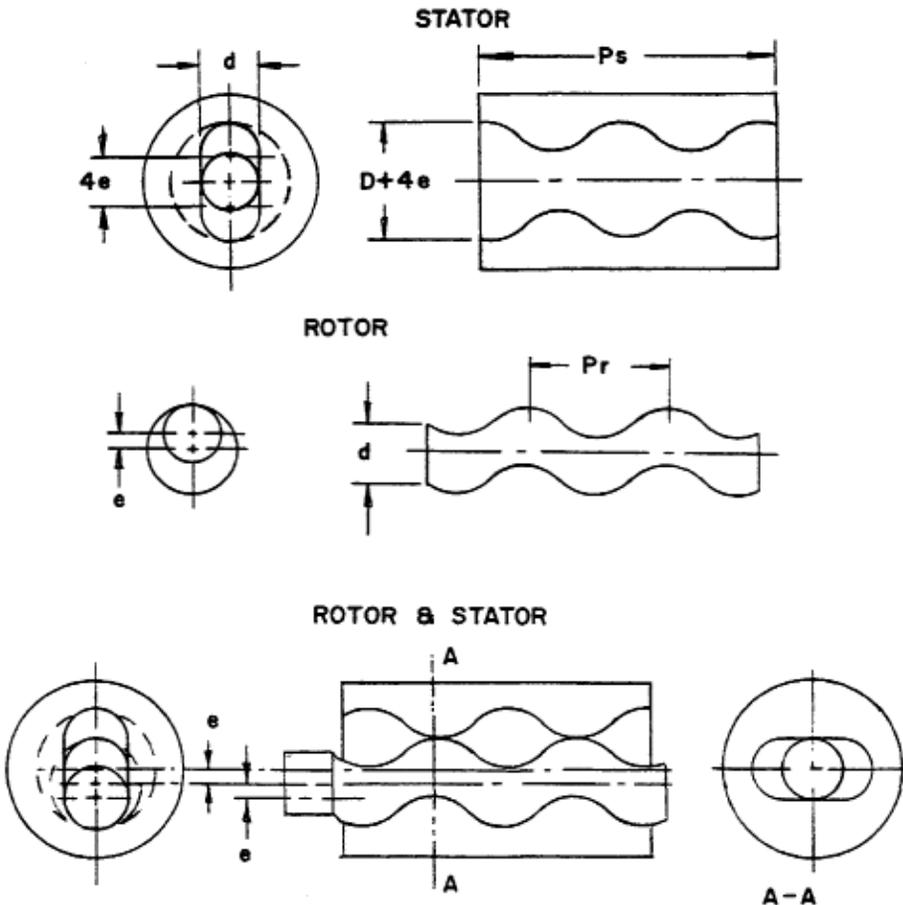
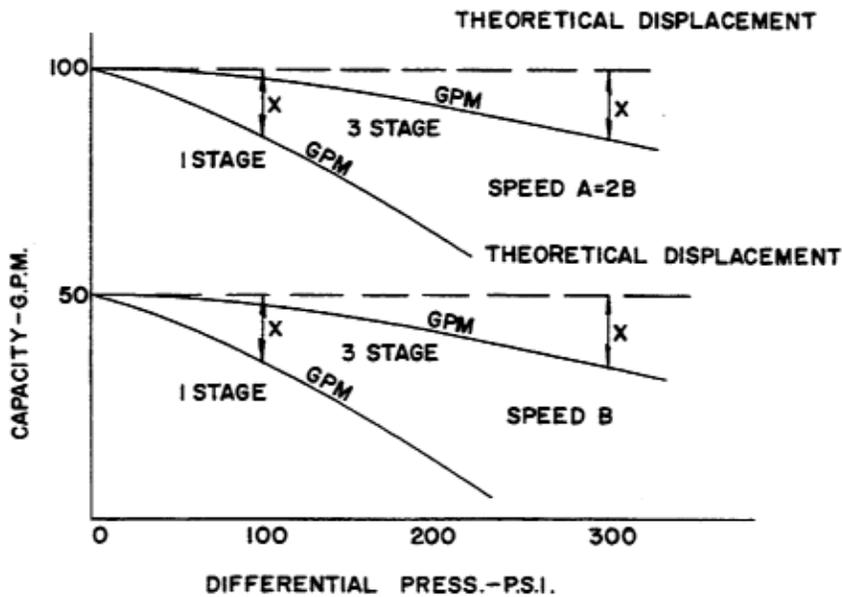
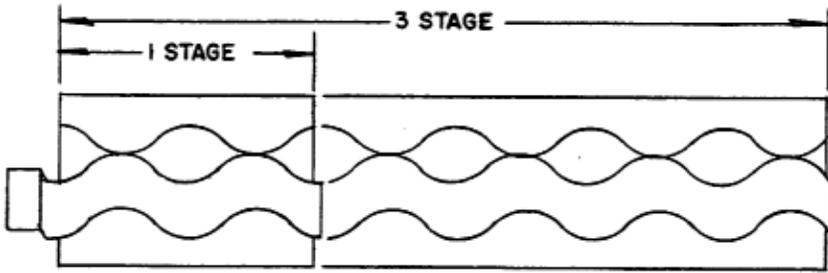


