THE COMPONENTS OF TOTAL HEAD

3

This chapter will introduce some of the terminology used in pumping systems. The components of Total Head will be examined one by one. Some of the more difficult to determine components, such as equipment and friction head, will be examined in more detail. I hope this will help get our heads together.

3.0 THE COMPONENTS OF TOTAL HEAD

Total Head is the measure of a pump's ability to push fluids through a system. Total Head is proportional to the difference in pressure at the discharge vs. the suction of the pump. It is more useful to use the difference in pressure vs. the discharge pressure as a principal characteristic since this makes it independent of the pressure level at the pump suction and therefore independent of a particular system configuration. For this reason, the Total Head is used as the Y-axis coordinate on all pump performance curves (see Figure 4-3).

The system equation for a typical single inlet — single outlet system (see equation [2-12]) is:

$$\Delta H_P = \Delta H_{F1-2} + \Delta H_{EQ\,1-2} + \frac{1}{2g}(v_2^{\,2} - v_1^{\,2}) + z_2 + H_2 - (z_1 + H_1) \qquad [3\text{-}1]$$

$$\Delta H_P = \Delta H_F + \Delta H_{EQ} + \Delta H_v + \Delta H_{TS} \qquad [3\text{-}1a]$$

$$\Delta H_P = \Delta H_F + \Delta H_{EQ} + \Delta H_v + \Delta H_{DS} + \Delta H_{SS} \qquad [3\text{-}1b]$$

Equations [3-1a] and [3-1b] represent different ways of writing equation [3-1], using terms that are common in the pump industry. This chapter will explain each one of these terms in details.

3.1 TOTAL STATIC HEAD (ΔH_{TS})

The total static head is the difference between the discharge static head and the suction static head, or the difference in elevation at the outlet including the pressure head at the outlet, and the elevation at the inlet including the pressure head at the inlet, as described in equation [3-2a].

$$\Delta H_{TS} = \Delta H_{DS} - \Delta H_{SS} \qquad [3\text{-}2]$$

$$= z_2 + H_2 - (z_1 + H_1) \qquad [3\text{-}2a]$$

H_2 and H_1 are the pressure heads at points 2 and 1 respectively. Some people include these pressure heads with the elevation head, others do not. I will be using the former approach, but either way the pressure heads have to be considered. The discharge static head (ΔH_{DS}) is normally positive (assuming $H_2 = 0$), since fluid is usually pumped to a higher elevation. However, it may on occasion be negative (for example, lower discharge pipe end than pump centerline) and the complete system must be evaluated before it is known if a pump is required or not.

The suction static head (ΔH_{SS}) can either be negative or positive, depending on whether the pump centerline is below the suction fluid surface or above, and the value of suction tank fluid surface pressure head (H_1).

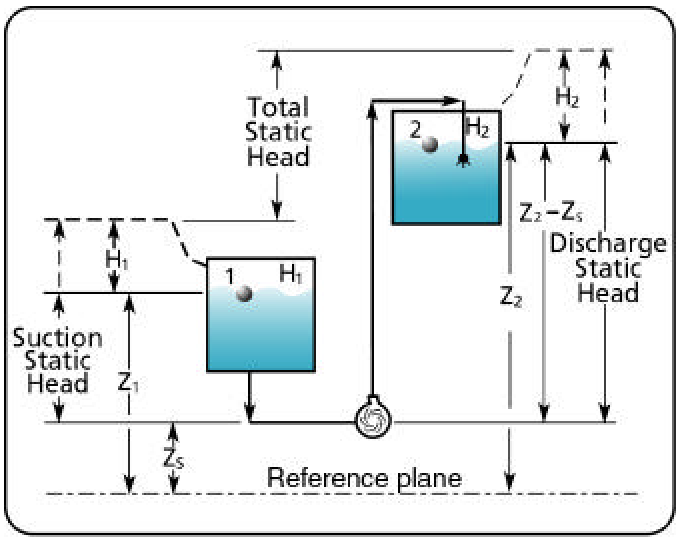

Figure 3-1 Relationship between the various heads in a pumping system.

3.2 SUCTION STATIC HEAD (ΔH_{SS})

The Suction Static head is the sum of the elevation and pressure head at the inlet of the system, minus the elevation of the pump center line, as stated in equation [3-3]. The inlet of the system is located at point 1, which is the surface of the suction tank fluid. H_1 is the pressure head at the suction tank fluid surface. If the tank is open to atmosphere then H_1 = 0.

Typical pumping configurations are shown in Figure 3-2, the pump suction is under positive pressure in A, and possibly under negative relative pressure in B.

The Suction Static head in these two cases is:

$$\Delta H_{SS} = z_1 + H_1 - z_s \qquad [3\text{-}3]$$

Figure 3-2B presents a situation where the pump has to lift the fluid up to the pump suction. The head at the suction is described as suction lift. This head is normally negative with respect to atmospheric pressure since the term $z_1 - z_S$ is negative (assuming $H_1 = 0$ or the same as the atmospheric pressure).

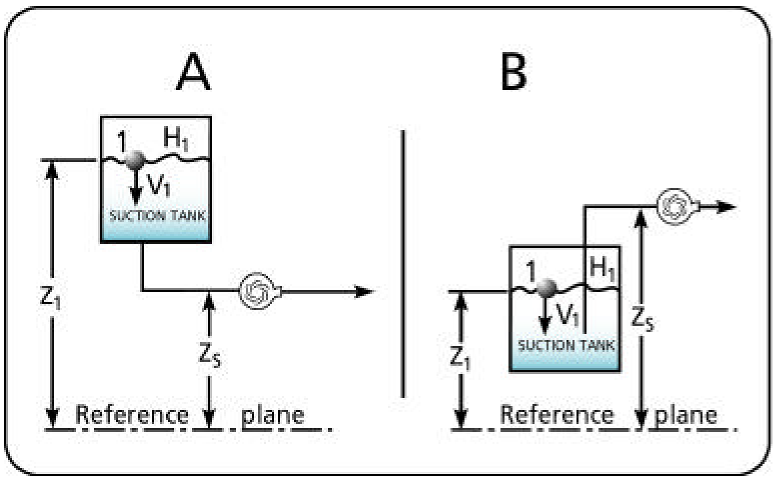

Figure 3-2 Suction static head and suction static lift.

3.3 NET POSITIVE SUCTION HEAD AVAILABLE (N.P.S.H.A.)

The Net Positive Suction Head Available (N.P.S.H.A.) is the total energy per unit weight, or head, at the suction flange of the pump less the vapor pressure head of the fluid. This is the accepted definition which is published by the Hydraulic Institute's Standards books (see the HI web site at www. pumps.org). The Hydraulic Institute is the organization that formulates and promotes the use of common standards used throughout the pump industry in the United States. The term "Net" refers to the actual head at the pump suction flange, since some energy is lost in friction prior to the suction.

Why do we need to calculate the N.P.S.H.A.? This value is required to avoid cavitation of the fluid. Cavitation will be avoided if the head at the suction is higher than the vapor pressure head of the fluid. In addition, the pump manufacturers require a minimum N.P.S.H. to guarantee proper operation of the pump, they call this the N.P.S.H.R., where “R” stands for required.

To determine N.P.S.H.A., first we calculate the pressure head H_S at point S. A control volume is positioned (see Figure 3-3) to intersect the suction inlet of the pump and the fluid surface of the suction tank. The pressure head at any point on the suction side of the pump is given by equation [2-15] where the subscript X is replaced by S:

$$H_S(\,ft\,of\,fluid\,) = -\,(\Delta H_{F1-S} + \Delta H_{EQ1-S}) + \frac{(v_1^{\,2} - v_S^{\,2})}{2g} + (z_1 - z_S + H_1) \qquad [3\text{-}4]$$

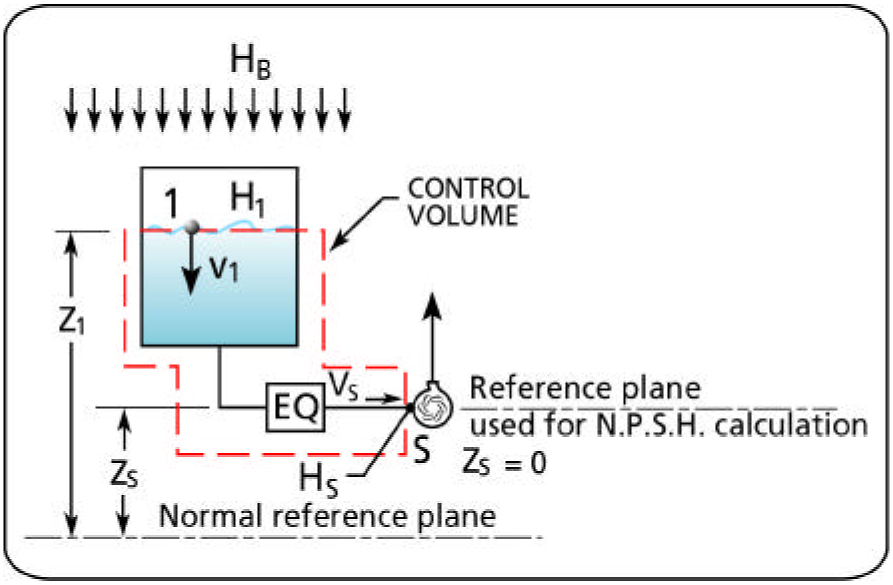

Figure 3-3 Using the control volume for calculating the pressure head at point S.

