10

Pipeline System Design

10.1 Pipeline Standards

Standards are established for pipeline materials so engineers can select the proper material for the required application and have confidence in the ability of the pipeline to perform as expected. Standards are established which relate the pipeline dimensions and type of material to its strength and ability to resist loads. The standards described in this section are taken from those developed by the American Society of Agricultural Engineers (ASAE) for thermoplastic pipe (Standard ASAE S376.1). Common thermoplastic pipe materials are polyvinyl chloride (PVC) and polyethylene (PE). These materials are used in a wide variety of applications in irrigation systems in the United States and other countries. Because of ease of fabrication, locally produced PVC pipe is employed in many irrigation development projects worldwide. The standards described in this section apply specifically to thermoplastic pipe. Other types of pipe materials have similar standards which are found in other references (e.g., ASAE Standards, 1987; Stephenson, 1981). Standards for aluminum tubing are found in Appendix D.

Pressure Category

Thermoplastic pipes are divided into low and high pressure categories. The pressure categories are based on both the pipe diameter and the design operating pressure.

- Low Pressure
  Nominal diameter—114 to 630 mm (4 to 24 inch)
  Internal pressure—545 kPa (79 psi) or less
- High Pressure
  Nominal diameter—21 to 710 mm (0.5 to 27 inch)
  Internal pressure—550 to 2170 kPa (80 to 315 psi) including surge pressure

Surge pressure occurs in unsteady flow regimes and is associated with rapid changes in flow velocity and resulting rapid changes in pressure. It is commonly referred to as water hammer and is covered later in this chapter.

Pressure Rating and Hydrostatic Design Stress

The pressure rating and hydrostatic design stress both refer to parameters related to long-term operation of the pipeline. The pressure rating is the maximum pressure that water in the pipe can exert continuously with a high degree of certainty that failure will not occur. The hydrostatic design stress is the maximum tensile stress due to an internal hydrostatic pressure that can be applied continuously with a high degree of certainty that failure will not occur.

These two parameters are related to the dimension ratio, DR, which is given by

$$DR = \frac{D}{t} \tag{10-1}$$

where

D = outside or inside pipe diameter depending on how the pipe size is controlled, mm

t = wall thickness, mm

The dimension ratio is dimensionless. The minimum wall thickness for thermoplastic pipe is specified as 1.52 mm (0.060 inch). Certain dimension ratios have been selected as standard and are designated in tabulated data as Standard Dimension Ratios.

For outside diameter based pipe, the pressure rating is given as

$$PR = \frac{2S}{DR - 1} \tag{10-2}$$

or

$$PR = \frac{2S}{\frac{D_0}{t} - 1} \tag{10-3}$$

where

PR = pressure rating, kPa (psi)

S = hydrostatic design stress, kPa (psi)

D_0 = average outside diameter, mm (in.)

t = minimum wall thickness, mm (in.)

For inside diameter based pipe,

$$PR = \frac{2S}{DR + 1} \tag{10-4}$$

or

$$PR = \frac{2S}{\dfrac{D_i}{t} + 1} \tag{10-5}$$

where

$$D_i = \text{average inside diameter, mm (in.)}$$

Hydrostatic design stresses for different thermoplastic pipe material and different strengths of pipe are given in Table 10-1. ABS refers to pipes fabricated from acrylonitrile-butadiene-styrene. The hydrostatic design stress is derived from the long-term hydrostatic strength which is determined by standardized tests to failure of the pipe material. The relationship is given by

$$S = \frac{S_{lt}}{2.0} \tag{10-6}$$

where

$$S_{lt} = \text{long-term hydrostatic strength}$$

and the 2.0 represents a factor of safety. Table 10-2 indicates pressure ratings for different strengths of PVC, PE, and ABS materials as a function of Standard Dimension Ratios.

TABLE 10-1 Maximum hydrostatic design stress for thermoplastic pipe. (Adapted from ASAE Standard S376.1.)

Compound	Standard Code Designation	Hydrostatic Design Stress (MPa)	Hydrostatic Design Stress (psi)
PVC	PVC 1120	13.8	2000
PVC	PVC 1220	13.8	2000
PVC	PVC 2120	13.8	2000
PVC	PVC 2116	11.0	1600
PVC	PVC 2116	8.6	1250
PVC	PVC 2110	6.9	1000
PE	PE 3408	5.5	800
PE	PE 3406	4.3	630
PE	PE 3306	4.3	630
PE	PE 2306	4.3	630
PE	PE 2305	3.4	500
ABS	ABS 1316	11.0	1600
ABS	ABS 2112	8.6	1250
ABS	ABS 1210	6.9	1000

TABLE 10-2 Pressure ratings (PR) for nonthreaded thermoplastic pipe.* (Taken from ASAE Standard S376.1.)