FIGURE 2



ation to partially accommodate large solids such as rags, nuts and bolts, or tramp metal. There are no valves to foul. All of these features led to such chauvinistic statements in our earlier days as, "If it can be pushed through a pipe, we can pump it".

Considering our diverse applications, it became necessary for us to develop a fast, easy, empirical method for fluid evaluation that would enable us to select proper pumps, speeds, drives, etc., without an actual pumping test. Not only could full scale tests be costly and time consuming on a "one-time" application; but, in many instances only a small sample of the liquid would be available.

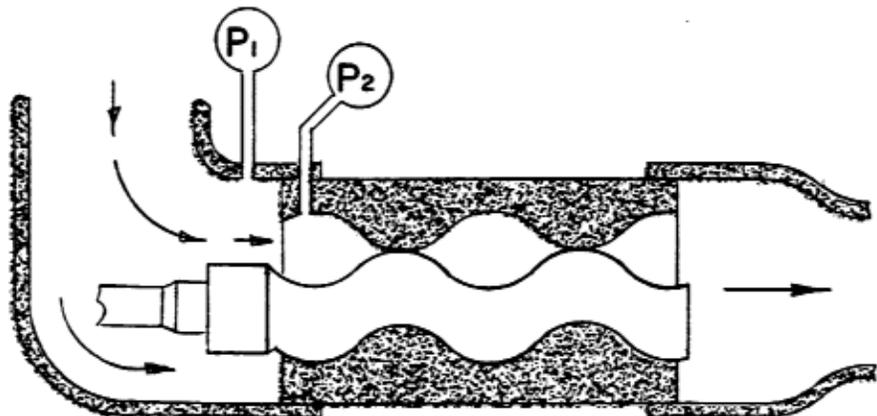
For Newtonian or true fluid samples, the problem is simple.

Viscometer readings at various shear rates can prove a fluid's Newtonian nature; and, hydraulic handbooks abound with pipe friction data. The problem was to find the effect of viscosity on the

Net Positive Suction Head required and the power requirements for various pump models. In any 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 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.), (Fig. 3).

Assuming a negligible fluid vapor pressure and negligible flooded head or friction losses in the pump suction line, 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 pumping 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 pump element entrance requires a greater pressure than the 14.7 psi available for full displacement flow to enter, cavitation obviously will occur 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 closes behind it and applies positive pressure. The result is a pulsating, noisy, erratic flow and deviation from a straight line, "Capacity vs. Speed" curve.

FIGURE 3



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 oils) of various viscosities at various Net Positive Suction Heads, it is a simple task to develop curves such as the solid line curves in Figure 6, which indicate the maximum speed the pump should operate at a given NPSH available.

