Pressure and Flow Characteristics

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Any discussion of water system pumps must begin with a discussion of pressure.

Pressure is the force exerted per unit area of a fluid. It can be considered a compressive stress. The most common unit for designating pressure is pounds per square inch (psi), which can be calculated as follows:

\[
\text{Equation 2-1} \quad \text{psi} = \frac{\text{Head (in feet)}}{2.31} \times \text{Specific gravity}
\]

According to Pascal’s principle, if pressure is applied to the surface of a fluid, this pressure is transmitted undiminished in all directions.

The types of pressure are:

- **Gauge pressure**
- **Atmospheric pressure**
- **Absolute pressure**

Gauge pressure is a corrected pressure and is the difference between a given pressure and that of the atmosphere. Atmospheric pressure is the force exerted on a unit area by the weight of the atmosphere. The pressure at sea level due to atmosphere is 14.7 psi (1 atmosphere = 34 feet of water/14.7 psi = 2.31).

The sum of gauge pressure and atmospheric pressure is absolute pressure. The absolute pressure in a perfect vacuum is zero. Absolute pressure of the atmosphere at sea level is 14.7 psi (0 psi gauge).

The term *vacuum* is frequently used in referring to pressures below atmospheric. Due to the common use of a column of mercury to measure vacuum, units are expressed in inches of mercury (in. Hg). (14.7 psi atmospheric pressure equals 30 in. Hg.)

Since water weighs 0.0361 pounds per cubic inch, a column of water 1 square inch in area and 1 foot high will weigh 0.433 pounds. Increasing the pressure 1 psi requires a 2.31-foot increase in height.

While discussing various types of pressures, one should consider vapor pressure. The vapor pressure of a liquid at a specified temperature is the pressure at which the liquid is in equilibrium with the atmosphere or with its vapor in a closed container. At pressures below this vapor pressure at a given temperature, the liquid will start to vaporize due to the reduction in pressure at the surface of the liquid. (At 50°F, the vapor pressure of water is 0.256 psi. At 212°F, it is 14.7 psi.)

**VELOCITY HEAD**

Velocity head represents the kinetic energy per unit weight that exists at a particular point. If velocity at a cross-section were uniform, then the velocity head calculated with this uniform or average velocity would be the true kinetic energy per unit weight of fluid. However, in general, velocity distribution is not uniform. True kinetic energy is found by integrating the differential kinetic energies from streamline to streamline. The kinetic energy correction factor (α) to be applied is given by the following expression:

\[
\text{Equation 2-2} \quad \alpha = \frac{1}{A} \int A (v/V)^3 \, \mathrm{d}A
\]

\[\text{where}\]
\[V = \text{Average velocity in the cross-section, feet per second (fps)}\]
\[v = \text{Velocity at any point in the cross-section, fps}\]
\[A = \text{Area of the cross-section, ft}^2\]

Studies indicate that α equals 1 for uniform distribution of velocity, α equals 1.02 to 1.15 for turbulent flows, and α equals 2 for laminar flow. In most fluid mechanics computations, α is taken as 1, without serious error being introduced into the result, since the velocity head is generally a small percentage of the total head (energy).

If the two velocity heads are unknown, relate them to each other by means of the equation of continuity:
Equation 2-3

\[ Q = A_1 V_1 = A_2 V_2 = \text{constant} \]

where

\[ Q = \text{Rate of flow, gallons per minute (gpm)} \]

NET POSITIVE SUCTION HEAD

NPSH is the total suction head in feet absolute, determined at the suction nozzle and corrected to the pump datum, less the vapor pressure of the liquid in feet absolute. In other words, it is an analysis of energy conditions at the suction side of a pump to determine if the liquid will vaporize at the lowest pressure point in the pump. It should be stressed that only absolute pressures are used in all calculations to determine NPSH. To convert gauge pressure (psig or psi) to feet absolute, add the barometric pressure (14.7 psi at sea level) to the liquid psi to obtain psi absolute and then multiply by 2.31.