The specific energy or head $\bar{E}$ for any point in the system is the sum of the elevation (potential) energy, the velocity (kinetic) energy and the pressure energy. $\bar{E}$ is given by:

$$\bar{E} = H + \frac{v^2}{2g} + z \qquad [3\text{-}5]$$

By definition, the N.P.S.H available at the pump suction (point S) is based on a reference plane located at the pump suction centerline (*z = 0*). We can understand why since using any other reference will increase or decrease the energy level at point S which is obviously incorrect (see Figure 3-3). Therefore equation [3-5] becomes:

$$\bar{E}_S = H_S + \frac{v_S^{\,2}}{2g} \qquad [3\text{-}6]$$

The head $\overline{E}_S$ is given in equation [3-6], the barometric head (H_B) is added to H_S to convert $\overline{E}_S$ from feet of fluid to feet of fluid absolute. Therefore equation [3-6] becomes:

$$\overline{E}_S(\, ft\ of\ \ fluid\ \ absol.) = H_S + \frac{v_S^{\,2}}{2g} + H_B \qquad [3\text{-}7]$$

The value of H_S in equation [3-4] is substituted in equation [3-7] to give:

$$\boxed{\overline{E}_S(\, ft\ of\ fluid\ absol\,) = -(\Delta H_{F1-S} + \Delta H_{EQ1-S}) + \frac{v_1^{\,2}}{2g} + (z_1 - z_S + H_1) + H_B} \qquad [3\text{-}8]$$

BOILING LIQUIDS

Different liquids boil at different temperatures for a fixed pressure; also, different liquids boil at different pressures at a fixed temperature. The temperature required to vaporize a liquid varies as the pressure in the surrounding environment. For example, water boils at a temperature of 212 ^{0}F at a surrounding pressure of 14.7 psia (the air pressure at sea level). However, a temperature of 189 ^{0}F is required to boil water at a pressure of 11 psia which is the atmospheric pressure at 8,500 feet of elevation above sea level (or the altitude of Mexico city).

A short digression is in order. Since water boils at a lower temperature in, say, Mexico City than a city which is close to sea level, does this mean that it takes a longer time to boil an egg in Mexico city? Yes, it will take longer in Mexico City. Why? Because the same amount of heat transfer is required to get the egg to the right consistency regardless of the water temperature. It will take longer to transfer the amount of heat required to cook the egg if the water is boiling at a lower temperature. For most of us water boils at the high temperature of 212 ^{0}F (100 ^{0}C), it is very surprising to find that it takes 4 minutes to boil a "3-minute" egg in Mexico City.

The pressure at which a liquid boils is called the **vapor pressure** and is always associated with a specific temperature. When pressure decreases in the fluid's environment, the boiling temperature drops. Many liquids (i.e. acetone, methyl alcohol, benzene, etc.) have a lower vapor pressure than water at the same temperature. Since the pressure throughout a system can vary drastically, it is important to consider the vapor pressure of the liquid in order to avoid vaporization. Data on vapor pressure vs. temperature for many liquids is readily available (see reference 1, 2 and 8).

VAPOR PRESSURE AND CAVITATION

The pressure near the impeller eye is lower than the pressure at the pump suction flange, and depending on the kind of fluid and temperature, may be low enough for vaporization to occur. When this happens, both, vapor and liquid, will enter the pump, and the capacity of the pump will be reduced. The point of lowest pressure is near the eye of the impeller on the underside of the vane (see Figure 3-6), where bubbles can form. Only small bubbles are formed because the fluid is rapidly compressed as it travels from the start of the impeller vane to its tip.

The rapid compression of bubbles causes small pieces of metal to be dislodged from the surface. These bubbles collapse rapidly in the high pressure near the tip of the vane causing noise and vibration. This rapid collapse of vapor bubbles is known as cavitation and is accompanied by a distinct gravely sound similar to the sound made by a cement mixer. *The system should be designed in such a way as to provide sufficient N.P.S.H. available to avoid cavitation under normal running conditions.*

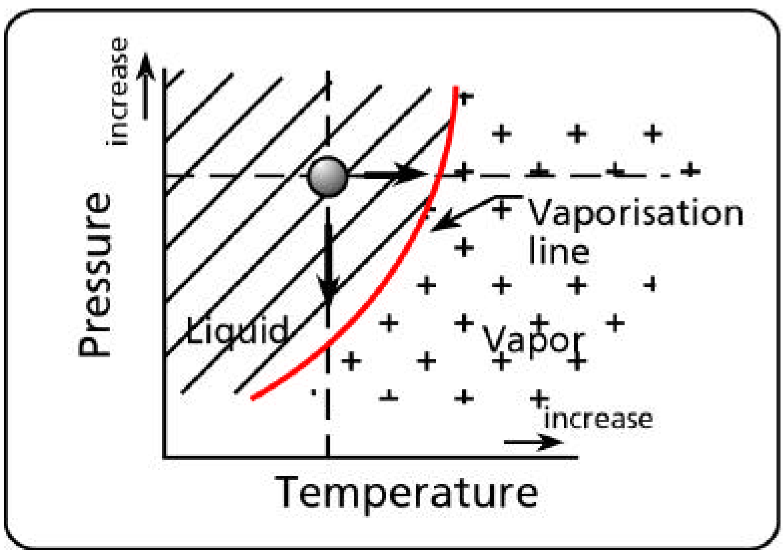

Figure 3-4 Vapor pressure vs. temperature.

In order for the liquid to stay in a fluid state and not vaporize, the head at the inlet of the pump must be above the vapor pressure head of the fluid:

$$\overline{E}_S \geq H_{va}$$

where H_{va} is the vapor pressure head of the liquid. The Net Positive Suction Head Available (N.P.S.H.A.) is the difference between the head ($\overline{E}_S$) at the pump suction and the vapor pressure head (H_{va}).

$$N.P.S.H._{avail.} = \overline{E}_S - H_{va} \qquad [3\text{-}9]$$

By substituting the value of $\overline{E}_S$ from equation [3-8] into equation [3-9] then:

$$\boxed{N.P.S.H._{avail.}(ft\ of\ fluid\ absol.) = -(\Delta H_{F1-S} + \Delta H_{EQ1-S}) + \frac{v_1^2}{2g} + (z_1 - z_S + H_1) + H_B - H_{va}} \qquad [3\text{-}10]$$

where H_B and H_{va} are in feet of fluid.

Vapor and barometric pressures are often given in pounds per square inch absolute (psia). The conversion to feet of fluid absolute is:

$$H(ft\ of\ \ fluid) = \frac{2.31}{SG} \times p(psia)$$

by substitution into equation [3-10]:

$$\boxed{N.P.S.H._{avail.}(ft\ of\ fluid\ absol.) = -(\Delta H_{F1-S} + \Delta H_{EQ1-S}) + \frac{v_1^2}{2g} + (z_1 - z_S + H_1) + H_B - H_{va} + \frac{2.31}{SG}(p_B(psia) - p_{va}(psia))} \qquad [3\text{-}11]$$

The N.P.S.H. in equation [3-10] and [3-11] is in feet of fluid absolute and is a head term, which is independent of fluid density. Since the pump manufacturers use water as the fluid, the N.P.S.H. value they provide is in feet of water absolute. The pump requires a minimum suction pressure head in order to function properly and avoid cavitation. This is known as the N.P.S.H. required, which the pump manufacturer gives for a specific pump model, impeller diameter, speed and flow rate. In order to satisfy the pump manufacturer's requirements:

$$N.P.S.H._{avail.} \geq N.P.S.H._{req.}$$

Figure 3-5 shows typical relative proportions of the terms in equation [3-10].

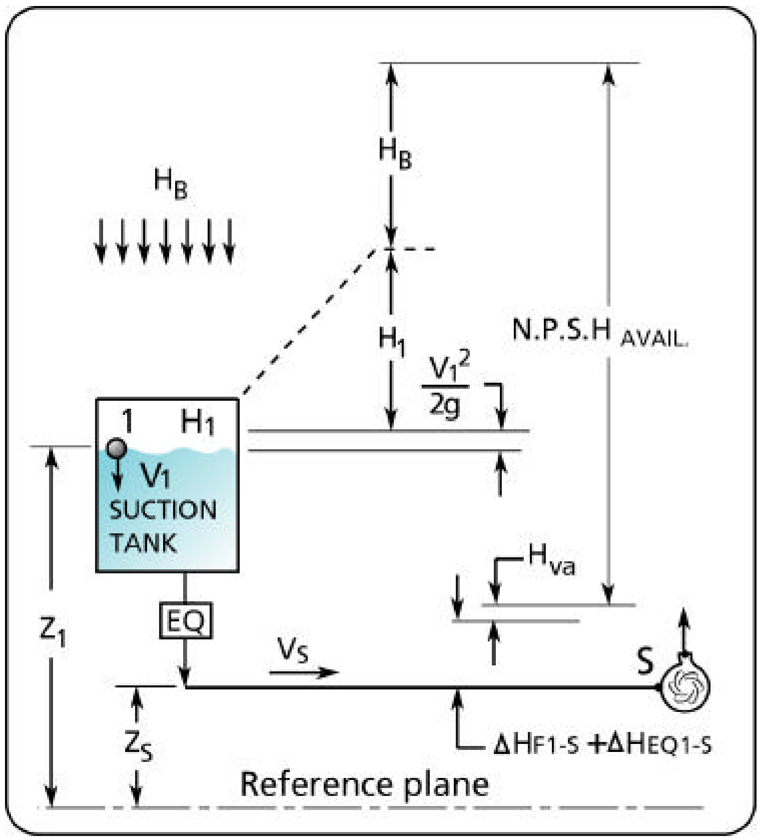

Figure 3-5 Relative size of N.P.S.H. components.

N.P.S.H. REQUIRED

The N.P.S.H. required provides us with the level of head in terms of feet of water absolute required at the pump suction flange. When that level of head is insufficient the capacity and head of the pump will drop and cavitation will occur.

Figure 3-6 shows how the pressure varies between the pump suction flange and the discharge flange.

