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SESSION VII

EQUIPMENT

WEDNESDAY, MAY 12, 1982

CHAIRPERSON: E.T. McHALE

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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 collected by analyzing coal/alcohol and coal/water mixtures 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 of 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 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 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.

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 stright 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 can ignore this time-dependence on viscosity and treat the materials as a near-Newtonian liquid. Although using the high viscosity derived from a "first pass" rate of shear will oversize our selected pump and pipe system to a small degree, this oversizing is necessary, as explained later, to counteract the effect of abrasion on pump life.

Probably one of the most convenient formulas for the prediction of non-Newtonian flow is the Power Law originally proposed by Ostwald.[1] 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 curve, one can extrapolate to the nominal shear rate and pick off the apparent 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.

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.

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[2] 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 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 coal slurries in our laboratory using a Brookfield rotating disk 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. A linear extrapolation of Brookfield data is relevant to predicting actual viscosity up to approximately 1,000 inverse seconds, in an much as an accuracy range of 200 cps can be attained. Since we are dealing with rotary positive displacement pumps, this accuracy is more than adequate. In fact, a confidence band of several hundred centipoise would be adequate for most pump selection calculations.

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 the maximum velocity at the center of the pipe is three times the average. This dilatent or rheopectic phenomena ("n" greater than 1) does not normally exist in slurries. On the other hand, the perfect pseudoplast, 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 pseudoplast 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