The vapor pressure is a unique characteristic of every fluid and increases with increasing temperature (see Figure 2-1 for the vapor pressure of water). When the vapor pressure of a fluid reaches the pressure of the surrounding medium, the fluid begins to vaporize. The temperature at which this vaporization occurs decreases as the pressure of the surrounding medium decreases.

For a fluid to be effectively pumped, it must be kept in a fluid state. Required NPSH is a measure of the suction head required to prevent vaporization at the lowest pressure point in the pump, and NPSH available is a measure of the actual suction pressure provided by the system.

NPSH Required

NPSHR is a function of pump design. As the liquid flows from the pump suction to the eye of the impeller or vane, velocity increases and pressure decreases. Additional pressure losses occur due to shock and turbulence as the liquid strikes the impeller or vane. As the impeller or gear rotates, centrifugal force increases the velocity with a further decrease in the liquid pressure. The NPSH required is the positive head, in feet absolute, required at the pump suction to overcome these pressure drops in the pump and keep the pressure of the liquid above its vapor pressure. NPSHR varies with pump design, pump size, and operating conditions and is supplied by the pump manufacturer.

In a centrifugal pump, NPSHR is the amount of energy (in feet of liquid) required to:
- Overcome friction losses from the suction opening to the impeller vanes.
- Create the desired velocity of flow into the vanes.

In a rotary pump, NPSHR is the amount of energy (in psi) required to:
- Overcome friction losses from the suction opening into the gears or vanes.
- Create the desired velocity of flow into the gears or vanes.

Available NPSH

NPSHA is a characteristic of the system and is defined as the energy in a liquid at the suction connection of the pump (regardless of the pump type) over and above that energy in the liquid due to its vapor pressure. In other words, it is the excess pressure of the liquid, in feet absolute, over its vapor pressure at the pump suction. Figure 2-2 illustrates four typical suction conditions and the applicable NPSHA formula for each.

Since a liquid may have three types of energy, and since NPSH is an energy term, the two methods of determining available NPSH should take potential, pressure, and kinetic energy into account.

To determine available NPSH by calculating system head, consider the energy at station 1 of Figure 2-2(a):

\[ L_H + P_B + \frac{V_i^2}{2g} \]

This is the sum of the potential, pressure, and kinetic energies at the liquid surface. Since the surface of the liquid supply is large compared to the area of the suction pipe, the velocity head is negligible, and kinetic energy or velocity head is zero. The total energy at station 1 is:

\[ L_H + P_B \]
P<sub>b</sub> represents the pressure energy at station 1, or atmospheric pressure. To ensure the liquid does not vaporize in the suction line, subtract the vapor pressure (V<sub>p</sub>) of the liquid from the pressure energy at station 1.

\[ L_h + P_b - V_p \]

The pressure terms are expressed in psi absolute and converted to feet of head, the unit commonly used to express available NPSH:

\[ L_h + (P_b - V_p) \times 2.31 \]

It is equally important to correct for the specific gravity of the liquid if it is not water. The above equation can be rewritten as:

\[ L_h + \frac{(P_b - V_p)}{\text{Specific gravity}} \times 2.31 \]

This is because the energy at station 2 (the point at which available NPSH is required), is equal to the energy at station 1 with the exception of losses due to friction. Available NPSH after subtracting these losses (h<sub>f</sub>) at station 2 becomes:

**Equation 2-4**

\[ L_h + \frac{(P_b - V_p)}{\text{Specific gravity}} \times 2.31 - h_f \]

**Example 2-1**

A cold water system at 600°F has a vapor pressure of 0.256 psia and a specific gravity of 1. The static suction head is 10 feet, and the pipe friction loss is 8 feet. What is the available NPSH for this system?

\[
\text{NPSHA} = 10 + \frac{(14.7 - 0.256)}{1} \times 2.31 - 8 = 35.4 \text{ ft}
\]

**Example 2-2**

A hot water system at 2,000°F has a vapor pressure of 11.53 psia and a specific gravity of 0.965. The static suction head is 10 feet, and the pipe friction loss is 8 feet. What is the available NPSH for this system?

\[
\text{NPSHA} = 10 + \frac{(14.7 - 11.53)}{0.965} \times 2.31 - 8 = 9.85 \text{ ft}
\]

It is sometimes possible to determine available NPSH if test readings are available.