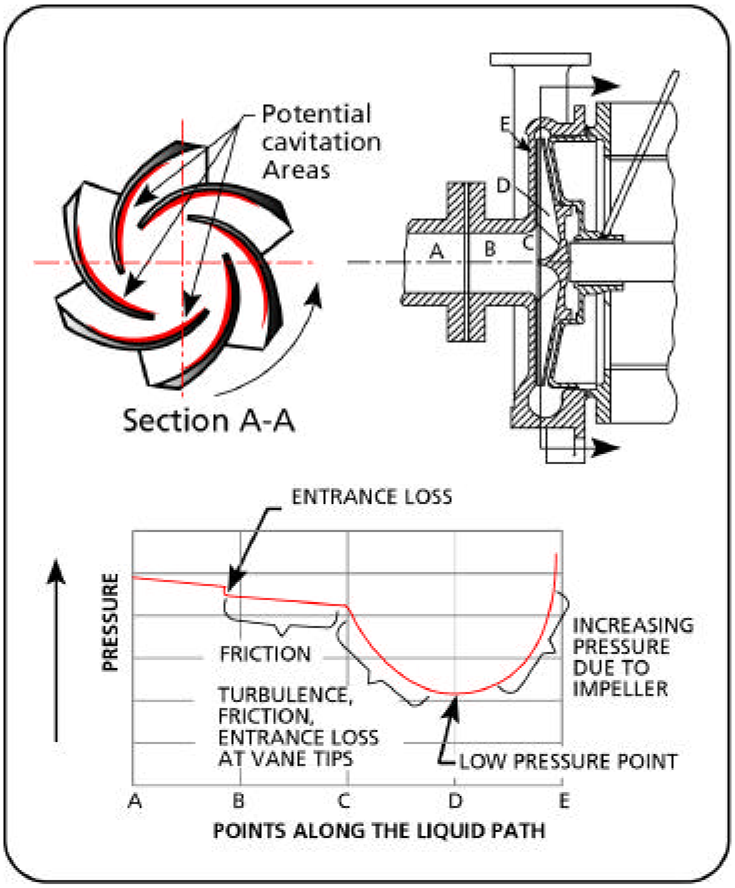

Figure 3-6 The pressure variation within a pump (source: the Durco company).

Let's follow the path of the fluid particles form point A to E in Figure 3-6 to see where and why the pressure is varying. There is a small friction loss in the short piece of piping of the pump suction. Both, friction loss and turbulence, accompany the right angle turn that the liquid makes in going from horizontal flow to outward radial flow in the impeller. The leading edges of the vanes act as obstruction in the path of the liquid, and entrance loss occurs at each vane edge. Any pre-rotation of the liquid as it enters the impeller, changes the inlet angle and results in more turbulence. All of these losses occur before the liquid is acted upon positively by the impeller vane. The combination of these losses, plus the losses that occurred in the suction line, can be great enough to lower the pressure of the fluid sufficiently to reach the vapor pressure, at which point the fluid will vaporize. This condition must be avoided. Once the liquid is in the impeller, with the vanes pushing from behind, the pressure starts to increase and eventually reaches the full discharge pressure head.

HOW DO THE PUMP MANUFACTURERS MEASURE N.P.S.H. REQUIRED?

The pump manufacturers measure the N.P.S.H. required in a test rig similar to that shown in Figure 3-7. The system is run in a closed loop where flow, total head and power consumed is measured. In order to provide a low N.P.S.H., a vacuum pump is used to lower the pressure in the suction tank which will provide a low head at the pump suction. The pressure in the suction tank is lowered until a drop of 3% of the total head is measured (see Figure 3-8). When that occurs the N.P.S.H. is calculated and recorded as the N.P.S.H. required for that operating point. Heating coils are also used which increase the water temperature thereby increasing the vapor pressure and further lowering the N.P.S.H. as needed.

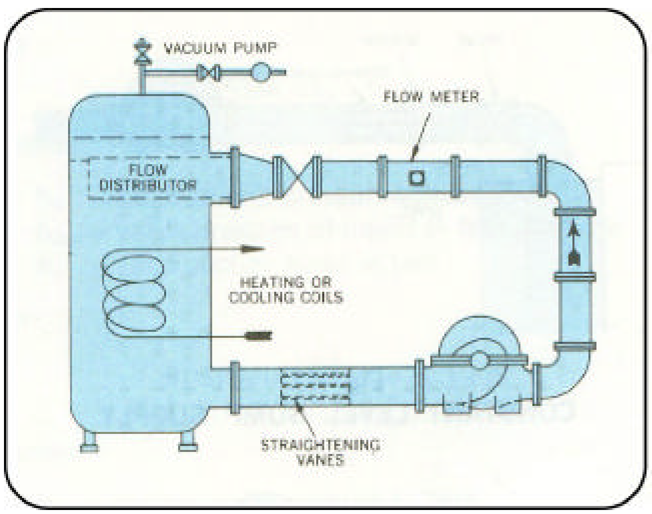

Figure 3-7 Test rig used to measure the N.P.S.H. required (courtesy of the Hydraulic Institute www.pumps.org).

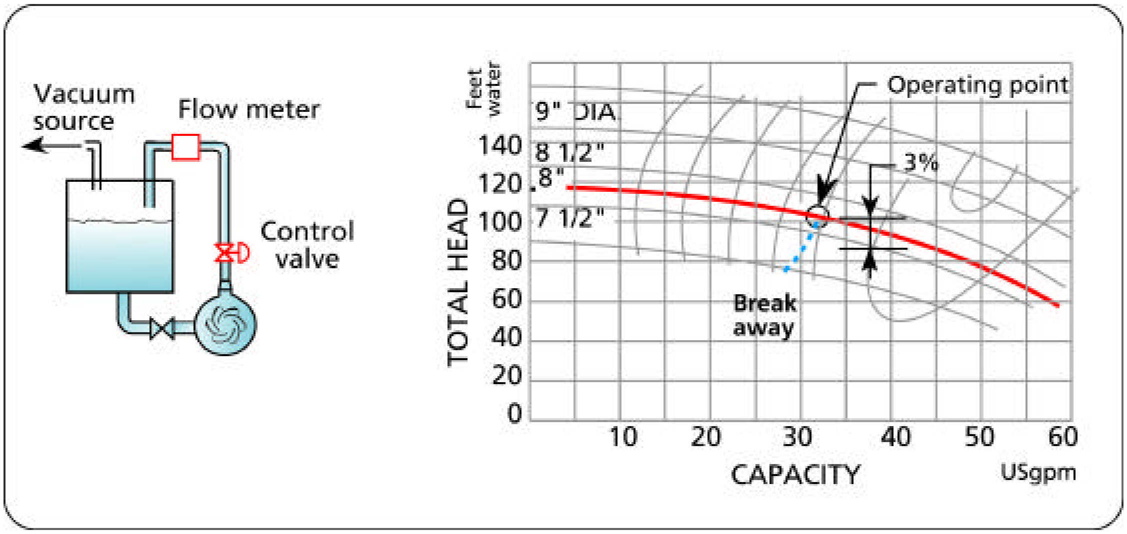

Figure 3-8 Measuring the drop in total head to define the N.P.S.H. required at the operating point.

GUIDELINE FOR THE LEVEL OF N.P.S.H. AVAILABLE

As stated, a total head drop of 3% is the criteria for setting the level of N.P.S.H. required. Since this results in a performance drop then the user should ensure that there is a higher N.P.S.H. available. The recommendation that you will find in the literature is to have 5 ft of water absolute or a 15% margin above the N.P.S.H. required whichever is greatest.

HOW CAN THE N.P.S.H. AVAILABLE BE INCREASED AND CAVITATION AVOIDED

Table 1 gives the major components of N.P.S.H. available and how they affect the level of N.P.S.H. available.

The N.P.S.H. available depends on:	**Effect on N.P.S.H.A.**
1. The friction loss in the pump suction line	The higher the friction loss, the lower the N.P.S.H.
2. The height of the suction tank fluid surface with respect to the pump suction	The lower the height of the fluid surface, the lower the N.P.S.H.
3. The pressure in the suction tank	This cannot be changed for atmospheric tanks. For tanks that are pressurized, the lower the pressure, the lower the N.P.S.H. available.
4. The atmospheric pressure	This cannot be changed. The lower the atmospheric pressure, the lower the N.P.S.H. available.
5. Fluid temperature	An increase in fluid temperature, increases the vapor pressure of the fluid which decreases the N.P.S.H. available.

Table 3-1. How to affect the N.P.S.H. available.

EXAMPLE 3.1 – CALCULATE THE NET POSITIVE SUCTION HEAD AVAILABLE

Water is to be pumped at a rate of 500 USGPM from a sump. The owner prefers to use a self-priming centrifugal pump rather than a submersible pump. Determine the N.P.S.H. available.

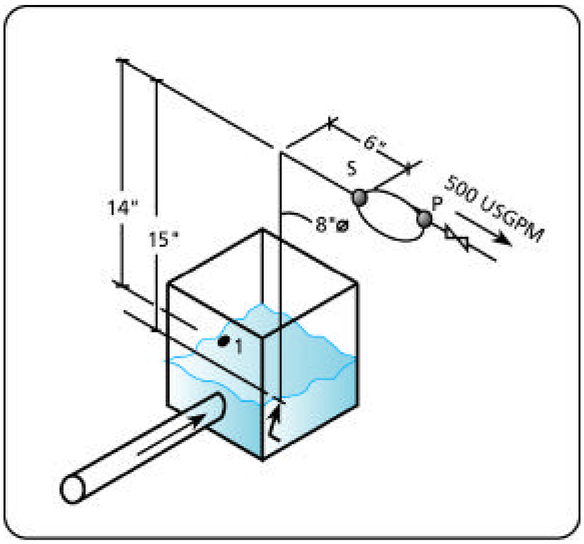

Figure 3-9 Typical example of a pump with low suction head.

Equation [3-10] is applied and SG = 1, $H_1 = 0$, $v_1 = 0$:

$$N.P.S.H._{avail}(ft\ fluid\ abs.) = -(\Delta H_{F1-S} + \Delta H_{EQ1-S}) + z_1 - z_S + \frac{2.31}{1} \times (p_B(psia) - p_{va}(psia))$$

The check valve has a pressure head drop of 5 feet, $\Delta H_{EQ1-S} = 5$ ft. The suction line total friction loss (ΔH_{F1-S}) is 0.54 ft of fluid. The barometric pressure is $p_{va} = 0.25$ psia and the vapor pressure for water at 60^0F is $p_B = 14.7$ psia. The height z_1 - z_S between the surface level of the tank and the pump centerline is -14 ft.

$$N.P.S.H._{avail}(ft\ fluid\ abs.) = -(0.54+5) - 14 + \frac{2.31}{1} \times (14.7 - 0.25) = 13.8\ ft$$

The N.P.S.H.A. is 14 ft of fluid absolute.