A series of tests to determine the additional pump driver power requirement for a Newtonian viscosity is also quite simple. The only portion of the "horsepower vs. differential pressure" curve that changes is part of the constant friction portion (Fig. 4).

These values for maximum speeds, horsepower additives to pump water curves, along with pipe friction tables for viscous fluids, furnish the necessary tools of proper pump selection for Newtonian fluids.

Unfortunately, though, most slurries and the great majority of Moyno applications are non-Newtonian fluids. A survey of the current literature on homogeneous and heterogeneous flow of non-Newtonian fluids leaves little doubt in one's mind; however, that any friction loss calculations, without extensive testing on the fluid in question, would have to be risky at best. Fortunately, most of our areas of concern have to do with viscous fluids; and, their flow is laminar or streamline.

Zandi¹ 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."

In 1957, we began our first attempts 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 and system shear rates. The Brookfield viscometer

was used with various spindles and spindle speeds to determine the degree of non-Newtonian properties (Fig.5).

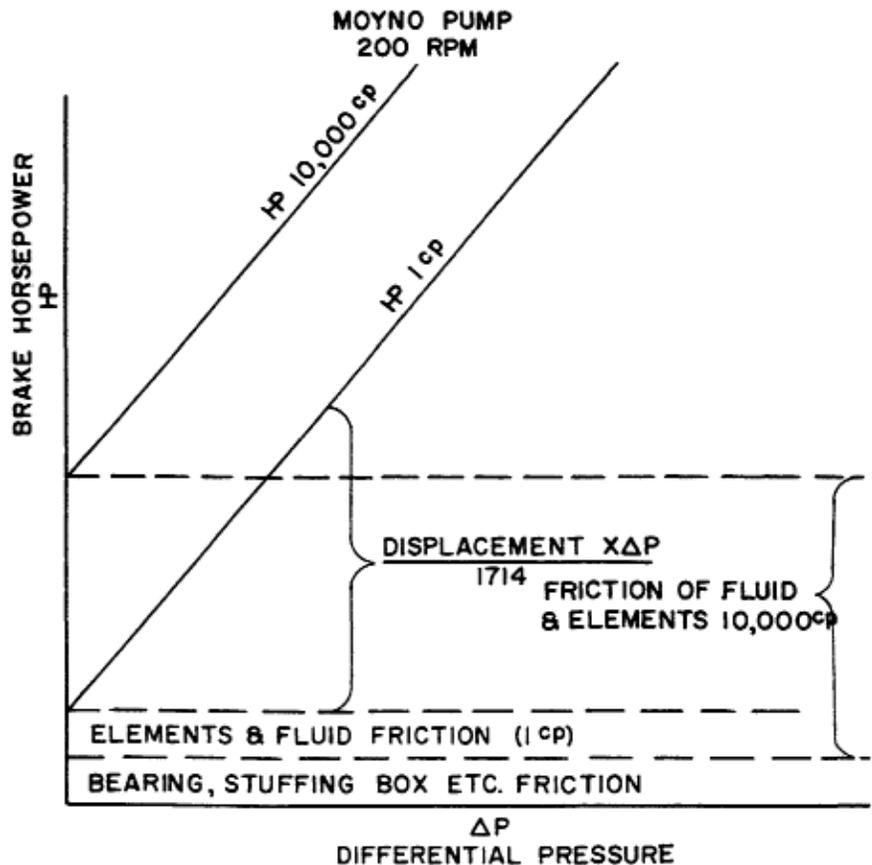
Friction losses were calculated using Poiseuille's formula for laminar flow (since, as mentioned previously, flow was almost entirely laminar or viscous on applications where friction losses were of concern to us); and, an apparent viscosity "somewhat" lower than the viscometer reading at the highest 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. Viscosity data for a great number of fluids plotted against viscometer shear rate on log-log paper revealed relatively straight lines. This indicated 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 Ostwald²;

$$\begin{aligned} \tau &= \text{shear stress} \\ \delta &= \text{shear rate} \\ \mu_N &= \text{nominal viscosity} \\ \tau &= K \delta^n \\ \text{since } \mu_N &= \frac{\tau}{\delta} \\ & \quad \quad \quad n-1 \\ \text{then } \mu_N &= K \delta \\ \text{where } n &= 1 \text{ for Newtonian fluids} \\ & \quad \quad n > 1 \text{ for Dilatent fluids} \\ & \quad \quad n < 1 \text{ for Pseudoplastic fluids} \end{aligned}$$

