Fluid Viscosity Effects on Centrifugal Pumps

BY: GUNNAR HOLE

When sizing a pump for a new application or evaluating the performance of an existing pump, it is often necessary to account for the effect of the pumped fluid's viscosity. We are all aware that the head-capacity curves presented in pump vendor catalogs are prepared using water as the pumped fluid. These curves are adequate for use when the actual fluid that we are interested in pumping has a viscosity that is less than or equal to that of water. However, in some cases—certain crude oils, for example—this is not the case.

Heavy crude oils can have viscosities high enough to increase the friction drag on a pump's impellers significantly. The additional horsepower required to overcome this drag reduces the pump's efficiency. There are several analytical and empirical approaches available to estimate the magnitude of this effect. Some of these are discussed below.

Before beginning the discussion, however, it is vital to emphasize the importance of having an accurate viscosity number on which to base our estimates. The viscosity of most liquids is strongly influenced by temperature. This relationship is most often shown by plotting two points on a semi-logarithmic grid and connecting them with a straight line. The relationship is of the form:

$$\mu = Ae^{B/T}$$

where

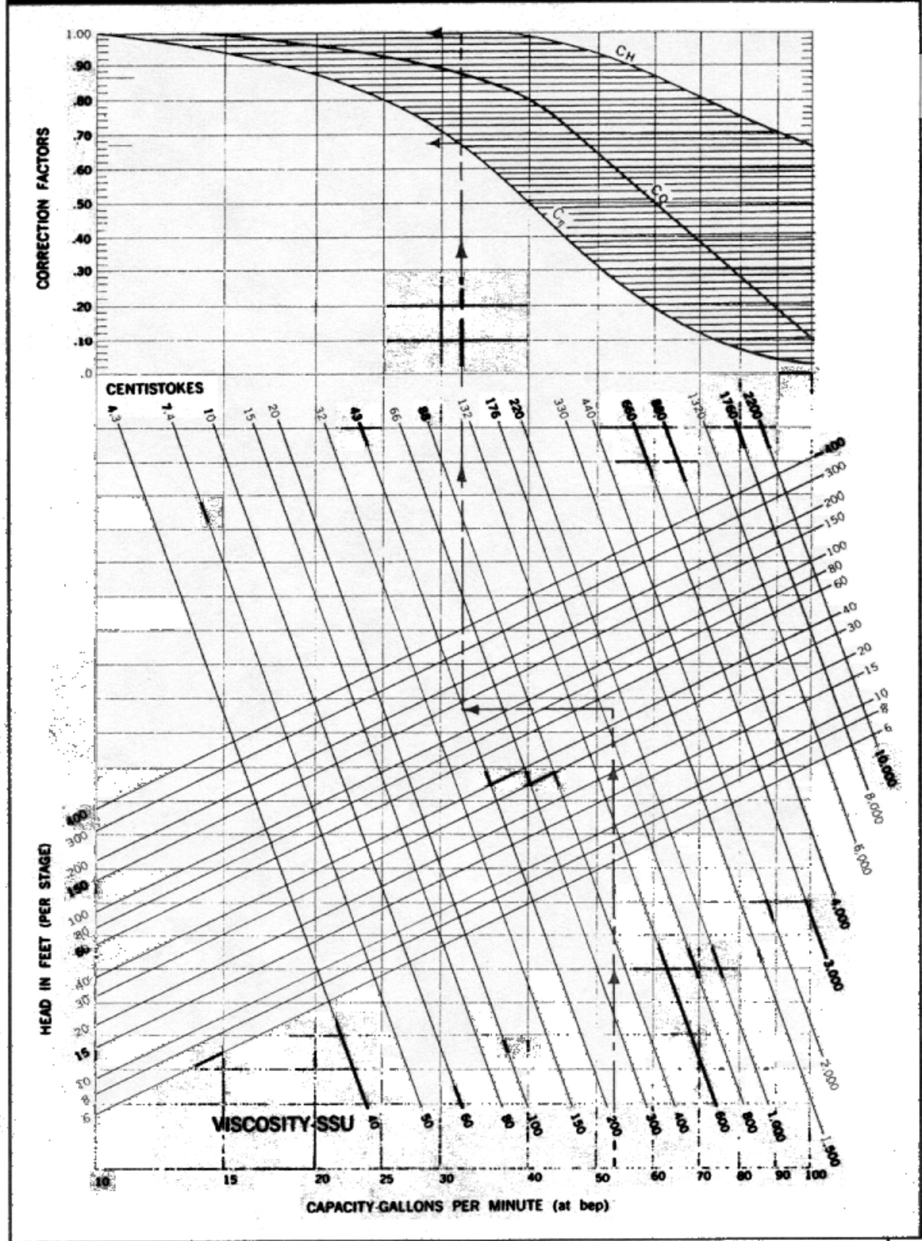
μ = the absolute viscosity of the fluid

A and B = constants

T = the absolute temperature of the fluid

Plotting this relationship requires knowledge of two data points, and using them effectively requires some judgement as to the normal operating temperature as well as the minimum temperature that might be expected during other off-design conditions such as start-up.