Since station 2 is at the datum, the liquid has no potential energy, and L<sub>h</sub> equals zero. P<sub>g</sub> equals gauge reading. By adding atmospheric pressure to the gauge reading to obtain absolute pressure head, subtracting vapor pressure, and correcting for the elevation of the suction gauge (Y), available NPSH can be obtained:

**Equation 2-5**

\[
\frac{(P_g + P_b - V_p)}{\text{Specific gravity}} \times 2.31 + Y + \frac{V_2^2}{2g}
\]

where

- V<sub>2</sub> = Velocity of the fluid, feet per second (fps)

All of the prior information is applicable when a single pump is used in a system. However, it is generally desirable to use two or more pumps in parallel, rather than a single larger pump. This is particularly advantageous when the system demand varies greatly and repairs or maintenance can be performed easily on one unit without completely shutting down the entire system.
Pump Cavitation
When the pressure in the suction line falls below the vapor pressure, vapor forms and moves along with the stream. These vapor bubbles, or cavities, collapse when they reach regions of higher pressure on their way through the pump. The most obvious effects of cavitation are noise and vibration. These are caused by the collapse of the vapor bubbles as they reach the high-pressure side of the pump. Bigger pumps create greater noise and vibration than smaller pumps. If operated under cavitating conditions for a sufficient length of time, especially on water service, impeller vane pitting will occur. The violent collapse of vapor bubbles forces liquid at high velocity into vapor-filled pores of the metal, producing surge pressures of high intensity on small areas. These pressures can exceed the tensile strength of the metal and actually blast out particles, giving the metal a spongy appearance. This noise and vibration also can cause bearing failure, shaft breakage, and other fatigue failures in the pump.

The other major effect of cavitation is a drop in pump efficiency, apparent as a drop in capacity (see Figure 2-3). The drop in the efficiency and head capacity curve may occur before the vapor pressure is reached, particularly in petroleum oils, because of the liberation of light fractions and dissolved and entrained air. Pitting is not as serious when the pump is handling oils due to the cushioning effect of the more viscous liquid.

In general, cavitation indicates insufficient available NPSH. Excessive suction pipe friction, combined with low static suction head and high temperatures, contributes to this condition. If the system cannot be changed, it may be necessary to change conditions so a different pump with lower NPSH requirements can be used. Larger pumps might require the use of a booster pump to add pressure head to the available NPSH.

HEAD
Head is commonly used to represent the vertical height of a static column of liquid corresponding to the pressure of a fluid at the point in question. Head can also be considered as the amount of work necessary to move a liquid from its origin to the required delivery position. This includes the extra work necessary to overcome the resistance to flow in the line.

In general, a liquid may have three kinds of energy, or the capacity to do work may be due to three factors:
- Potential head (energy of position): The work possible in a dropping vertical distance
- Static pressure head (energy per pound due to pressure): The height a liquid can be raised by a given pressure
- Velocity head (kinetic energy per pound): The vertical distance a liquid would have to fall to acquire a specific velocity.

Static Suction Lift
When the supply source is below the pump, the vertical distance from the free surface of the liquid to the pump datum is called the static suction lift (see Figure 2-4). The net suction head, in this case, is the sum of the static suction lift and friction losses (negative net suction lift).

Static Suction Head
When the supply is above the pump, the vertical distance from the free surface of the liquid to the pump datum is called the static suction head (see Figure 2-5). The net suction head is the static suction head minus friction losses (either positive or negative).