There should be no problem finding a suitable pump with 14 ft fluid absolute N.P.S.H.A..

3.4 PUMP INTAKE SUCTION SUBMERGENCE

The pump suction intake must be submerged sufficiently to avoid the formation of vortexes on the liquid surface of the suction tank. These vortexes can take many shapes and forms (see Figure 3-10).

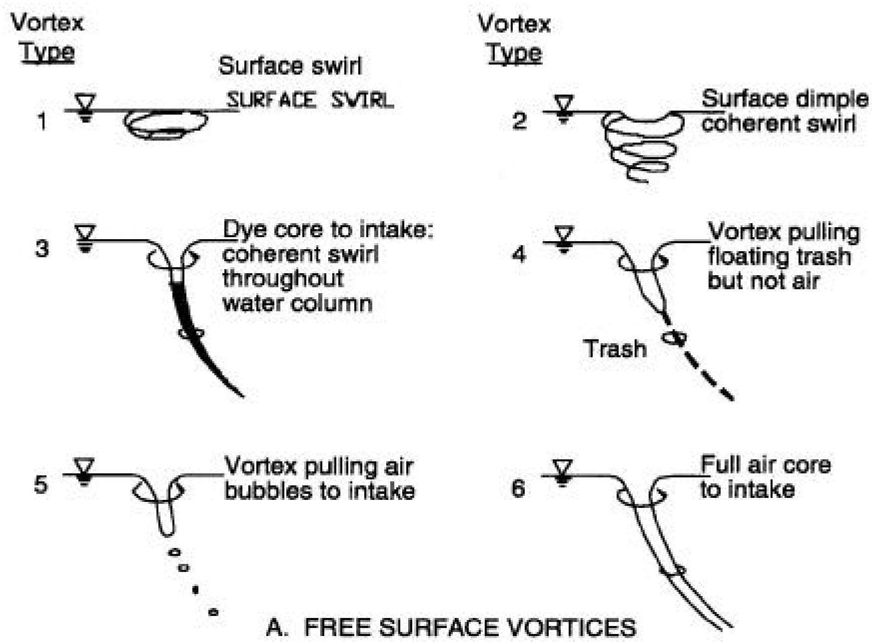

Figure 3-10 Vortex shapes (reprinted with permission of the Hydraulic Institute).

The formation of vortexes between the pump suction intake and the suction tank fluid surface causes air to enter the pump suction. This mixture of air and water in the pump reduces the pump capacity. The formation of such vortexes must therefore be avoided. There is a relationship between the intake velocity at the suction intake, and the submergence (S) of the intake. (For more information consult the Hydraulic Institute's Pump Intake Design manual ANSI/HI9.8-1998).

There are many possible intake design geometries, a few are shown in Figure 3-10; they all have in common a minimum requirement for submergence to avoid the formation of vortexes. The minimum value for submergence (S) to avoid vortex formation is given in Figure 3-12.

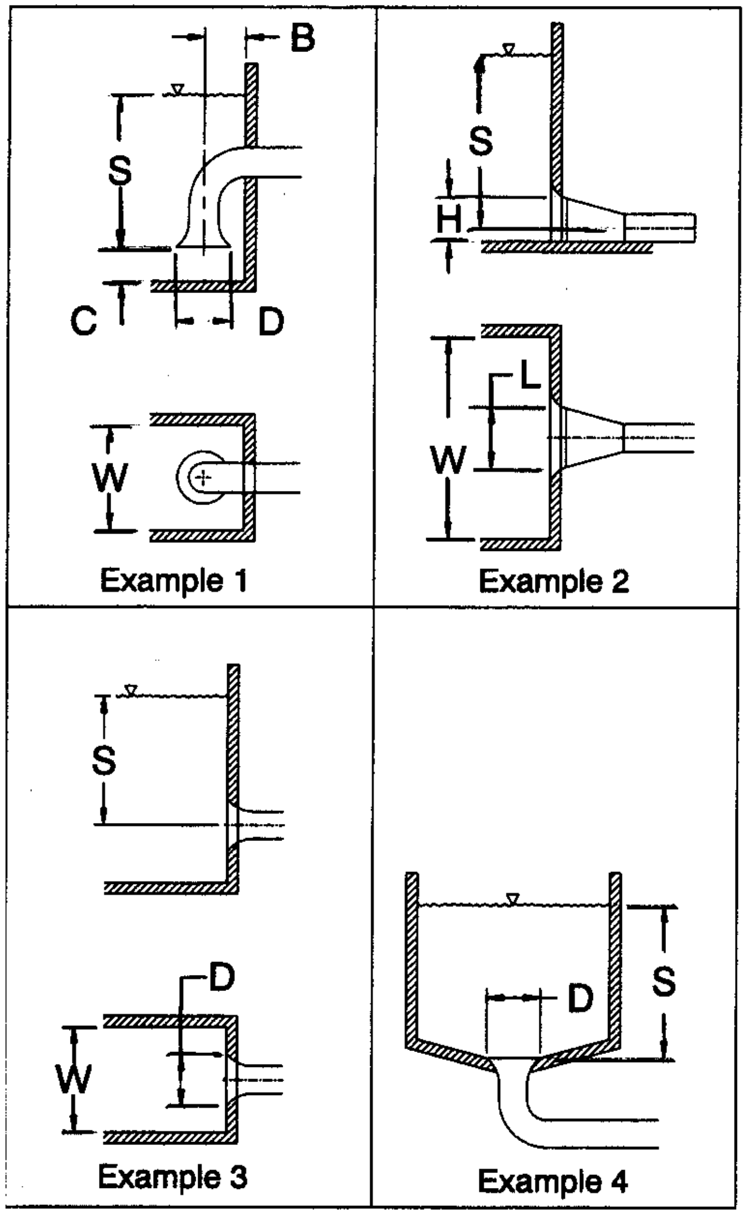

Figure 3-11 Examples of different intake designs (reprinted with permission of the Hydraulic Institute).

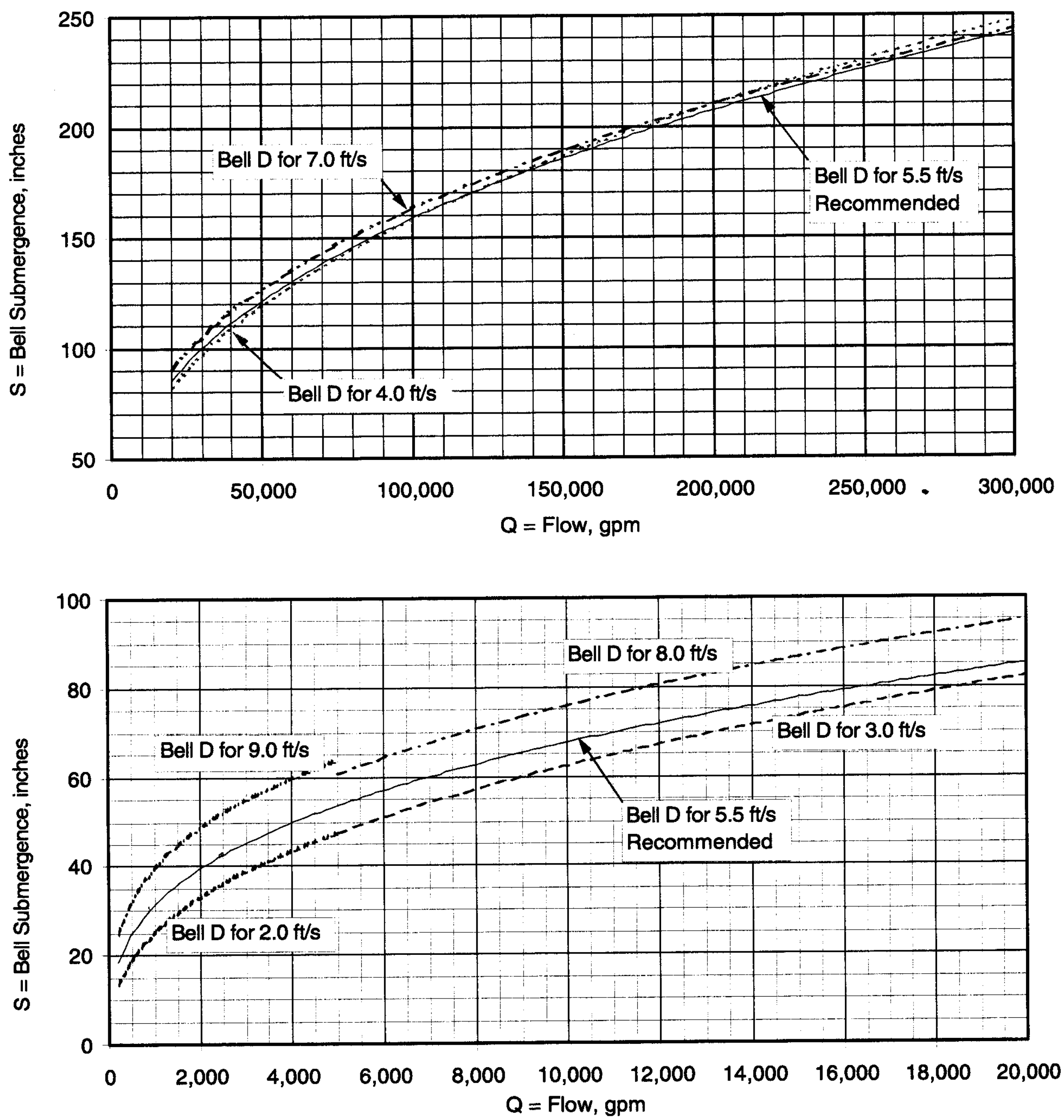

Figure 3-12 Minimum submergence requirements vs. flow and various velocities at the suction pipe intake (reprinted with permission of the Hydraulic Institute).