SDR[‡]		PVC materials (all pipes OD based)								PE materials (pipes made to both OD & ID basis)						ABS materials (all pipes OD based)					
		PVC 1120 PVC 1220 PVC 2120		PVC 2116		PVC 2112		PVC 2110		PE 3408		PE 3406 PE 3306 PE 2306		PE 2305		ABS 1316		ABS 2112		ABS 1210	
OD based pipe	ID based pipe	psi	kPa[§]	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa
	5.3									250	1725	200	1380	160	1105						
	7.0									200	1380	160	1105	125	860						
11.0	9.0									160	1105	125	860	100	690						
13.5	11.5	315	2170	250	1725	200	1380	160	1105							250	1725	200	1380	160	1105
17.0	15.0	250	1725	200	1380	160	1105	125	860	100	690	80	550	63	435	200	1380	160	1105	125	860
21.0		200	1380	160	1105	125	860	100	690	80	550	64	440			160	1105	125	860	100	690
26.0		160	1105	125	860	100	690	80	550	64	440	50	345			125	860	100	690	80	550
32.5		125	860	100	690	80	550	63	435	50	345	40	275			100	690	80	550	64	440
41.0		100	690	80	550	63	435	50	345	40	275	31	215			80	550	64	440	50	345
51.0		80	550	63	435	50	345	40	275							64	440	50	345	40	275
64.0		63	435	50	345	40	275	30	205												
81.0		50	345	40	275	30	205	25	170							40	275	30	205	25	170
93.5[∥]		43	295																		
50 ft head		22	150																		

*For water at 23°C (73.4°F).

‡SDR = Standard Dimension Ratio

§kPa = kilopascals, kN/m^2

∥The dimension ratio 93.5 is nonstandard and is referred to as DR (Dimension Ratio)

Example Problem 10-1

A PVC pipe is to be manufactured from PVC 2116 compound. The pipe size is based on its inside diameter which is 25 cm. Two categories of the pipe will be manufactured: (a) low pressure with a wall thickness of 1.8 mm and (b) high pressure with a wall thickness of 9.6 mm. Compute the pressure rating for each category of pipe.

Solution For ID based pipe,

$$PR = \frac{2S}{\frac{D_i}{t} + 1}$$

Case (a) low strength:

$$\frac{D_i}{t} = \frac{250 \text{ mm}}{1.8 \text{ mm}} = 139$$

Case (b) high strength:

$$\frac{D_i}{t} = \frac{250 \text{ mm}}{9.6 \text{ mm}} = 26$$

From Table 10-1 for PVC 2116,

$$S = 11.0 \text{ MPa} = 11{,}000 \text{ kPa}$$

Pressure rating:

Case (a) low strength:

$$PR = \frac{2(11{,}000 \text{ kPa})}{139 + 1} = 157 \text{ kPa } (\cong 23 \text{ psi})$$

Case (b) high strength:

$$PR = \frac{2(11{,}000 \text{ kPa})}{26 + 1} = 815 \text{ kPa } (\cong 117 \text{ psi})$$

10.2 Pressure Distribution in Pipelines

By application of Bernoulli's law, the total head available at any point in a pipeline is equal to the pressure head plus elevation head plus velocity head. The difference in total head between any two points 1 and 2 on a pipeline under steady-state flow conditions is equal to the friction headloss between the two points so that

$$H_{T2} = \left(\frac{p}{\gamma}\right)_1 + z_1 + \left(\frac{v^2}{2\,g}\right)_1 - (h_f)_{1-2} \tag{10-7}$$

where

$$H_T = \text{total head, m}$$

$$\frac{p}{\gamma} = \text{pressure head, m}$$

$$\gamma = \text{specific weight of the fluid, kN/m}^3$$

$$z = \text{elevation head, m}$$

$$h_f = \text{friction headloss, m}$$

The specific weight of the fluid is given by

$$\gamma = Sg(\gamma_w) \tag{10-8}$$

where

$$Sg = \text{specific gravity of the fluid, dimensionless}$$

$$\gamma_w = \text{specific weight of water, kN/m}^3$$

The specific weight of water under standard conditions applicable to design of irrigation systems is 9.81 kN/m^3. The units required for pressure in Eq. (10-7) are kPa (equal to kN/m^2) to enable the pressure head to be expressed in meters.