This led to an attempt (in 1963) to extrapolate our viscometer figures to the average shear rate values in the pump and piping. However, 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, where we

FIGURE 4



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 the application justifies running tests, a cavitation or speed versus capacity test at a given Net Positive Suction Head is run and compared to the previously mentioned Newtonian fluid curves (Fig. 6).

The points of intercept of the non-Newtonian fluid curve with the curve for Newtonian fluids give us apparent viscosities at several different pump speeds. The average shear rate within the Moyno pump elements for these pump speeds can be calculated as

$$\gamma = \frac{NV \sqrt{(1.5\pi D)^2 + Ps^2}}{e}$$

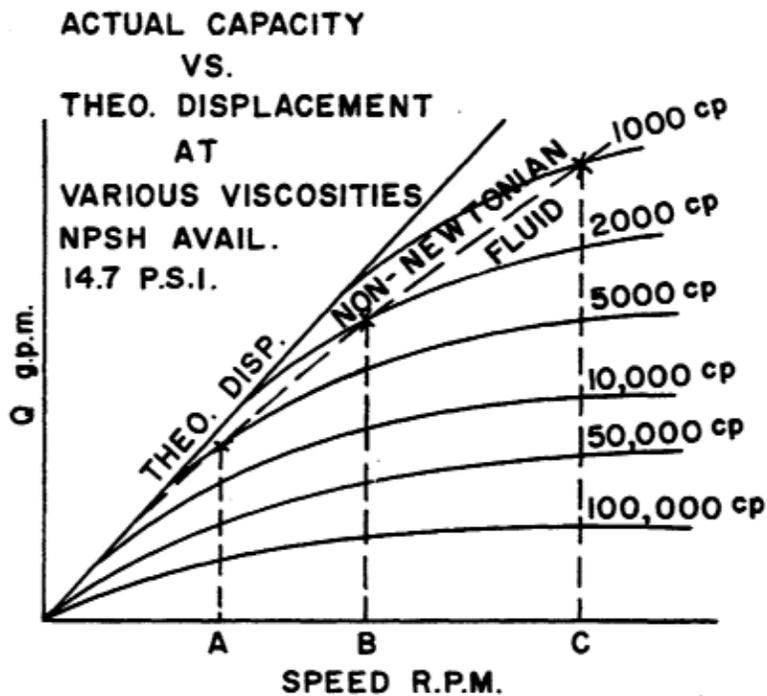
where: N = speed

e, D, Ps are pump constants (see Fig. 1).

This then gives us additional points of apparent viscosity at various shear rates to reinforce or alter our log-log viscometer curve extrapolation (Fig. 7).

If the application warrants (and, again, if sufficient fluid is available), we may further solidify our fluid analysis by checking pressure drop through one pipe size and calculating the apparent viscosity using the Poiseuille formula.

FIGURE 6



In 1965, Penkala and Escarfail³ showed the relevance of rotational viscometer readings to pipe friction losses using the Poiseuille formula in a paper for the Technical Association of the Pulp and Paper Industry. I've plotted their data (rather than some of our own) on our log-log analysis chart; because, not only have they used three different slurries (60% solids, 51% solids, and 45% solids), but they have run their tests through three different pipe sizes (Fig. 8).