The values for submergence that are given in the chart above can be calculated with the following equation:

$$S(in) = D(in) + 0.574 \times \frac{q(USgpm)}{D(in)^{1.5}}$$

3.5 DISCHARGE STATIC HEAD (ΔH_{DS})

The Discharge Static Head is the sum of the elevation and pressure head at the outlet of the system, minus the elevation of the pump center line as stated in equation [3-12]. Its value depends on the elevation and pressure head at the outlet (point 2) of the system. When the discharge pipe end is submerged, the outlet of the system is located at the discharge fluid surface (see Figure 3-13A). When the discharge pipe end is not submerged, the outlet of the system is located at the discharge pipe end (see Figure 3-13B).

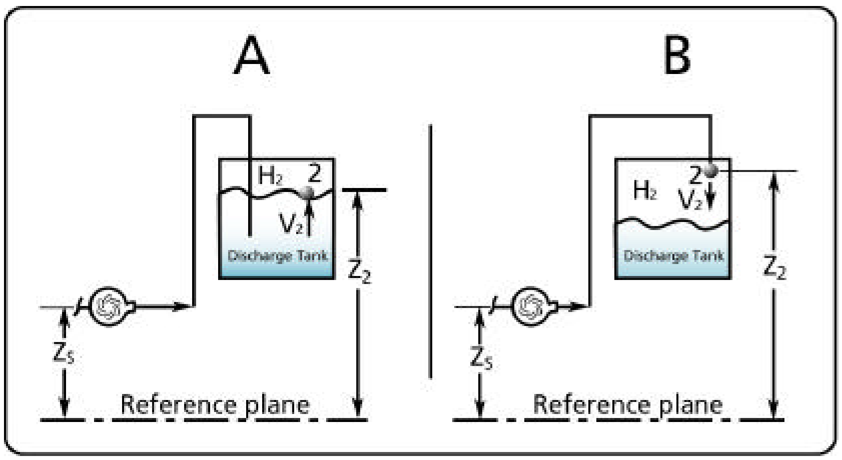

Figure 3-13 Discharge static head with submerged and open discharge pipe end.

H_2 is the pressure head at the discharge tank fluid surface. If the tank is open to atmosphere then $H_2 = 0$.

In all cases, the discharge static head is:

$$\Delta H_{DS} = (z_2 + H_2 - z_S) \qquad [3\text{-}12]$$

Very often pipes enter a tank from the top and finish lower than the liquid surface of the tank as in Figure 3-13A. The outlet of the system remains at point 2 since the fluid particles eventually have to get to point 2.

EXAMPLE 3.2 – CALCULATE THE SUCTION & DISCHARGE STATIC HEAD

A paper machine uses large volumes of steam in the paper drying process. This steam is condensed and recovered in a final stage under vacuum conditions and the condensate is pumped to a flash tank. The flash tank is kept under controlled pressure conditions and receives condensate from other areas of the plant. Its purpose is to collect condensate from various sources and return it to the boilers. What is the suction and discharge static head of the vacuum receiver system if p_1 = -10 in of Hg and p_2 = 2 psig?

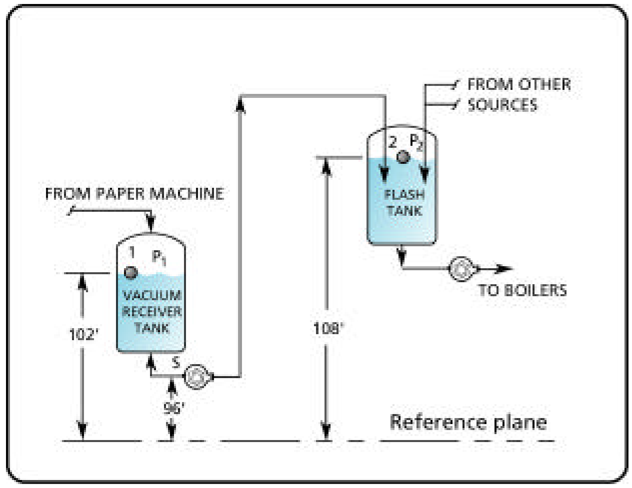

Figure 3-14 An example of a suction tank with low pressure and a discharge tank under pressure.

Suction Static Head

$$\Delta H_{SS} = z_1 + H_1 - z_S \qquad [3\text{-}13]$$

The pressure p_1 is converted to psia:

$$p_1(psia) = 14.7 - (p_1(in\,Hg) \times 0.4918) = 9.8\,psia$$

The specific gravity of condensate at the temperature corresponding to 9.8 psia is very close to 1 (SG=1). To convert from psia to feet of fluid relative:

$$H_1 = \frac{(14.7 - p_1(psia)) \times 2.31}{SG} = \frac{(14.7 - 9.8) \times 2.31}{1} = -11.3\,ft\,of\ \,fluid$$

The above values are substituted into the equation [3-13]:
ΔH_{SS} = (102 - 11 - 96) = -5.3 ft of fluid.

The suction static head is -5.3 feet of fluid.

Discharge Static Head

$$\Delta H_{DS} = z_2 + H_2 - z_S \qquad [3\text{-}14]$$

Convert the pressure p_2 to feet of fluid:

$$H_2 = \frac{p_2 \times 2.31}{SG} = \frac{2 \times 2.31}{1} = 4.6\ ft\,of\ \,fluid$$

The above values are substituted into the equation [3-14]:
ΔH_{DS} = (108 + 4.6 - 96) = **16.6 feet of fluid.**
The discharge static head is 16.6 feet of fluid.

3.6 VELOCITY HEAD DIFFERENCE (ΔH_v)

The Velocity Head Difference is the head (expressed in ft of fluid) corresponding to the kinetic energy variation between the outlet and the inlet of the system.

$$\Delta H_v = \frac{1}{2g}(v_2^2 - v_1^2)$$

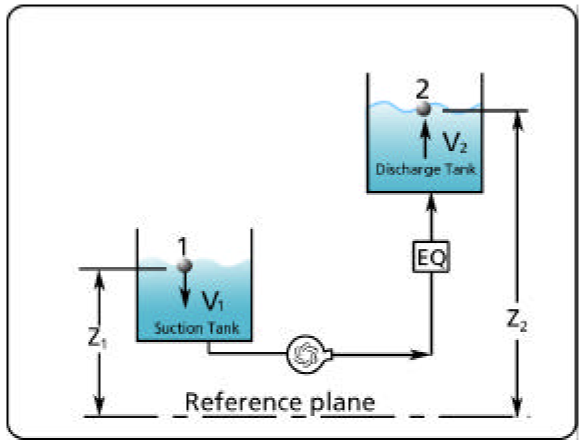

Figure 3-15 Inlet and outlet velocities of a typical system.

The Velocity Head is defined as $v^2/2g$. In the case where a suction line is connected to a tank, point 1 is on the fluid surface of the suction tank (see Figure 3-15). This surface will be moving downwards very slowly, so that v is very small and therefore $v_1^2/2g \approx 0$. If point 1 is in fact a connection on another line, the velocity will have to be considered. Identical reasoning can be applied to the exit point 2. The Velocity Head is not normally a significant part of the Total Head, typically 1 or 2 ft of fluid. However, some systems are designed with nozzles at the exit point, in order to accelerate the fluid and generate a high velocity. A good example of this is a paper machine head-box. The fiber slurry is ejected from the wide and narrow opening (slice lip) of the head-box at the same velocity as the paper machine, which in certain cases is as high as 4000 ft/min (45 miles per hour). In this case, the velocity head may represent as much as half of the pump Total Head.

3.7 EQUIPMENT PRESSURE HEAD DIFFERENCE (ΔH_{EQ})

The Equipment Pressure Head difference is pressure head loss due to friction imposed by the equipment (for example, control valve. filter, etc.). Isolation valves and pipe fittings are not considered equipment. The pressure head loss for equipment vs. flow rate is usually obtained from tables, charts or graphs provided by the supplier of the equipment.

The total Equipment Pressure Head Difference between the inlet and the outlet of a system is made up of the sum of the pressure head drops across each piece of equipment.

$$\Delta H_{EQ1-2} = \Delta H_{EQ1} + \Delta H_{EQ2} + \ldots$$

CONTROL VALVES

A typical pumping system has at least one control valve in the circuit. Depending on the type of valve, its opening, the upstream pressure, and the flow, it is possible to calculate the pressure head drop by consulting the appropriate supplier data tables. This pressure head drop is then added to the Total head of the pump to ensure that there is enough energy to move the fluid through the system at the design flow rate. During the design stage, a simple way to account for this is to assume or fix the pressure head drop of the control valve. If we assume a pressure head drop across the valve of **10 ft of fluid,** then it will be generally possible to select a valve that will give this pressure head drop at a reasonable opening of say 90%. In other words, by using a Δp of 10 ft for the pressure head drop, we have fixed one of the parameters required to size a valve, without unduly restricting the task of sizing the valve. Ten (10) ft of pressure drop is a common value used in designing systems with control valves. This criterion will generally result in a valve size that is one size smaller than the line.

A more practical approach is required for existing systems with a control valve. We will need the manufacturer's tables for the valve which gives a CV coefficient which is proportional to the pressure drop and flow rate for a given valve opening (see equation [3-15]). Obtain the valve opening, while the system is running normally. With this information, using the manufacturer's catalog you can obtain a CV value. Calculate the pressure drop and convert to pressure head drop. Use this value in the calculations for Total Head.

The definition of the *CV* coefficient is:

$$CV\left(\frac{USgpm}{psi^{1/2}}\right)=\frac{q\ (USgpm)}{\sqrt{\frac{\Delta p\ (psi)}{SG}}} \qquad [3\text{-}15]$$

where q and Δp are respectively the flow rate and pressure drop across the valve for a given valve model and opening.

EQUIPMENT

Any equipment in the line, such as filters, nozzles, etc., will have a specified pressure drop at a certain flow rate that is available in the literature or from the equipment supplier. Occasionally, certain types of equipment require a specific upstream pressure in order to operate properly. To accommodate this, we need to determine what the pressure is at the proposed equipment location. If the calculation shows the upstream pressure to be lower than required, it will have to be raised artificially by closing a manual valve on the downstream side of the equipment. The Total Head will then have to be increased to accommodate this added pressure head. Another option is to move the equipment closer to the pump where the line pressure is higher.