Figure 10-1 is a schematic of the distribution of different forms of energy given as head in a pipeline. The static head line refers to the distribution of head if

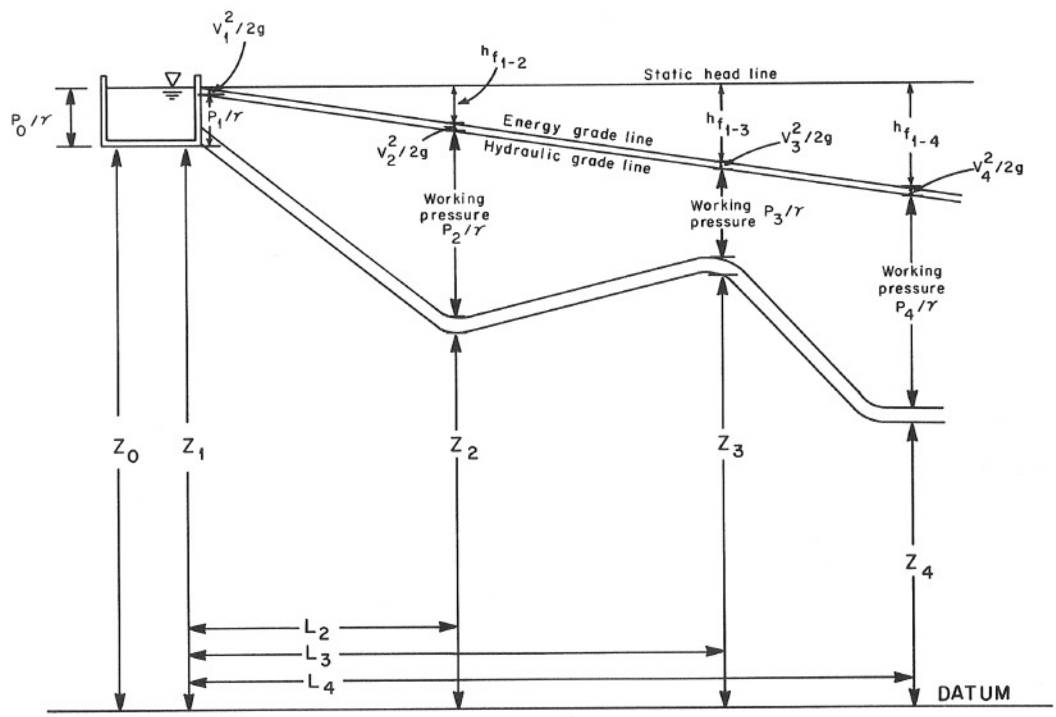

Figure 10-1 Distribution of heads in a pipeline system.

there is no flow in the system. The energy grade line describes the variation of total head along the pipeline, including velocity head, when flow occurs. The hydraulic grade line is defined by the energy grade line minus the velocity head. It indicates the height to which water would rise in a piezometer inserted at that position in the pipeline. The hydraulic grade line defines the working pressure available at any point along the pipeline. The pressure distribution in a pipeline is determined by the relative positions of water sources along the line and friction headloss. The following sections consider two methods of computing friction headloss.

Hazen-Williams Friction Headloss Equation

There are numerous methods for computing friction headloss in pipelines. One of the most common and convenient methods applicable to pumping water through irrigation systems is the Hazen-Williams equation given as

$$h_f = k_1 \, L \frac{\left(\frac{Q}{C}\right)^{1.852}}{D^{4.87}} \tag{10-9}$$

where

$$\begin{aligned} k_1 &= \text{conversion constant} \\ L &= \text{length of pipe, L} \\ Q &= \text{volumetric flow rate, } L^3/T \\ C &= \text{Hazen-Williams coefficient} \\ D &= \text{pipe diameter, L} \end{aligned}$$

Table 10-3 indicates common units associated with flow in pipes in the SI and English systems and the required conversion constant k_1 for Eq. (10-9). Table 10-4 lists the Hazen-Williams C coefficient for various types of pipe materials. The Hazen-Williams equation is only applicable to water at standard operating temperature (i.e., 20°C), or more specifically to fluids with a specific gravity of 1.0. Such an assumption is almost always valid for analysis of flow in irrigation systems. Example problem 10-2 indicates application of the Hazen-Williams equation to calculation of pressure distribution in a pipeline. For fluids with different specific gravities, and therefore different viscosities, a friction factor which accounts for the effect of fluid viscosity must be applied. Such a method is described following the example problem.

TABLE 10-3 Conversion constants for the Hazen-Williams equation given different combinations of units.

h_f	L	Q	D	k_1
m	m	L/s	mm	1.22×10^{10}
m	m	L/h	mm	3163
m	m	m^3/d	mm	3.162×10^6
ft	ft	ft^3/s	ft	4.73
ft	ft	gpm	in	10.46

TABLE 10-4 Friction factor C for Hazen-Williams equation.