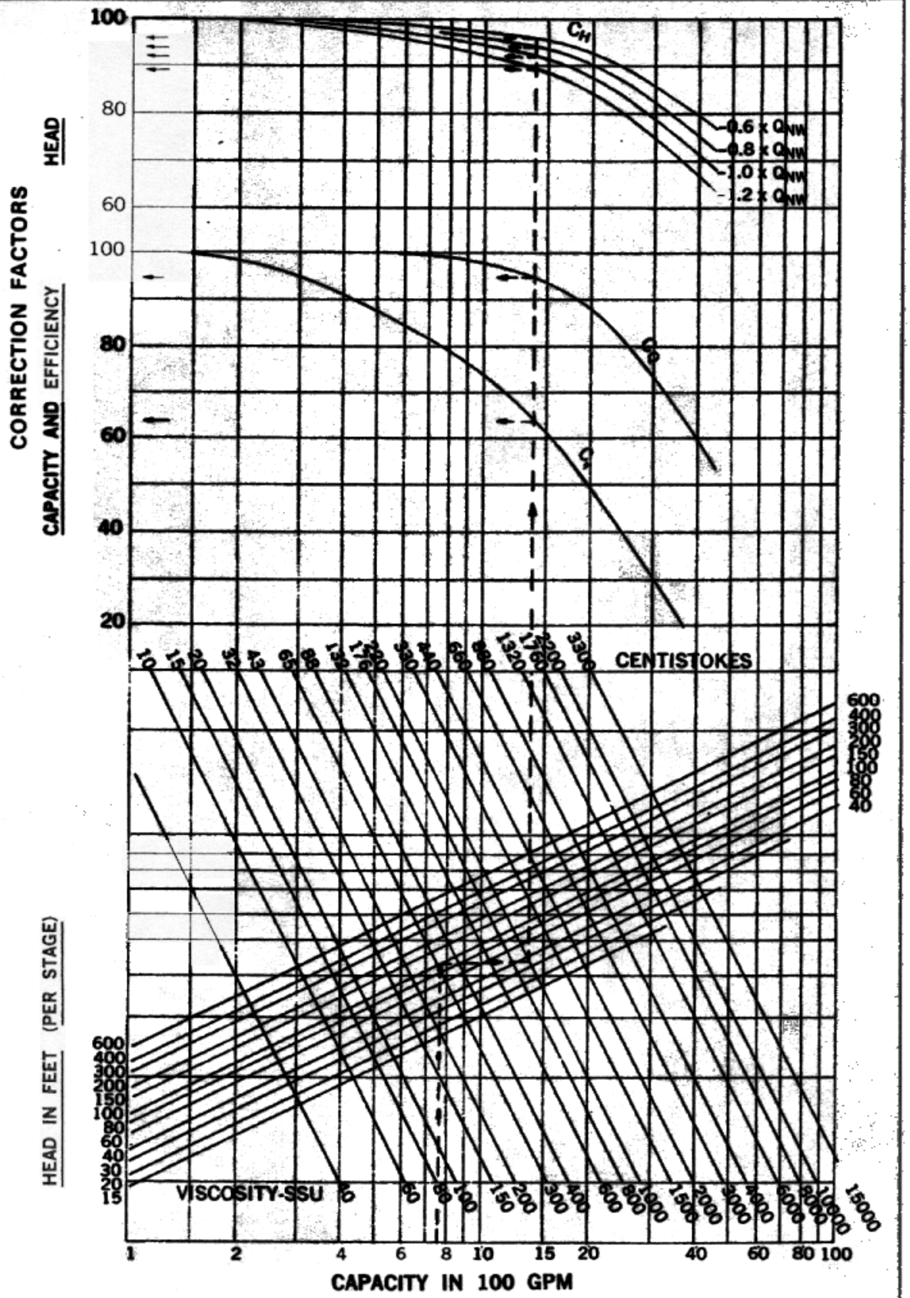
FIGURE 1



Reproduced from the Hydraulic Institute Standards (Figure 71)

The effect of pressure on the viscosity of most fluids is small. For mineral oils, for example, an increase of pressure of 33 bars (≈480 psi) is equivalent to a tem-

FIGURE 2



Reproduced from the Hydraulic Institute Standards (Figure 72)

perature drop of 1°C.