In the event that the calculation shows the pressure immediately upstream of the equipment to be higher than required, it will have to be lowered artificially by closing a manual valve on the upstream side of the equipment. The Total Head will then have to be increased to accommodate this added pressure head. Alternatively, we can try moving the equipment closer to the discharge end where the line pressure is usually lower.

3.8 PIPE FRICTION HEAD DIFFERENCE FOR NEWTONIAN FLUIDS (ΔH_{FP})

The Friction Head is the friction due to the movement of fluid in a piping system and is proportional to flow rate, pipe diameter and viscosity. Tables of values for Friction Head are available in references 1 & 8.

The Friction Head, as defined here, is made up of the friction loss due to the fluid movement and the friction loss due to the effect of pipe fittings (for example, 90° elbows, 45° bends, tees, etc.):

$$DH_F = DH_{FP} + DH_{FF}$$

the subscript FP refers to pipe friction loss and the subscript FF to fittings friction loss.

NEWTONIAN FLUIDS

Newtonian fluids are a large class of fluids, whose essential property VISCOSITY, was first defined by Newton (see Appendix A for a list of Newtonian and non-Newtonian fluids). Viscosity is the relationship between the velocity of a given layer of fluid and the force required to maintain that velocity. Newton theorized that for most pure fluids, there is a direct relationship between force required to move a layer and its velocity. Therefore, to move a layer at twice the velocity, required twice the force. His hypothesis could not be tested at the time, but later the French researcher, Poiseuille, demonstrated its validity. This resulted in a very practical definition for viscosity (see Appendix A for more details).

The Darcy-Weisbach formula expresses the resistance to movement of any fluid in a pipe:

$$\frac{\Delta H_{FP}}{L} = f\,\frac{v^2}{D \times 2g}$$

where f is a non dimensional friction factor. Often, the tables give values for friction loss in terms of ft of fluid per 100 ft of pipe. When the appropriate units are used (Imperial system), the Darcy-Weisbach equation becomes:

$$\frac{\Delta H_{FP}}{L}\left(\frac{ft\ of\ fluid}{100\ ft\ of\ pipe}\right) = 1200\ f\ \frac{v^2\,(ft/s)^2}{D(in) \times 2g\,(ft/s^2)} \qquad [3\text{-}16]$$

The friction factor is proportional to the Reynolds number which is defined as:

$$R_e = 7745.8 \frac{v(ft/s)\ D(in)}{n\ (cSt)} \qquad [3\text{-}17]$$

The Reynolds number is proportional to the kinematic viscosity, the average velocity, and the pipe inside diameter. It is a non dimensional number. The kinematic viscosity (ν) is the ratio of the absolute viscosity (μ) to the fluid specific gravity (SG).

$$n(cSt) = \frac{m(cP)}{SG}$$

Values of viscosity for many fluids can be found in reference 8.

Laminar flow - R_E < 2000

Distinct flow regimes can be observed as the Reynolds number is varied. In the range of 0 to 2000, the flow is uniform and is said to be laminar. The term laminar refers to successive layers of fluid immediately adjacent to one another, or laminated. Looking at a longitudinal section of the pipe, the velocity of individual fluid particles is zero close to the wall and increases to a maximum value at the center of the pipe with every particle moving parallel to its neighbor. If we inject dye into the stream, we would notice that the dye particles maintain their cohesion for long distances from the injection point.

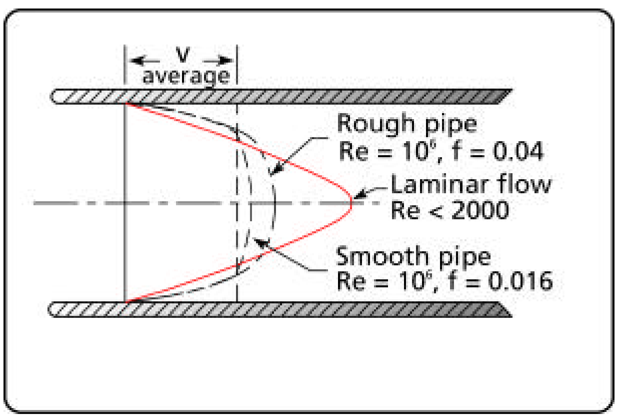

Figure 3-16 Laminar and turbulent flow velocity profiles.

The friction loss is generated within the fluid itself. Figure 3-16 shows that each layer (in this case each ring) of fluid is moving progressively faster as we get closer to the center. The difference in velocity between each fluid layer causes the friction loss.

The friction factor *f* is given by:

$$\boxed{f = \frac{64}{R_e}} \qquad [3\text{-}18]$$

For viscous fluids (i.e.: $n \geq 50$ SSU), the combination of velocity and viscosity usually produces a low Reynolds number and therefore laminar flow. Pumping viscous fluids at a faster rate may cause the fluid to become turbulent resulting in high friction losses. The tables for viscous fluid friction loss given in references 1 & 8 are based on the equation for laminar flow, equation [3-18]. This equation can be theoretically derived and is found in most fluid dynamic volumes (see reference 11). An interesting aspect of laminar flow is that pipe roughness is not a factor in determining friction loss.

Unstable flow - 2000 <R_E <4000

The flow is pulsing and unstable and appears to possess characteristics of both laminar and turbulent flow.

Turbulent flow - R_E > 4000

At Reynolds number larger than 4000, it is very difficult to predict the behavior of the fluid particles, as they are moving in many directions at once. If dye is injected into the stream, the dye particles are rapidly dispersed, demonstrating the complex nature of this type of flow. Reynolds, who originally did this experiment, used it to demonstrate the usefulness of a non-dimensional number (the Reynolds number) related to velocity and viscosity. Most industrial applications involve fluids in turbulent flow. The geometry of the wall (pipe roughness) becomes an important factor in predicting the friction loss.

Many empirical formulas for turbulent flow have been developed. Colebrook's equation is the one most widely accepted :

$$\boxed{\frac{1}{\sqrt{f}} = -2\,\log_{10}\left(\frac{e}{3.7\,D} + \frac{2.51}{R_e\sqrt{f}}\right)} \qquad [3\text{-}19]$$

where *e* is the average height of protuberances (absolute roughness) of the pipe wall surface (for example, 0.00015 ft for smooth steel pipe). The term *e/D* is called the pipe roughness parameter or the relative roughness. The tables for friction loss given in references 1 & 8 are based on Colebrook's equation. Since it is not possible to derive an explicit solution for *f,* L.F. Moody (see Figure 3-18) developed a graphical solution. The diagram shows the linear relationship of the friction factor (f) with the Reynolds number (R_e) for the laminar flow regime. For Reynolds numbers in the medium range (4,000 to 1,000,000, turbulent flow), the friction factor is dependent on the Reynolds number and the pipe roughness parameter, which is known as the transition zone. For high Reynolds numbers (1,000,000 and higher, fully turbulent), the friction factor is independent of the Reynolds number and is proportional only to the pipe roughness parameter. This is the zone of complete turbulence.

Some typical values for the absolute roughness *e* :

PIPE MATERIAL	Absolute roughness ε (ft)
Steel or wrought iron	0.00015
Asphalt-dipped cast iron	0.0004
Galvanized iron	0.0005

Table 3-2 Typical values for pipe wall roughness.

Values for ε are given in reference 1 and 8.

A numerical method, the Newton-Raphson iteration technique, can also be used to solve the Colebrook equation (see Appendix B). Note: the Colebrook equation is valid only for Newtonian fluids (see Appendix A for a list of Newtonian and non-Newtonian fluids).

Another equation developed by Swamee and Jain, gives an explicit result for f and agrees with the Colebrook equation within 1%:

$$\boxed{f = \frac{0.25}{\left(\log_{10}\left(\frac{e}{3.7\,D} + \frac{5.74}{R_e^{0.9}}\right)\right)^2}} \qquad [3\text{-}20]$$

3.9 FITTINGS FRICTION HEAD DIFFERENCE FOR NEWTONIAN FLUIDS, K METHOD AND 2K METHOD (ΔH_{FF})

THE K METHOD

The fittings friction loss is given by:

$$\Delta H_{FF}(ft\ fluid) = K\frac{v^2(ft/s)^2}{2g(ft/s^2)} \qquad [3\text{-}21]$$

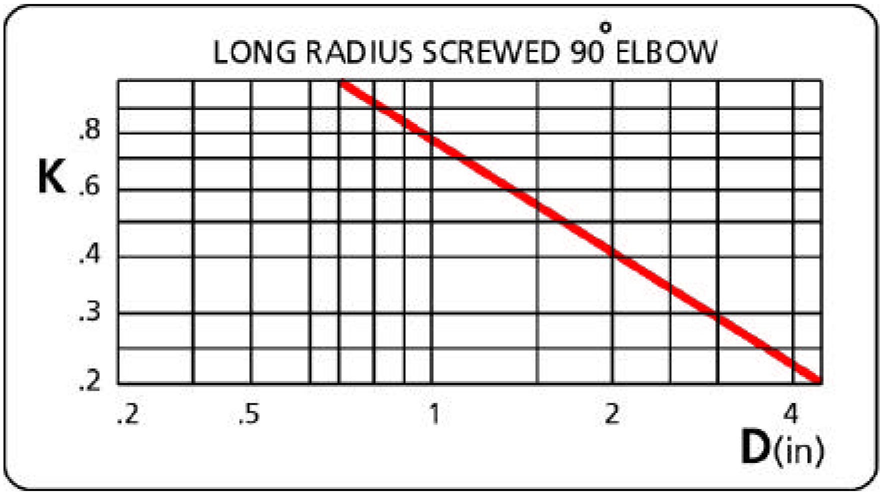

Figure 3-17 Typical values for K with respect to fitting diameter.

The K factor for various fittings can he found in many publications (references 1 and 8). As an example, Figure 3-17 depicts the relationship between the K factor of a 90° screwed elbow and the diameter (D). The type of fitting dictates the relationship between the friction loss and the pipe size.

Note: this method assumes that the flow is fully turbulent (see the demarcation line on the Moody diagram of Figure 3-18).