Pipe Material	Values of C		
	Design	New Pipe	Corroded Pipe
Polyethylene (PE) and polyvinyl chloride (PVC)	140	150	130
Cement–Asbestos	140	150	140
Fiber	140	150	—
Bitumastic-enamel-lined iron or steel centrifugally applied	140	148	130
Cement-lined iron or steel centrifugally applied	140	150	—
Copper, brass, lead, tin, or glass pipe and tubing	130	140	120
Wood-stave	110	120	110
Welded and seamless steel	100	130	80
Interior riveted steel (no projecting rivets)	100	139	—
Wrought-iron, cast-iron	100	130	80
Tar-coated cast iron	100	130	50
Girth-riveted steel (projecting rivets in girth seams only)	100	130	—
Concrete	100	120	85
Full-riveted steel (projecting rivets in girth and horizontal seams)	100	115	—
Vitrified, spiral-riveted steel (flow with lap)	100	110	—
Spiral-riveted steel (flow against lap)	90	100	—
Corrugated steel	60	60	—

Example Problem 10-2

A pipeline with 200 mm inside diameter and 340 m in length made of new PVC is laid along a horizontal grade. The required flow rate in the pipeline at steady state is 40.0 L/s and the total head available at the inlet is 330 kPa. Compute the working pressure head at the discharge 340 m from the inlet by application of the Hazen-Williams friction headloss equation.

Solution From Eq. (10-9),

$$h_f = k_1 L \frac{\left(\frac{Q}{C}\right)^{1.852}}{D^{4.87}}$$

From Table 10-3 for L in m, Q in L/s, and D in mm,

$$k_1 = 1.22 \times 10^{10}$$

From Table 10-4 for new PVC,

$$C = 150$$

$$h_f = 1.22 \times 10^{10}(340 \text{ m})\frac{[(40.0 \text{ L/s})/150]^{1.852}}{(200 \text{ mm})^{4.87}}$$

$$h_f = 2.232 \text{ m}$$

To convert to kPa,

$$h_f' = h_f\gamma = 2.232 \text{ m}(9.81 \text{ kN/m}^3)$$

$$h_f' = 21.897 \text{ kN/m}^2 = 21.897 \text{ kPa}$$

$$H_{T\text{-}340} = 330 \text{ kPa} - 22 \text{ kPa}$$

$$= 308 \text{ kPa}$$

Darcy-Weisbach Friction Headloss Equation

When the fluid to be pumped is other than water or the specific weight is substantially different from water due to additions of chemicals to the flow or temperature effects, an equation which accounts for differences in viscosity must be applied to calculate the friction headloss. A common equation applied is that of Darcy-Weisbach which is given in its most basic form as

$$h_f = f\frac{L}{D}\left(\frac{v^2}{2g}\right) \tag{10-10}$$

where

$$f = \text{Darcy-Weisbach friction factor, dimensionless}$$

and the units for the friction headloss are that of the velocity head if L and D have the same units. Revising this equation to a form conveniently applicable to pipe flow

$$h_f = k_2 f L \frac{Q^2}{D^5} \tag{10-11}$$

where

$$k_2 = \text{conversion constant}$$

The conversion constant k_2 is given for various combinations of SI and English units common to pipe flow in Table 10-5.

The friction factor is a function of the flow regime—that is, whether the flow is laminar, turbulent, or in transition between the two, and the roughness of the pipe material. The flow regime is designated as a function of the dimensionless Reynold's number, R_N, which is defined by the following equation:

$$R_N = \frac{v\,D}{1000\,\nu} \tag{10-12}$$

where

$$v = \text{flow velocity, m/s}$$
$$D = \text{pipe diameter, mm}$$
$$\nu = \text{kinematic viscosity of the fluid, m}^2\text{/s}$$

TABLE 10-5 Conversion constants for the Darcy-Weisbach equation given different combinations of units.

h_f	L	Q	D	k_2
m	m	L/s	mm	8.2627×10^7
m	m	L/h	mm	6.3755
m	m	m^3/d	mm	1.10686×10^4
ft	ft	ft^3/s	ft	0.02517
ft	ft	gpm	in	0.03107

The kinematic viscosity of water under standard operating conditions for irrigation systems (i.e., 20°C) is 1.0×10^{-6} m^2/s.

For $R_N \leq 2000$, flow is laminar and the Darcy-Weisbach friction factor is given as

$$f = \frac{64}{R_N} \tag{10-13}$$

In the range $2000 < R_N < 4000$, the flow is in the critical state and it cannot be said precisely whether it is laminar or turbulent. In typical engineering applications, flow tends to be either laminar or fully turbulent so the critical range is not of concern (Mott, 1979). For $R_N > 4000$ flow is said to be turbulent. However there are two zones of turbulence as indicated in the Moody diagram for the Darcy-Weisbach f versus R_N shown in Fig. 10-2. To the right of the transition zone line indicated in the figure, the flow is fully turbulent and the friction factor is independent of R_N and depends only on the roughness of the pipe material. This roughness is described by the relative roughness which is expressed as a dimensionless ratio of wall roughness to pipe diameter, ε/D. For flow in the completely turbulent zone, the Darcy-Weisbach f is given by the expression

$$\frac{1}{(f)^{1/2}} = 2 \log\left(3.7\frac{D}{\varepsilon}\right) \tag{10-14}$$