As mentioned previously, we

very seldom run tests 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 disk, 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 our own lab, we have added a special high pressure (nitrogen gas) extrusion type viscometer with capillaries up to 1" diameter. The analogies between pipe flow and capillary viscometry are 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 for relatively small samples, is assuring that time-consuming pump tests are unnecessary for proper selection. Our log-log fluid analysis curves for materials ranging from

FIGURE 5

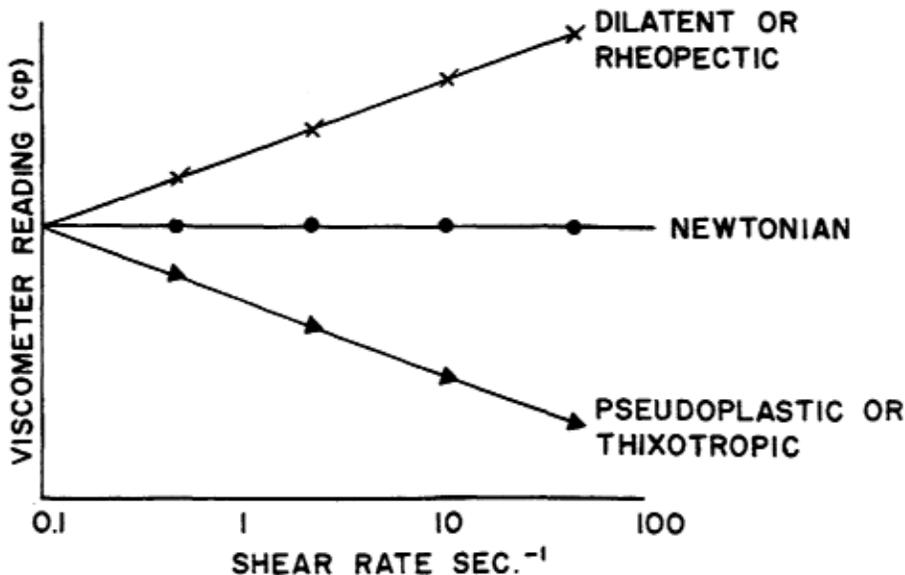
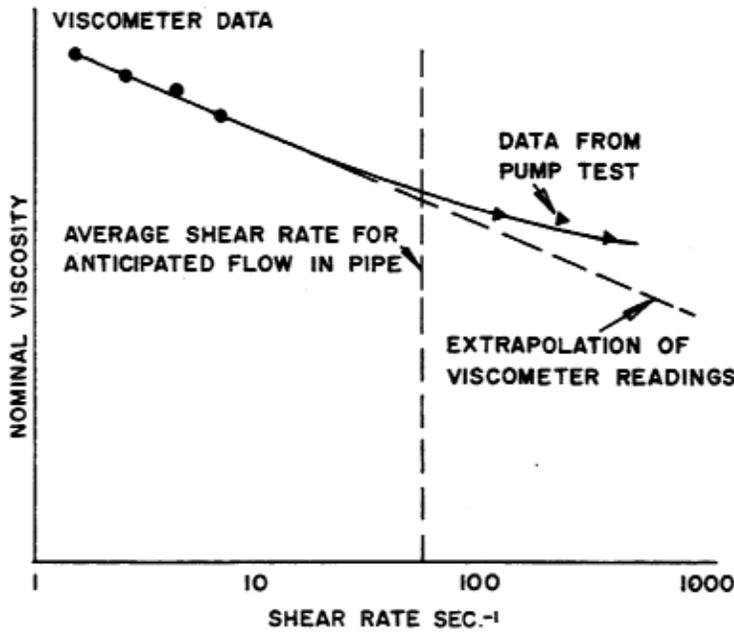


FIGURE 7



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 in pump and piping selection.

However, there are some areas where we lose our security blanket and find ourselves not as capable of standing on sound technical slurry analysis as we are of standing on the slurry itself. One such area involved pumping filter cake from rotary drum vacuum filters, centrifuges, or filter presses to dryers, incinerators, or disposal equipment.