Total Head
The total head developed by the pump can be expressed by one of the following:

Equation 2-6 (Pump with Suction Lift)

\[ H = h_d + h_s + f_d + f_s + \left( \frac{V^2}{2g} \right) \]

Equation 2-7 (Pump with Suction Head)

\[ H = h_d - h_s + f_d + f_s + \left( \frac{V^2}{2g} \right) \]
where

\( H = \) Total head of liquid pumped when operating at the desired capacity, ft

\( h_d = \) Static discharge head equal to the vertical distance between the pump datum and the surface of the liquid in the discharge reservoir, ft

\( h_s = \) Static suction head or lift equal to the vertical distance from the water surface to the pump datum (positive when operating with a suction lift and negative when operating with a suction head), ft

\( f_d = \) Friction head loss in the discharge piping, ft

\( f_s = \) Friction head loss in the suction piping, ft

\( V^2/2g = \) Velocity head, ft

For vertical turbine and submersible pumps, the velocity head is measured at the discharge flange. However, for booster pumps and centrifugal pumps, the velocity head developed by the pump is the difference between the \( V^2/2g \) at the discharge flange and the \( V^2/2g \) at the suction flange. That is:

**Equation 2-8**

\[ V^2/2g = (V_d^2/2g) - (V_s^2/2g) \]

where

\( V_d^2/2g = \) Velocity head at the discharge flange

\( V_s^2/2g = \) Velocity head at the suction flange

Since the discharge flange is usually a size smaller than the suction flange, the difference in the velocity head is always positive. Usually, it is a small percentage of the total head and frequently is erroneously neglected.

**BERNOULLI THEOREM**

The energy equation results from the application of the principle of energy conservation to fluid flow. The energy possessed by a flowing fluid consists of internal energy and energies due to pressure, velocity, and position. In the direction of flow, the energy principle is summarized by a general equation:

**Equation 2-9a**

\[ \text{Energy at section 1} + \text{Energy added} - \text{Energy lost} - \text{Energy extracted} = \text{Energy at section 2} \]

This equation, for the steady flow of incompressible fluids where the change in internal energy is negligible, simplifies to:

**Equation 2-9b**

\[ \left( \frac{p_1}{w} + \frac{V_1^2}{2g} + z_1 \right) + H_A - H_L - H_E = \left( \frac{p_2}{w} + \frac{V_2^2}{2g} + z_2 \right) \]

Energy cannot be created or destroyed. The sum of three types of energy (heads) at any point in a system is the same as in any other point in the system, assuming no friction losses or extra work performed. The above equation, in this case, can be further simplified to:

**Equation 2-10**

\[ \left( \frac{p_1}{w} + \frac{V_1^2}{2g} + z_1 \right) = \left( \frac{p_2}{w} + \frac{V_2^2}{2g} + z_2 \right) \]

This equation is known as the Bernoulli theorem. The units used are foot-pound per pound of fluid or feet of fluid. Almost all problems dealing with liquid flow utilize this equation as the solution basis.

Application of the Bernoulli theorem should be rational and systematic. A suggested procedure follows.

1. Draw a sketch of the system, choosing and labeling all stream cross-sections under consideration.
2. Apply the Bernoulli equation in the direction of flow. Select a datum plane for each equation. The low point is logical, as minus signs are avoided and mistakes reduced.
3. Evaluate the energy upstream at section 1. The energy is in foot-pound per pound units, which reduce to feet of fluid units. For liquids, the pressure head may be expressed in gauge or absolute units, but the same basis must be used for the pressure head at section 2. Gauge units are simpler to use for liquids. Absolute pressure head units must be used where specific weight \( (w) \) is not constant. As in the equation of continuity, \( V_1 \) is taken as the average velocity at the section, without loss of acceptable accuracy.
4. Add, in feet of fluid, any energy contributed by mechanical devices such as pumps.
5. Subtract, in feet of fluid, any energy lost during flow.
6. Subtract, in feet of fluid, any energy extracted by mechanical devices such as turbines.
7. Equate this energy summation to the sum of pressure head, velocity head, and elevation head at section 2.
8. If the two velocity heads are unknown, relate them to each other by means of the equation of continuity.