From the left of the transition line in Fig. 10-2 to the lower limit of $R_N = 4000$, the flow is in transition and not yet fully turbulent. In this zone the friction factor is a function of both the Reynold's number and the relative roughness. The expression for the Darcy-Weisbach f in this zone developed by Colebrook (1939) is given as

$$\frac{1}{(f)^{1/2}} = -2 \log\left\{\frac{(\varepsilon/D)}{3.7} + \frac{2.51}{R_N(f)^{1/2}}\right\} \tag{10-15}$$

Equation (10-15) requires an iterative solution since f appears on both sides of the equation. However a method has been developed to simplify this procedure which gives an initial estimate of f as a function of the relative roughness and R_N (Murdock, 1976). Labelling the first estimate as f_1, the function is

$$f_1 = 0.0055\left\{1 + \left[\frac{20000}{(D/\varepsilon)} + \frac{10^6}{R_N}\right]^{1/3}\right\} \tag{10-16}$$

This initial estimate is substituted into the right-hand side of Eq. (10-15) to solve for the first iteration of f. The value of f_1 computed in Eq. (10-16) is close enough to the final value of f that a second iteration is normally not required (Mott, 1979). Table 10-6 indicates values of surface roughness for various types of pipeline materials.

The lower boundary of the turbulent zone indicated in Fig. 10-2 is that for smooth pipes. Pipes made of plastic and glass normally fall into this range as do

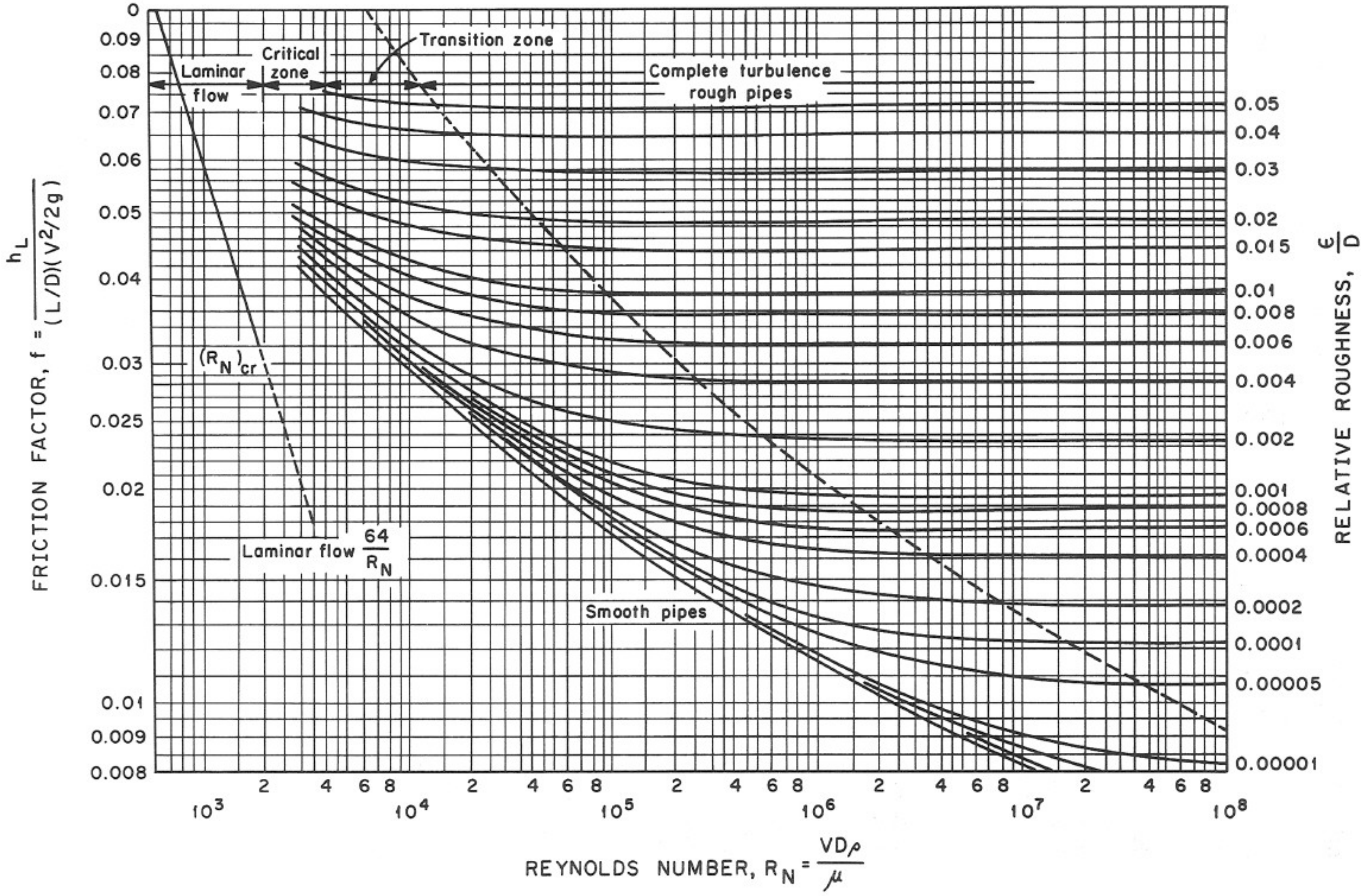

Figure 10-2 Moody diagram for friction loss in pipes and tubing. (Adapted from Moody, 1944.)