If we were to attempt to categorize it rheologically, most filter cake would fit into a type of Bingham plastic termed "false-body" by Pryce-Jones⁴ in his attempt to distinguish different types of thixotropic behavior. Theoretically, thixotropic materials break down completely under shear stresses that exceed the yield stress and behave like time-dependent pseudoplastic fluids, even after the stress has been removed; whereas, filter cake and "false-bodies" exhibit a continuing but diminished yield strength (Fig. 9).

At this point all practical theory stops. Yet, we have found that we can produce a workable pump system. Most remote laboratory pumping tests tend to be irrele-

vant because the cake is both shear sensitive and time dependent. If collected from the drum filter and shipped to us in 55 gallon drums or other containers, it usually arrives as compacted cake with a layer of water on top, settled in shipment due to vibration, etc.

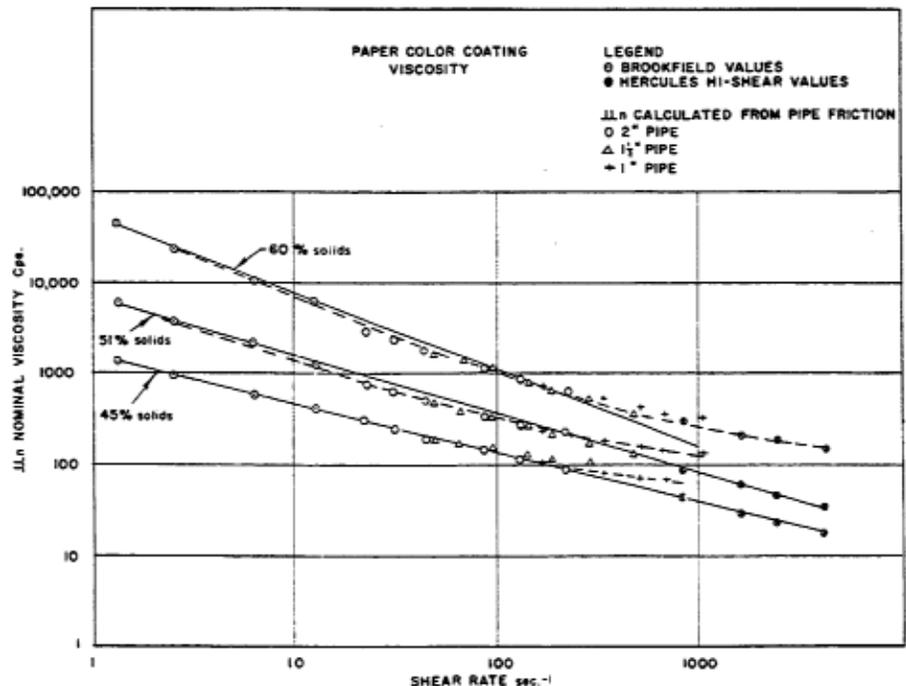
Mixing of the separated components results in another homogeneous mixture but obviously not of the same flow properties

as the original cake. Given enough time and mixing, the slurry can become of such low apparent viscosity (obviously not indicative of its properties as it is discharged from the vacuum filter) that a conventional pump and piping system suffices. Sometime between this point and the time of shoveling the separated solids from the drum, the cake probably passes through its original condition as it's gathered from the drum filter. Exactly when is anybody's guess.

Lab tests then can become quite misleading unless performed on cake freshly off the vacuum filter, transported without an undue amount of vibration, evaporation, and time lapse. By necessity the first of the development work was achieved on the location of a vacuum drum filter at the Richmond, Indiana, Water Pollution Control Center, on municipal waste sludge cake.