**Example 2-3**

Water flows through the turbine in Figure 2-6 at the rate of 7.55 cubic feet per second (cfs). The pressures at A and B, respectively, are 21.4 psi and -5 psi. Determine the horsepower delivered to the turbine by the water.

Step 1. Find the velocities at A and B respectively.

\[ V_{12} = \frac{7.55}{A_{12}} = 9.6 \text{ and } V_{24} = \frac{9.60}{4} = 2.4 \text{ fps} \]

Step 2. Set up the energy equation.

\[
\begin{align*}
(P_a/w + V_{12}^2/2g + z_a) + 0 - H_{turbine} &= (P_b/w + V_{24}^2/2g + z_b) \\
\frac{21.4(144)}{62.4} + \frac{9.6^2}{2g} + 3 - H_t &= \frac{-5(144)}{62.4} + \frac{2.4^2}{2g} + 0
\end{align*}
\]

\[ H_t = 65.4 \text{ ft} \]

Step 3. Determine the horsepower delivered to the turbine by the water:

\[ hp = wQH_t/550 = 62.4(7.55)(65.4)/550 = 56 \]

**FRICTION HEAD**

Friction head is the head lost in overcoming pipe friction. It depends on pipe size, smoothness of the inside surface, the number and type of fittings, orifice plates and control valves, flow velocity, and liquid viscosity and density. The following is a list of principal, semi-theoretical flow equations for estimating frictional head losses.

**Darcy-Weisbach Equation**

The Darcy-Weisbach equation can be used to estimate friction head losses:

**Equation 2-11**

\[ H_l = f \left( \frac{L V^2}{2gd} \right) = f \left( \frac{L V^2}{8gR^2} \right) = \frac{\Delta P}{\rho g} \]

where

- \( H_l \) = Friction head loss, ft
- \( f \) = Friction factor
- \( L \) = Length of pipe
- \( d \) = Inside diameter, ft
- \( R \) = Hydraulic radius, ft
- \( V \) = Fluid velocity, fps
- \( g \) = Acceleration due to gravity (32.2 ft/s²)
- \( \rho \) = Fluid density

**Hazen-Williams Equation**

For fluid velocity, the Hazen-Williams equation is used:

**Equation 2-12a**

\[ V = 1.318 C R^{0.63} S^{0.54} \]

where

- \( C \) = Coefficient of roughness
- \( S \) = Slope of the energy line \( (H_l/L) \)
- \( V \) = Fluid velocity, fps

A more useful form of the equation is:

**Equation 2-12b**

\[ H_l = 0.2083 \left( \frac{100/C}{d^{1.85}} \right) q^{1.85}/d^{4.8655} \]

where

- \( H_l \) = Friction head loss, feet per 100 feet of pipe
- \( d \) = Inside diameter of pipe, in.
- \( q \) = Quantity of flow, gpm
- \( C \) = Dimensionless constant reflecting the roughness of the pipe
Manning Equation
The Manning equation is an empirical expression originally developed for open-channel flows. However, it can be applied to full-flow pipe:

**Equation 2-13**

\[ V = \left( \frac{1.486}{\beta} \right) \frac{R^{2/3}}{s^{1/6}} \]

or

\[ H_L = \left( \frac{4.6666}{\beta^2} \frac{L}{D^{5.33}} \right) Q^2 \]

Friction Factor
The friction factor (f) can be derived mathematically for laminar flow, but no simple mathematical relation for the variation of f with the Reynolds number (RE) is available for turbulent flow. Furthermore, Nikuradse and others found that the relative roughness of the pipe (the ratio of the size of surface imperfections to the pipe’s inside diameter) affects the value of f also.