TABLE 10-6 Surface roughness of pipe materials. (Adapted from Stephenson, 1981.)

Pipe Material	ε(mm) Smooth	Average	Rough
Glass, drawn metals	Smooth	0.003	0.006
Steel or polyvinyl chloride	0.015	0.03	0.06
Coated steel	0.03	0.06	0.15
Galvanized, vitrified clay	0.06	0.15	0.3
Cast iron or cement lined	0.15	0.3	0.6
Spun concrete or wood stave	0.3	0.6	1.5
Riveted steel	1.5	3	6
Foul sewers	6	15	30
Unlined rock, earth	60	150	300

large diameter pipes made of drawn brass or copper. The equation for the Darcy-Weisbach f as defined by the smooth pipe line is given by

$$\frac{1}{(f)^{1/2}} = 2\log\left[\frac{R_N(f)^{1/2}}{2.51}\right] \tag{10-17}$$

This equation also requires an iterative solution. The following example problem indicates the Darcy-Weisbach solution to the same situation given in example problem 10-3.

Example Problem 10-3

A pipeline with 200 mm inside diameter and 340 m in length made of new PVC is laid along a horizontal grade. The required flow rate in the pipeline at steady state is 40.0 L/s and the total head available at the inlet is 330 kPa. Compute the working pressure head at the discharge 340 m from the inlet by application of the Darcy-Weisbach headloss equation.

Solution From Eq. (10-11),

$$h_f = k_2 f L \frac{Q^2}{D^5}$$

From Table 10-5 for L in m, Q in L/s and D in mm,

$$k_2 = 8.263 \times 10^7$$

From Table 10-6 for smooth PVC,

$$\varepsilon = 0.015 \text{ mm}$$

Solving for flow velocity,

$$v = \frac{Q}{A} = 40.0 \text{ L/s}\left(\frac{1 \text{ m}^3}{1000 \text{ L}}\right) \Big/ \left\{\pi\left[\frac{(200 \text{ mm})^2}{4}\right]\left[\frac{1 \text{ m}}{1000 \text{ mm}}\right]^2\right\}$$

$$v = 1.273 \text{ m/s}$$

Solving for Reynold's number,

$$R_N = \frac{v\,D}{1000\,\nu}$$

$$R_N = 1.273\text{ m/s}\left[\frac{200\text{ mm}}{1000(1 \times 10^{-6}\text{ m}^2/\text{s})}\right]$$

$$R_N = 2.55 \times 10^5$$

Flow is fully turbulent and f is function of ε/D only. Using Eq. (10-14).

$$\frac{1}{\sqrt{f}} = 2\log\left[3.7\left(\frac{200\text{ mm}}{0.015\text{ mm}}\right)\right]$$

$$\frac{1}{\sqrt{f}} = 9.386$$

$$f = 0.0114$$

Substituting into Eq. (10-11),

$$h_f = 8.263 \times 10^7(0.0114)\,(340\text{ m})\,\frac{(40.0\text{ L/s})^2}{(200\text{ mm})^5}$$

$$h_f = 1.594\text{ m}$$

Converting to kPa,

$$h_f' = h_f\gamma = 1.594\text{ m}(9.81\text{ kN/m}^3)$$

$$h_f' = 15.64\text{ kPa}$$

$$H_{T-340} = 330\text{ kPa} - 16\text{ kPa}$$

$$= 314\text{ kPa}$$

It should be noted that the Darcy-Weisbach equation for headloss can be applied for fluids of different viscosities in contrast to the Hazen-Williams equation which is only valid for water. This is because fluid viscosity is required for the Reynold's number which is in turn required to compute the Darcy-Weisbach friction factor for all cases other than fully turbulent. However, for the wide majority of applications in irrigation system design, the Hazen-Williams equation is sufficient and will normally be applied in this book.

10.3 Unsteady Flow in Pipelines

Concept of Water Hammer

Water hammer, or a pressure surge, is caused in pipelines as a result of changes in fluid flow velocities. Changes in the flow velocity essentially cause the kinetic energy associated with velocity to be converted to pressure. This condition in a pipeline is termed one of hydraulic transients. The more rapid the change in flow velocity, the greater will be the magnitude of the resulting pressure surge. Pressure surges occur when valves are opened or closed, when pumps are started or stopped, or by sudden releases of entrapped air.