Our starting point, in terms of equipment, was the conventional open throat Moyno pump. It was found, several years previously, that many materials which could be pumped, if allowed to travel into the elements where positive pumping action was applied, would bridge or dewater in conventional piping or conical

FIGURE 8



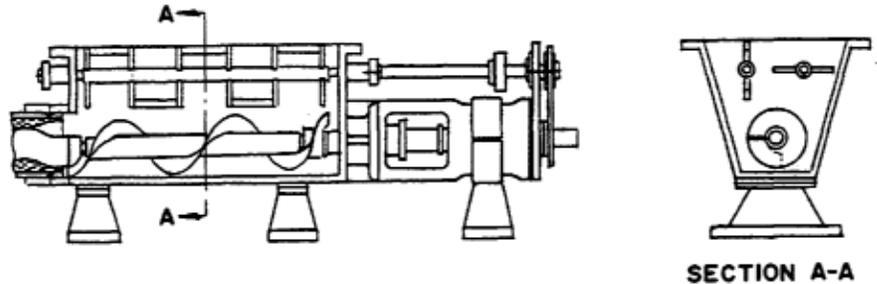
hoppers, thereby never reaching the pump elements. At that time we found that by opening up the suction port of the pump and adding a helical conveyor to our drive train, the range of materials that could be pumped increased drastically.

Matching the displacement of the auger to the displacement of the pump was dimensionally impossible. It would have resulted in too small a conveyor opening to be practical. Therefore, it was necessary to leave the conveyor open to allow easy recirculation within the suction housing to effect agitation and general direction of the slurry flow.

In applying this pump directly to filter cake with no modifications, it was found to be only partially effective. If the pump was underfed (run at a pump displacement well in excess of the filter discharge so that the total auger feed was taken by the pump elements), the system would work. It was found though, that with a build up in the hopper above the auger, the overfeed would turn the relatively dry and brittle cake into a gummy putty-like fluid that would build a shroud over the auger flight, thus cutting off flow into the auger and stopping pump flow.

Various devices were used in an attempt to eliminate this problem. Obviously most systems at one time or another, regardless of an underfeed condition, would have to start with a full hopper or chute. Vibrating the pump was of

FIGURE 10
OPEN THROAT PUMP w/ BRIDGE BREAKERS



no value, it was found to be actually a detriment. Similarly, enclosing a portion of the auger in an attempt to force cake into the elements was a failure. We've found that any attempt to force a material, that tends to dewater or compact, into a positive displacement pump separates the solids from the liquid. The liquid is pumped and the solids remain to foul the suction port. Devices to force the cake into the auger also proved disastrous.

Eventually, it was found that the most effective means was simply a series of fingers rotating (Fig. 10) above the auger to break up any bridge forming that might shut off the flow. Since power requirements for such a device were minimal, a separate drive was not necessary. It was driven by gears, chains and sprockets from the main pump drive shaft.

Hopper or chute inlet design into the pump is very important. It must be such that the cake falls

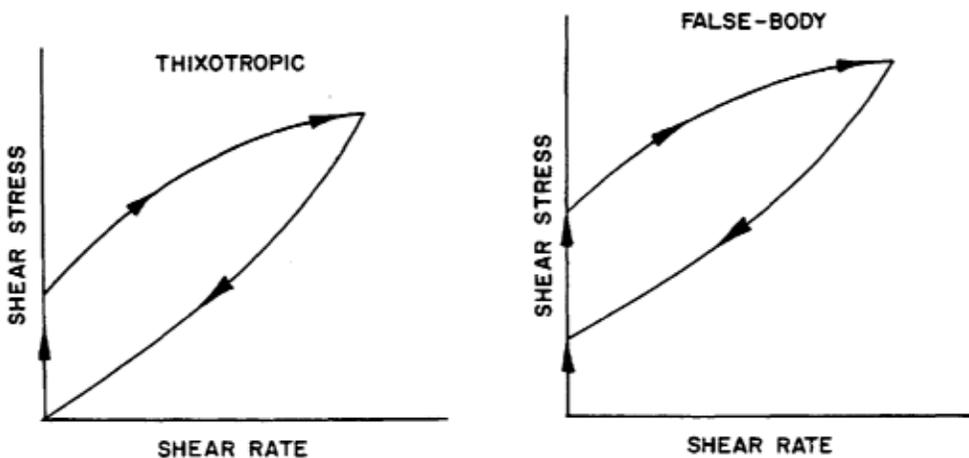
directly onto the breaker-auger assembly. It is recommended that all four sides be at least vertical with possibly even a slight negative slope to the sides. If so designed, wedging of the cake cannot occur above the pump. It has been found that the two shorter sides of the rectangular hopper can be sloped to a certain degree without problems; but, if at all possible, it should be avoided.