THE TWO K METHOD (SEE REFERENCE 14)

Tests conducted on various fittings have determined that the *K* value is not dependent on size, but on the Reynolds number. This approach takes into account the different nature of laminar and turbulent flow.

$$K = \frac{K_1}{\mathrm{Re}} + K_{\infty} \qquad [3\text{-}22]$$

where K_1 and $K_{¥}$ are constants appropriate to the geometry of the fitting (see Table 3-3). The examples in this book use the 2K method.

2K METHOD PARAMETERS

FITTING TYPE	K_1	K¥
90° ELBOWS		
Standard (R/D =1) screwed	800	0.4
Standard (R/D =1) flanged/welded	800	0.25
Long radius (R/D=1.5) all types	800	0.2
Mitered (R/D=1.5) 1 weld 90°	1000	1.15
Mitered (R/D=1.5) 2 weld 45°	800	0.35
Mitered (R/D=1.5) 3 weld 30°	800	0.3
Mitered (R/D=1.5) 4 weld 22 1/2°	800	0.27
Mitered (R/D=1.5) 5 weld 18°	800	0.25
45° ELBOWS		
Standard (R/D =1) all types	500	0.2
Long radius (R/D=1.5) all types	500	0.15
Mitered 1 weld 45°	500	0.25
Mitered 2 weld 22 1/2°	500	0.15
180° ELBOWS		
Standard (R/D =1) screwed	1000	0.6
Standard (R/D =1) flanged/welded	1000	0.35
Long radius (R/D=1.5) all types	1000	0.3
TEES, AS ELBOWS		
Standard screwed	500	0.7
Long radius screwed	800	0.4
Standard flange or welded	800	0.8
Stub-in branch type	1000	1
TEES, AS RUN-THROUGH		
Standard screwed	200	0.1
Long radius screwed	150	0.5
Stub-in branch type	100	0
VALVE: GATE, BALL OR PLUG		
Full line size β=1.0	300	0.1
Full line size β=0.9	500	0.15
Full line size β=0.8	1000	0.25
Globe standard	1500	4
Globe, angle or Y-type	1000	2
Diaphragm, dam type	1000	2
Butterfly	800	0.25

FITTING TYPE	K_1	$K_{¥}$
CHECK VALVES		
Lift	2000	10
Swing	1500	1.5
Tilting Disk	1000	0.5
PIPE ENTRANCES AND EXITS		
Pipe entrance, normal	160	0.5
Pipe entrance, inward projecting	160	1
Pipe exit	0	1

Table 3-3 Typical values for K_1 and $K_{¥}$ for the 2K method.

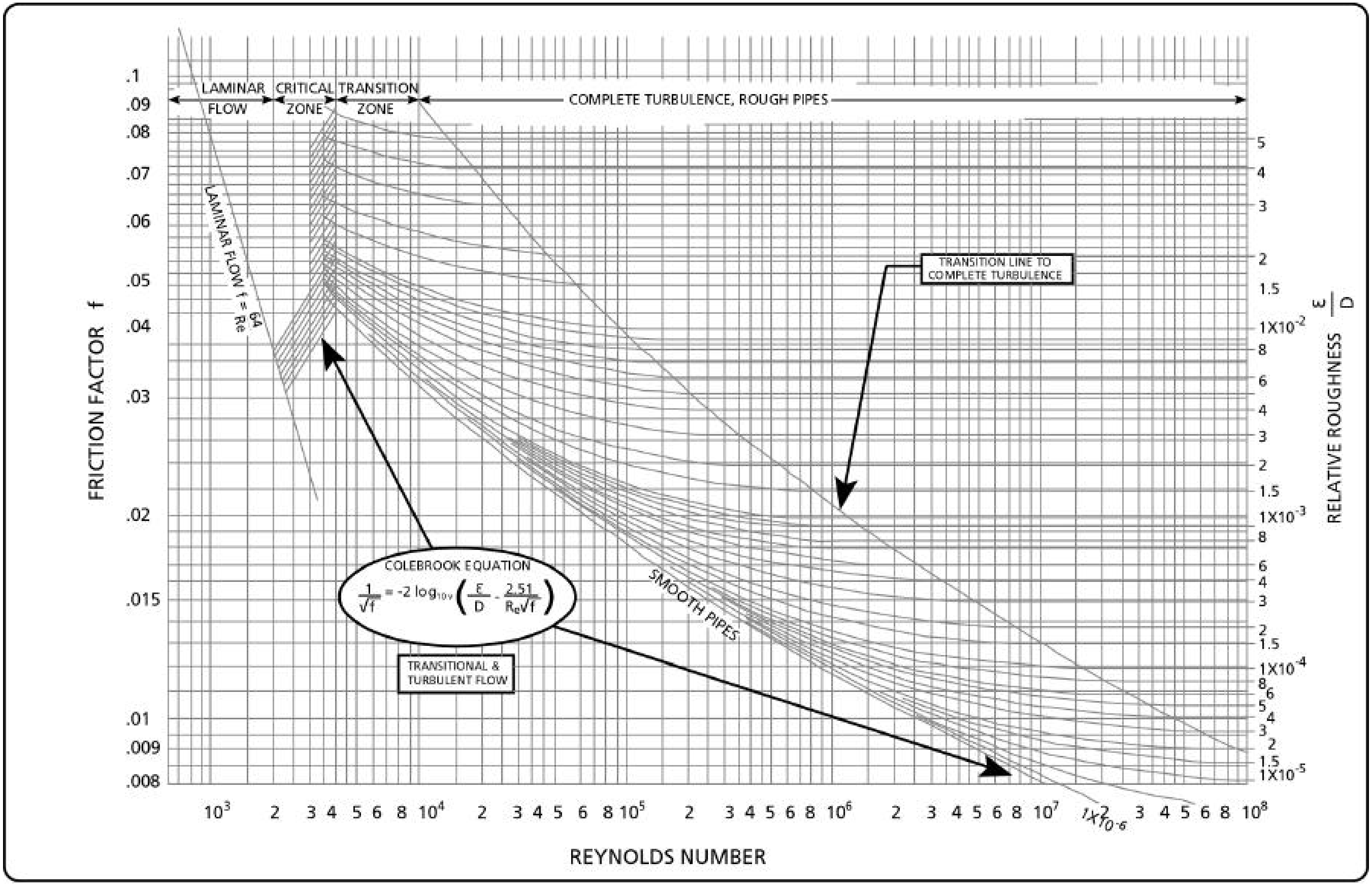

Figure 3-18 The Moody diagram, friction factor vs. Reynolds number for laminar and turbulent flow at various pipe roughness-values.

3.10 PIPE FRICTION HEAD DIFFERENCE FOR WOOD FIBER SUSPENSIONS (ΔH_{FP})

A. PIPE FRICTION LOSS

Empirical data for wood fiber suspensions (usually referred to as stock) have been gathered and correlated for many different types of pulps. Depending on flow rate and type of stock, different characteristic regions of flow friction loss vs. velocity have been established.

The friction loss curve for chemical pulp can be conveniently divided into three regions, as illustrated by the hatched areas of the two next figures.

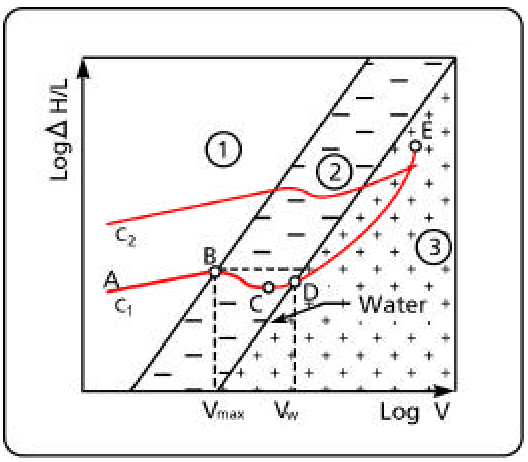

Figure 3-19 Pipe friction loss vs. velocity and consistency for chemical pulp.

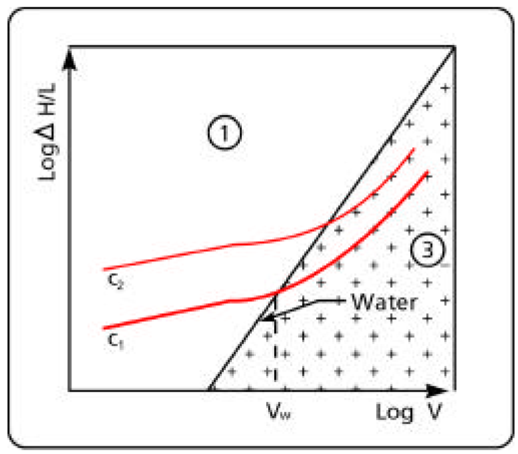

Figure 3-20 Pipe friction loss vs. velocity and consistency for mechanical pulp.

REGION 1

Curve AB is a linear region where friction loss for a given pulp is a function of consistency, velocity, and pipe diameter. The velocity at the upper limit of this linear region (Point B) is designated v_{max}.

REGION 2

Curve BCD shows an initial decrease in friction loss (to Point C), after which the friction loss again increases. The intersection of the pulp friction loss curve with the friction loss curve for water (Point D) is termed the onset of drag reduction. The velocity at this point is designated v_w.

REGION 3

Curve DE shows the friction loss curve for pulp fiber suspensions below the friction loss curve for water. This is due to a phenomenon called drag reduction.
Regions 2 and 3 are separated by the friction loss curve for water, which is a straight line with a slope approximately equal to 1.75, when plotted with log-log coordinates.

The friction loss curve for mechanical pulp, as illustrated in Figure 3-19, is divided into only two regions: Region 1 and 3. For this pulp type, the friction loss curve crosses the water curve at v_w. There is no true v_{max}.