The water hammer phenomenon will be described by reference to Fig. 10-3. The figure illustrates a pipeline fed by a reservoir with constant head H. A valve is located along the pipeline at distance L from the reservoir. The energy grade line and hydraulic grade line are indicated in the figure and the difference between them is the velocity head. Initial conditions are a steady-state flow at velocity v_0.

At time $t = 0$ the valve is suddenly closed and the fluid behind the valve becomes compressed and causes the pipe to expand. The high pressure in the compressed fluid layer adjacent to the valve is transferred to the next fluid layer upstream resulting in a pressure wave which moves up the pipe. On the reservoir side of the pressure wave, the velocity is the initial v_0. On the valve side of the pressure wave, the velocity is zero. This is the case illustrated in part (b) of Fig. 10-3. The change in head caused by the stopped flow is indicated as ΔH. The pressure wave moves up the pipeline at velocity a. At $t = L/a$ the pressure wave has moved all the way upstream to the reservoir.

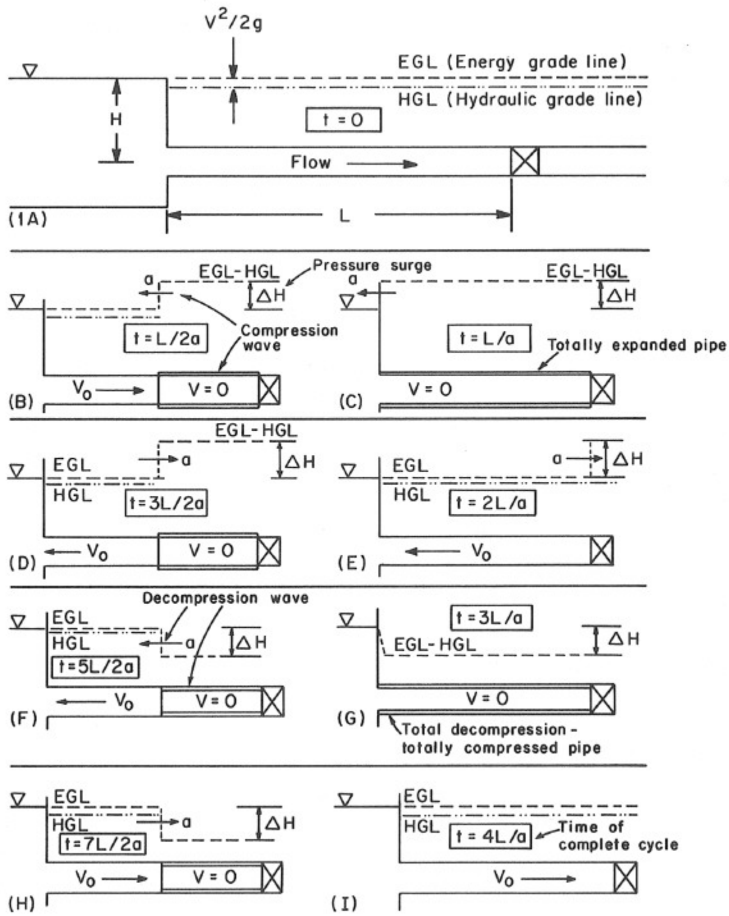

Figure 10-3 Schematic of the water hammer phenomenon.

At this time [refer to part (c) of Fig. 10-3] all the fluid in the pipeline has been brought to rest and the pipeline is subject to a pressure increase of ΔH over its entire length. As the pressure wave is reflected in the reservoir, the static pressure H is re-established at the reservoir end of the pipeline. The resulting pressure differential causes the fluid to move out of the pipeline towards the reservoir at velocity v_0 as indicated in part (d) of the figure. At $t = 2\ L/a$ the pressure wave has returned to the valve and the pressure in the pipeline is back to the original distribution that existed at $t = 0$ [as shown in part (e)].

Since the valve is still closed, the velocity at the valve is zero and the velocity differential causes a decrease in pressure. Now a reduced pressure wave moves back up the pipeline towards the reservoir at speed a [see Fig. 10-3(f)]. This continues until the pressure wave returns to the reservoir at time $t = 3\ L/a$ when the entire pipeline is in compression. This condition is shown in part (g) of the figure. At this point pressure is re-established in the pipeline at head H and the pressure wave again moves towards the valve at velocity a. The wave arrives back at the valve at $t = 4\ L/a$ at which time the pressure distribution is returned to the original condition depicted in Fig. 10-3(a). This process continues every $4\ L/a$ seconds until the fluctuations dampen out due to fluid friction and elasticity of the pipeline. When the fluctuations completely dampen out, the fluid comes to rest.