Variable speed drives are recommended to match the filter discharge to the pumping rates; although, a constant speed drive capable of handling the maximum discharge with underfeeding at lower rates is satisfactory, if excessive air from the system discharge is not detrimental. Speed usually should not exceed 200 RPM; and, volumetric displacement of the pump, depending on the degree of trapped air and amount of underfeed, will be between 50% and 90%. The lower figure should be used for pump selection. The friction additive for horsepower requirements of the total pump including the bridge breaker assembly on even the most rigid cakes has been found to be less than the values we have published for Newtonian fluids of 20,000 cp.

If one pump is taking the total discharge of the filter, the discharge piping normally should never be less than 6". Wilkinson⁵ notes in his book on non-Newtonian fluids that the relationship for power law fluids in pipe,

$$\Delta P \propto \frac{LQ^n}{d^{3n+1}}$$

FIGURE 9



shows that for a perfect pseudo-plastic fluid ($n = 0$), the flow rate has no effect on the pipe friction losses; and, that the effect of pipe diameter, rather than being inversely proportional to the fourth power, becomes a single power relationship. For a fairly wide variety of cakes, percentages of solids, and flow rates, we've found that in 6" pipe the friction loss varies between 0.25 psi and 2.0 psi per foot of straight pipe. For example, dewatered municipal waste sludge of 20 to 30% solids from rotary drum filters most commonly runs between $\frac{1}{3}$ and $\frac{1}{2}$ psi per foot; while, heat treated sludge at 45% solids content from the same dewatering device runs 1-1½ psi. Discharge of the same percentage of solids from a centrifuge appears to reduce these rotary drum values by 50%, since the sludge enters the pump at a lower apparent viscosity due to its thixotropic nature and the higher rate of shear within the centrifuge. Filter press discharge at approximately 65% solids moves through the 6" pipe from 1½ to 2 psi friction loss per foot of horizontal pipe.

In those few installations where we have had the opportunity to change pipe sizes for the same cake and flow rates, we have found that 4" pipe slightly more than doubles these values and 8" pipe approximately halves them. This data indicates an approximate relationship of d^2 or $n = 0.33$ in the previously stated power law, pipe friction relationship. Doubling the flow rate in these same installations did increase the pressure drop; but, the drop was not far moved from the

$\Delta P \propto \sqrt[3]{Q}$ relationship indicated by pipe size change data. Apparently because of the low pipe velocities (normally less than 0.2

ft/sec), the difference in friction loss between long sweep bends and straight 90° elbows has been found to be negligible in those installations where a change has been made.

In summation then, a workable system can be achieved by: selecting an open throat Moyno pump and bridge breaker with double the necessary displacement at a speed of less than 200 RPM of the desired cake flow rate, specifying an adequate pump driver by adding the horsepower requisite for a 20,000 cp Newtonian fluid to the horsepower requirements for water, specifying that the pump inlet is located in a position where the cake cannot bridge above the breakers (i.e. vertical hopper sides), and assuring that the model selected has pressure capabilities required for the static lift and friction loss from $\frac{1}{4}$ to 2 psi per foot of discharge pipe length for 6" pipe), depending on the dewatering device.

Of course there is no necessity to belabor the advantages of pumping filter cake through a

piping system compared to an open or closed conveyor system, in regard to the installation costs, space required, safety, and cleanliness, where a static lift or relatively long distances are involved. Fortunately, as the data from diverse installations rolls in, we are finding a surprising consistency from one filter cake to another; at least to the degree that, we can empirically select pumps and suggest piping sizes in spite of the complex rheological nature of filter cakes.

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