PIPE FRICTION ESTIMATION PROCEDURE

The bulk velocity (v) will depend upon the mass flow rate and pipe diameter (D) selected. The final value of *v* can be optimized to give the lowest capital investment and operating cost with due consideration of future demands or possible system expansion. The mass flow rate of wood fiber is of particular interest in the design of pipes and pumping systems since the purpose of the solution is to convey the fiber. The mass flow rate has the following relationship between the volumetric flow and the pulp fiber consistency:

$$M(Tons/day) = \frac{q(USgal./\mathrm{min}) \;\; \%C}{16.64} \qquad [3\text{-}23]$$

where *M* : mass flow rate of pulp;
C : pulp dry consistency ratio expressed as a percentage.

and

$$q(US\,gal./\mathrm{min}) = 2.45\; v(ft/s)\; (D(in))^2 \qquad [3\text{-}24]$$

The bulk velocity will fall into one of the regions previously discussed. When this region is identified, the appropriate correlations for determining pipe friction loss values may be selected. The following describes the procedure to be used for estimating pipe friction loss in each of the regions.

REGION 1

Region 1 is delimited by the bulk velocity of the stock (v), between the ranges:

$$v < v_{max}$$

where $v_{max} = K'C^s$ and

K' : Numerical coefficient (constant for a given pulp)
s : Exponent (constant for a given pulp)

The relationship between the friction loss and the governing parameters is:

$$\frac{\Delta H_{FP}}{L}\left(\frac{ft of\ water}{100\, ft of\ pipe}\right) = F\;\; K\;\; (v(ft/s))^a \;\; (C(\%))^b \;\; (D(in))^g \qquad [3\text{-}25]$$

where *K* : numerical coefficient (constant for a given pulp);
a, b, g : exponents (constant for a given pulp).

The factor *F* is composed of many possible correction factors:

$$F = F_1 \times F_2 \times F_3 \times F_4 \times F_5$$

where F_1 : correction factor for temperature;
F_2 : correction factor for pipe roughness;
F_3 : correction factor for pulp type;
F_4 : correction factor for beating;
F_5 : design contingency factor;

and F_1 is calculated by:

$$F_1 = 1.35 - 0.01T(^\circ C)$$

REGION 2

Region 2 is delimited by the bulk velocity of the stock v, between the ranges:

$$v_{\max} < v < v_W$$

where $v_W = 4 \times C^{1.4}$

If v is between v_{max} and v_W, equation [3-25] may be used to determine DH_{FP}/L at the maximum point v_{max}. The friction loss is then estimated and can be assumed to be constant for velocities in this region.

REGION 3

Region 3 is delimited by the bulk velocity of the stock v for the region:

$$v > v_W$$

A conservative estimate of friction loss is obtained by using the water curve as determined by Blasius' equation:

$$\frac{\Delta H_{FP}}{L}\left(\frac{ft\ of\ \ water}{100\ \ ft\ of\ \ pipe}\right) = 0.58\ \frac{(v(ft/s))^{1.75}}{(D(in))^{1.25}} \qquad [3\text{-}26]$$

Here Blasius' equation is used rather than Colebrook's (see equation [3-19], because the friction values for pulps were determined using smooth pipe (notably stainless steel copper and PVC), so that pipe roughness was not a factor in the determination of pressure drop. Blasius' equation is an accurate representation of friction values for water in such a case.

Previously published methods for calculating pipe friction loss of pulp suspensions gave a very conservative estimate of head loss, whereas the method just described gives a more accurate estimate.

Wood fiber does not significantly affect the overall density of the fiber-water solution. The specific gravity of the solution is therefore the same as water.

NO.	PULP TYPE	K`	σ	K	α	β	γ
1	Cooked Groundwood	0.75	1.8	6.2	0.43	2.31	-1.2
2	Hardwood Nssc Csf=620	0.59	1.8	4.56	0.43	2.31	-1.2
3	Long Fibered Kraft Never Dried Csf=260	0.75	1.8	17	0.31	1.81	-1.34
4	Unbleached Sulphite	0.98	1.2	12.69	0.36	1.89	-1.33
5	Stone Groundwood Csf=114	4.0	1.4	3.81	0.27	2.37	-0.85
6	Unbeaten Aspen Sulphite Never Dried	0.85	1.6	5.3	0.36	2.14	-1.04
7	Kraft	0.98	1.2	11.4	0.36	1.89	-1.33
8	Bleached Sulphite	0.98	1.2	11.4	0.36	1.89	-1.33
9	Bleached Straw	0.98	1.2	11.4	0.36	1.89	-1.33
10	Newsprint Broke Csf=75	4.0	1.4	5.19	0.36	1.91	-0.82
11	Long Fibered Kraft Never Dried Csf=550	0.75	1.65	12.1	0.31	1.81	-1.34
12	Long Fibered Kraft Never Dried Csf=725	0.98	1.85	11.8	0.31	1.81	-1.34
13	Long Fibered Kraft Never Dried Csf=650	0.85	1.9	11.3	0.31	1.81	-1.34
14	Soda	4.0	1.4	6.5	0.36	1.85	-1.04
15	Long Fibered Kraft Dried & Re-Slurried	0.49	1.8	9.4	0.31	1.81	-1.34
16	Bleached Kraft Pine Dried & Re-Slurried	0.79	1.5	8.8	0.31	1.81	-1.34
17	Refiner Groundwood Csf=150	4.0	1.4	3.4	0.18	2.34	-1.09
18	Unbleached Straw	0.98	1.2	5.7	0.36	1.89	-1.33
19	Kraft Birch Dried & Re-Slurried	0.69	1.3	5.2	0.27	1.78	-1.08
20	Refiner Groundwood (Hardboard)	4.0	1.4	2.3	0.23	2.21	-1.29
21	Refiner Groundwood (Insulating Board)	4.0	1.4	1.4	0.32	2.19	-1.16

Table 3-4 Friction factor parameters for pulp suspensions (use with equation [3-25]).

Notes: 1. Original data obtained in stainless steel and PVC pipe. PVC is taken to be hydraulically smooth pipe.
2. No safety factors are included in the above correlations.
3. The friction loss depends to a large extent on the condition of the inside surface of the pipe. For cast iron and galvanized pipe, the K values will be reduced. No systematic data are available for the effects of surface roughness.
4. If pulps are not identical to those shown, some engineering judgment is required.
5. Wood is New Zealand Kraft pulp.

The above pulps are presented sequentially in the approximate order of high to lower friction. This is only approximate since certain pulps can produce higher or lower friction depending on the pipe diameter. Some of these pulps have extremely high friction values when conveyed through a small pipe, which of course would contraindicate small pipes, larger pipes will have to be used. However, even if economically feasible, low velocities should be avoided as certain pulps de-water easily when at rest (i.e. Kraft) and can therefore plug the line if not subjected to sufficient movement (i.e. less than 3 ft/s).

B. FITTINGS FRICTION LOSS FOR PULP

The recommended method of handling friction fitting losses (see reference 2) is to use the same friction loss as for water and to adjust the value for pulp fiber consistency. The fittings friction loss is given by:

$$\Delta H_{FF}(ft\ of\ water) = K_P \frac{(v(ft/s))^2}{2g(ft/s^2)} \qquad [3\text{-}27]$$

The loss coefficient for pulp suspensions in a given fitting generally exceeds the loss coefficient for water in the same fitting. As an approximate rule, it is recommended that the fitting friction loss coefficient (K) be increased 20% for each 1% increase in oven-dried stock consistency above 2%. This means that K_P will have the following values with respect to K:

%BD Consistency	K_P
2	K
3	1.2K
4	1.4K
5	1.6K
6	1.8K

Table 3-5 Pulp fittings friction values.

Where can you find help when you need it?

Did you ever ask a pump supplier sales rep. for the price of a pump? Here is a sample dialogue:

- “Could I please have a price for a pump for 3000 gallons a minute?”

- “Certainly sir, what head is required?”

- “Well I'm not sure, what do you suggest?”

- “Well sir that's hard for me to say. You see, I have to know the details of the system, such as the static head, the pipe sizes, the length, the elevation difference, etc.”

- “Well, I want to get water from the ground floor to the 2nd floor level. That's about 40 feet up. For piping size just take something reasonable, it's all new anyway. The total piping length is probably around 400 feet. Is this enough information?”

- “Hum, yes, well I see. Will there be a control valve in the system? how many fittings and of what type will there be? what's your N. P. S.H. available?”

- “My N.P. … what?”

- “Sir, why don't you ask a professional engineer to look at this and have him give me a call later, OK? Have a nice day… .”

I think you get the point.

Many of us who have a general industrial engineering background will have experienced this type of situation. Most equipment can be purchased by specifying only a few parameters. Not so for a pump. Before you can order it, you must analyze the complete system in detail, know the type of fluid and specific gravity, and determine the Total Head, N. P. S.H. available, and motor size.

Let's try this conversation again:

-“M. Pump Salesman, could I please have a price for a centrifugal pump with a capacity of 3000 US gallons a minute of water and a Total Head of 70 feet?”

- “Certainly sir, is this water acidic and does it have any particles in it? If you have particles, are they fine or coarse and more importantly are they abrasive? Oh, and what type of seals do you want, pressurized or mechanical? Do you need....”

Does this wring a bell? Pump salespersons must be very brave since they seem to have no regard for their personal safety. I'm exaggerating of course. Pump reps. are at their best when you start by giving them a P.O. number with the serial number of the pump you want to replace. That's when they shine.

This dialogue shows clearly, how important terminology is in this business. Also, for such an innocent looking piece of equipment, I think you will agree that there is more than meets the eye.

Fluid friction calculation is the most difficult aspect of Total Head calculations. One important fact to establish: is the fluid Newtonian or not? This is because most reference books giving pressure drop coefficients for friction are based on Newtonian fluids; and this is rarely pointed out. Newtonian fluids, while very common, are by no means the majority of all fluids we deal with in industrial engineering (a non-exhaustive list is given in Appendix A).

Equipment head difference is probably the second most difficult element to deal with. If the system is in the design stage and it includes a control valve, then it will generally be possible to select a valve with 10 feet of pressure head drop. This is sufficient information on the valve for Total Head calculation. For a control valve in an existing system you may need to dig deeper and find the exact details (Cv vs. flow curves) and figure out the pressure head drop for that valve in operation. For other equipment, the manufacturer's literature should be checked to determine the correct pressure head drop for the design flow rate.