For laminar flow, the following equation can be used:

**Equation 2-14**

\[ \text{Lost head} = 64 \frac{L V^2}{V \, \frac{d}{d} \frac{2}{2g}} = \frac{64 \, L \, V^2}{\text{RE} \, \frac{d}{2g}} \]

Thus, for laminar flow in all pipes for all fluids, the value of f is:

**Equation 2-15**

\[ f = \frac{64}{\text{RE}} \]

RE has a practical maximum value of 2,000 for laminar flow.

For turbulent flow, many hydraulicians have endeavored to evaluate f from the results of their own, and other’s, experiments. For turbulent flow in smooth and rough pipes, universal resistance laws can be derived from:

**Equation 2-16**

\[ f = \frac{8 \tau_o}{(\rho V^2)} = \frac{8V^2}{V^2} \]

where

V = Shear velocity

For smooth pipes with Reynolds numbers between 3,000 and 100,000, Blasius suggests:

**Equation 2-17**

\[ f = 0.316/\text{RE}^{0.25} \]

For values of RE up to about 3,000,000, Von Karman’s equation modified by Prandtl is:

**Equation 2-18**

\[ 1/\sqrt{f} = 2 \log (\text{RE} \sqrt{f}) - 0.8 \]

For rough pipes:

**Equation 2-19**

\[ 1/\sqrt{f} = 2 \log (r_o / \varepsilon) + 1.74 \]

where

\[ \varepsilon = \text{Roughness} \]

For all pipes, the most up-to-date correlation of these factors is expressed in the Colebrook equation:

**Equation 2-20**

\[ \frac{1}{\sqrt{f}} = -2 \log \left( \frac{k}{3.7D} + \frac{2.51}{\text{RE} \sqrt{f}} \right) \]

where

\[ k = \text{Relative roughness} \]

This equation is cumbersome and contains too many variables to be of practical use in the above format. However, friction factor diagrams for pipe flow can be found in many resources, including the Plumbing Engineering Design Handbook published by ASPE.
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Expiration date: Continuing education credit will be given for this examination through August 31, 2016.

CE Questions — “Pressure and Flow Characteristics” (CEU 226)

1. _______ is the difference between a given pressure and that of the atmosphere.
   a. vacuum
   b. gauge pressure
   c. absolute pressure
   d. vapor pressure

2. _______ represents the kinetic energy per unit weight that exists at a particular point.
   a. velocity head
   b. static head
   c. flow volume
   d. net positive suction head

3. Only _______ pressures should be used in all calculations to determine net positive suction head.
   a. atmospheric
   b. gauge
   c. absolute
   d. vapor

4. _______ is the excess pressure of a liquid over its vapor pressure at the pump suction.
   a. NPSHR
   b. NPSHA
   c. absolute pressure
   d. gauge pressure

5. Which of the following is an effect of cavitation?
   a. noise
   b. decreased pump efficiency
   c. vibration
   d. all of the above

6. _______ is the height a liquid can be raised by a given pressure.
   a. potential head
   b. static pressure head
   c. static suction head
   d. total head

7. When the supply source is below the pump, the vertical distance from the free surface of the liquid to the pump datum is the _______.
   a. static pressure head
   b. static suction head
   c. static suction lift
   d. none of the above

8. Almost all problems dealing with fluid flow utilize the _______ as the solution basis.
   a. Manning equation
   b. Hazen-Williams equation
   c. Darcy-Weisbach equation
   d. Bernoulli theorem

9. Friction head depends on _______.
   a. liquid viscosity
   b. smoothness of pipe interior
   c. flow velocity
   d. all of the above

10. Which of the following can be used to estimate friction head losses?
    a. Manning equation
    b. Colebrook equation
    c. Darcy-Weisbach equation
    d. Bernoulli theorem

11. _______ is used to calculate fluid velocity.
    a. Colebrook equation
    b. Hazen-Williams equation
    c. Darcy-Weisbach equation
    d. Bernoulli theorem

12. For laminar flow, the Reynolds number has a practical maximum value of _______.
    a. 2,000
    b. 2,500
    c. 3,000
    d. 10,000