The following definitions are used when discussing fluids and viscosity. There are five basic types of liquid that can be differentiated on the basis of their viscous behavior; they are:

NEWTONIAN

These are fluids where viscosity is constant and independent of shear rate, and where the shear rate is linearly proportional to the shear stress. Examples are water and oil.

NON-NEWTONIAN

These are fluids where the shear rate-shear stress relationship is nonlinear. They can be divided into four categories:

- Bingham-plastic fluids are those in which there is no flow until a threshold shear stress is reached. Beyond this point, viscosity decreases with increasing shear rate. Most slurries have this property, as does America's favorite vegetable, catsup.
- Dilatant fluids are those of which viscosity increases with increasing shear rate. Examples are candy mixtures, clay slurries, and quicksand.
- Pseudo-plastic fluids are similar to Bingham-plastic fluids, except there is no definite yield stress. Many emulsions fall into this category.
- Thixotropic fluids are those of which viscosity decreases to a minimum level as their shear rate increases. Their viscosity at any particular shear rate may vary, depending on the previous condition of the fluid. Examples are asphalt, paint, molasses, and drilling mud.

There are two other terms with which you should be familiar:

- Dynamic or absolute viscosity is usually measured in terms of centipoise and has the units of force time/length^2.
- Kinematic viscosity is usually measured in terms of centistokes or ssu (Saybolt Seconds Universal). It is related to absolute viscosity as follows:

kinematic viscosity =
absolute viscosity/mass density

The normal practice is for this term to have the units of length^2/time. Note:

$1\ cSt = cP \times sp\ gr$

$1\ cSt = 0.22 \times ssu - (180/ssu)$

$1\ cP = 1.45E^{-7}\ lbf - s/in^2$

$1\ Reyn = 1\ lbf - s/in^2$

TABLE 1. WATER-BASED AND VISCOUS PERFORMANCE				
Water Curve-Based Performance	% of BEP Capacity			
	60%	80%	100%	120%
Capacity, gpm	450	600	750	900
Differential Head, ft.	120	115	100	100
Efficiency	0.70	0.75	0.81	0.75
Horsepower	18	21	21	27
Viscous (1,000 ssu) Performance				
Capacity, gpm	423	564	705	846
Differential Head, ft.	115	108	92	89
Efficiency	0.45	0.48	0.52	0.48
Horsepower	25	29	28	36

Note: Pumped fluid specific gravity = 0.9

The process of determining the effect of a fluid's viscosity on an operating pump has been studied for a number of years. In the book *Centrifugal and Axial Flow Pumps*, A.J. Stepanoff lists the losses that affect the performance of pumps as being of the following types:

- mechanical losses
- impeller losses
- leakage losses
- disk friction losses

Of all external mechanical losses, disk friction is by far the most important, according to Stepanoff. This is particularly true for pumps designed with low specific speeds. Stepanoff gives a brief discussion of the physics of a rotating impeller and emerges with a simple equation that summarizes the drag force acting upon it:

$$(hp)_d = Kn^3D^5$$

where

K = a real constant

n = the pump operating speed

D = the impeller diameter

The explanation further describes the motion of fluid in the immediate neighborhood of the spinning impeller. There Stepanoff mentions the experimental results of others demonstrating that, by reducing the clearance be-tween the stationary casing and the impeller, the required power can be reduced. He also writes about the details of some investigations that demonstrate the beneficial effect of good surface finishes on both the stationary and rotating surfaces. Included is a chart prepared by Pfleiderer, based on work by Zumbusch and Schultz-Grunow, that gives friction coefficients for calculating disk friction losses. The chart is used in conjunction with the following equation:

$$(hp)_d = KD^2\gamma u^3$$

where

K = a constant based on the Reynolds number

D = impeller diameter

γ = fluid density

u = impeller tip speed

Like most of Stepanoff's writing, this presentation contains great depth with considerable rigor. It makes interesting reading if you are willing to put forth the time. Those of us who need a quick answer to a particular problem may need to look elsewhere for help.

In the book, *Centrifugal Pumps*, V. Lobanoff and R. Ross discuss the effect of viscous fluids on the performance of centrifugal pumps. They make the point that because the internal flow passages in small pumps are proportionally larger than those in larger pumps, the smaller pumps will always be more sensitive to the effects of viscous fluids. They also introduce a diagram from the paper "Engineering and System Design Considerations for Pump Systems and Viscous Service," by C.E. Petersen, presented at Pacific Energy Association, October 15, 1982. In this diagram, it is recommended that the maximum fluid viscosity a pump should be allowed to handle be limited by the pump's discharge nozzle size. The relationship is approximately:

$$\text{viscosity}_{max} = 300(D_{\text{outlet nozzle}} - 1)$$

where

viscosity is given in terms of ssu

D is measured in inches

With respect to the prediction of the effects of viscous liquids on the performance of centrifugal pumps, Lobanoff and Ross direct the reader to the clearly defined methodology of the *Hydraulic Institute Standards*. This technique is based on the use of two nomograms on pages 112 and 113 of the 14th edition (Figures 71 and 72). They are reproduced here as Figures 1 and 2. Intended for use

TABLE 2. POLYNOMIAL COEFFICIENTS						
Correction Factor						
	D_{x1}	D_{x2}	D_{x3}	D_{x4}	D_{x5}	D_{x6}
C_η	1.0522	-3.5120E-02	-9.0394E-04	2.2218E-04	-1.1986E-05	1.9895E-07
C_Q	0.9873	9.0190E-03	-1.6233E-03	7.7233E-05	-2.0528E-06	2.1009E-08
$C_{H0.6}$	1.0103	-4.6061E-03	2.4091E-04	-1.6912E-05	3.2459E-07	-1.6611E-09
$C_{H0.8}$	1.0167	-8.3641E-03	5.1288E-04	-2.9941E-05	6.1644E-07	-4.0487E-09
$C_{H1.0}$	1.0045	-2.6640E-03	-6.8292E-04	4.9706E-05	-1.6522E-06	1.9172E-08
$C_{H1.2}$	1.0175	-7.8654E-03	-5.6018E-04	5.4967E-05	-1.9035E-06	2.1615E-08

on pumps with BEPs below and above 100 gpm, respectively, which permits the user to estimate the reduction of head, capacity, and efficiency that a viscous fluid will produce on a pump curve originally generated with water. A variation on this technique is described below.

The following example is taken from pages 114-116 of the *Hydraulic Institute Standards* section on centrifugal pump applications. There, the use of Figure 72, "Performance Correction Chart For Viscous Liquids," is discussed. Table 1 was calculated using polynomial equations developed to replace the nomogram presented in Figure 72. The results of the calculation are within rounding error of those presented in the standard. And the approach has the additional benefit of being more convenient to use, once it has been set up as a spreadsheet.

In the course of curve-fitting Figure 72, it was convenient to define a term known as pseudocapacity:

$$\text{pseudocapacity} = 1.95(V)^{0.5}[0.04739(H)^{0.25746}(Q)^{0.5}]^{-0.5}$$

where

V = fluid viscosity in centistokes

H = head rise per stage at BEP, measured in feet

Q = capacity at BEP in gpm

Pseudocapacity is used with the following polynomial coefficients to determine viscosity correction terms that are very close to those given by Figure 72 in the *Hydraulic Institute Standards*. These polynomials have been checked throughout the entire range of Figure 72, and appear to give answers within 1.0% of those found using the figure.

The polynomial used is of the form:

$$C_x = D_{x1} + D_{x2}\,P + D_{x3}\,P^2 + D_{x4}\,P^3 + D_{x5}\,P^4 + D_{x6}\,P^5$$

where

C_x is the correction factor that must be applied to the term in question

D_{xn} are the polynomial coefficients listed in Table 2

P is the pseudocapacity term defined above

For comparison, the correction factors for the example above (tabulated in Table 7 of the *Hydraulic Institute Standards*) and those calculated using the polynomial expressions above are listed in Table 3.

TABLE 3. CORRECTION FACTOR COMPARISON

	C_η	C_Q	$C_{H0.6}$	$C_{H0.8}$	$C_{H1.0}$	$C_{H1.2}$
Per Table 7 of HI Standards	0.635	0.95	0.96	0.94	0.92	0.89
Per Polynomial Expressions	0.639	0.939	0.958	0.939	0.916	0.887

The problem of selecting a pump for use in a viscous service is relatively simple once the correction coefficients have been calculated. If, for example, we had been looking for a pump that could deliver 100 feet of head at a capacity of 750 gpm, we would proceed as follows:

$$H_{water} = H_{viscous\ service}/C_{H1.0}$$

$$Q_{water} = Q_{viscous\ service}/C_Q$$

The next step would be to find a pump having the required performance on water. After determining the efficiency of the pump on water, we would correct it for the viscous case as shown above:

$$\eta_{viscous\ service} = \eta_{water} \times C_\eta$$

The horsepower required by the pump at this point would be calculated as follows:

$$hp_{viscous\ service} = \frac{(Q_{viscous\ service} \times H_{viscous\ service} \times sp\ gr)}{(3{,}960 \times \eta_{viscous\ service})}$$

As with water service, the horsepower requirements at off-design conditions should always be checked. ■

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