Hydraulics of Water Hammer

The magnitude of pressure surges associated with water hammer is a function of

- **(a)** System geometry.
- **(b)** Magnitude of velocity change.
- **(c)** Velocity of the pressure wave for a particular system.

The governing equation for this relationship is given by

$$\Delta H = \left(\frac{a}{g}\right)\Delta v \tag{10-18}$$

where

$$\begin{aligned} \Delta H &= \text{surge pressure, L} \\ a &= \text{pressure wave velocity, L/T} \\ g &= \text{acceleration of gravity, L/T}^2 \\ \Delta v &= \text{change in fluid velocity, L/T} \end{aligned}$$

The velocity, or celerity, of the pressure wave is a function of both pipe material properties and fluid properties. The pipe material properties which affect the celerity are (a) modulus of elasticity, (b) diameter, and (c) wall thickness. The fluid properties of major importance are (a) modulus of elasticity, (b) fluid density, and (c) amount of air entrained in the fluid. Applying the impulse-momentum equation and the principle of conservation of mass, the following equation for pressure wave velocity can be derived (Watters, 1984):

$$a = \frac{k_1\left[\dfrac{K}{\rho}\right]^{0.5}}{\left[1 + \left(\dfrac{K}{E}\right)\left(\dfrac{D}{t}\right)C_1\right]^{0.5}} \tag{10-19}$$

where

$$\begin{aligned}
a &= \text{pressure wave velocity, L/T} \\
k_1 &= \text{conversion constant dependent upon units} \\
K &= \text{bulk modulus of elasticity of water, F/L}^2 \\
\rho &= \text{density of water, M/L}^3 \\
D &= \text{inside pipe diameter, L} \\
t &= \text{pipe wall thickness, L} \\
E &= \text{pipe modulus of elasticity, F/L}^2 \\
C_1 &= \text{pipe support coefficient}
\end{aligned}$$

The pipe support coefficient is a function of the method of pipe constraint and the relationship between longitudinal and circumferential stress given by the Poisson ratio. The function is given by the following equation (Watters, 1984):

$$C_1 = 1.25 - \mu, \text{ pipes anchored on one end only} \tag{10-20a}$$

$$C_1 = 1 - \mu^2, \text{ pipes anchored at both ends only} \tag{10-20b}$$

$$C_1 = 1.0, \text{ pipes with expansion joints along the length} \tag{10-20c}$$

where

$$\mu = \text{Poisson ratio for pipe material}$$

The value of k_1 depends on the system of units employed. Consider that the same units are used for K and E in Eq. (10-19) and the same units for D and t. For SI units with K in Pa and ρ in kg/m^3, k_1 is equal to 1.0. For the English system with K given in psi and ρ in slugs/ft^3, k_1 is equal to 12.0. The relative properties of water and pipe materials required for Eq. (10-19) are given in Tables 10-7 and 10-8. Equation (10-19) does not account for entrainment of air in the pipeline which serves to reduce K and a. The result of Eq. (10-19) is therefore a conservative value of a which predicts the most severe water hammer pressure surge. This value is reasonable for design purposes since it represents the upper limit of operating conditions.

Application of Eq. (10-19) for the analysis of pressure surges in unsteady flow regimes is demonstrated in the following example problems. The analyses demonstrated are simplified in that they do not account for the nonlinearity of the pressure surge with time. An analysis considering nonlinearity requires solution of two simultaneous partial differential equations by the method of characteristics or more commonly by computerized numerical methods (Watters, 1984; Stephenson, 1981).

TABLE 10-7 Various properties of water as a function of temperature.

Temperature		Specific Weight		Density		Modulus of Elasticity	
(C)	(F)	(kN/m^3)	(lb/ft^3)	(kg/m^3)	(slug/ft^3)	(MPa)	(psi)
0	32	9.81	62.4	1000	1.94	1979	287,000
5	41	9.81	62.4	1000	1.94	2048	297,000
10	50	9.81	62.4	1000	1.94	2103	305,000
15	59	9.81	62.4	1000	1.94	2151	312,000
20	68	9.79	62.3	998	1.94	2193	318,000
25	77	9.78	62.3	997	1.93	2224	322,500
30	86	9.77	62.2	996	1.93	2251	326,500
35	95	9.75	62.1	994	1.93	2272	329,500
40	104	9.73	61.9	992	1.92	2286	331,500
45	113	9.71	61.8	990	1.92	2289	332,000
50	122	9.69	61.7	988	1.92	2289	332,000
55	131	9.67	61.6	986	1.91	2282	331,000
60	140	9.65	61.4	984	1.91	2275	330,000
65	149	9.62	61.2	981	1.90	2261	328,000
70	158	9.59	61.0	978	1.90	2248	326,000
75	167	9.56	60.9	975	1.89	2227	323,000
80	176	9.53	60.7	971	1.88	2206	320,000
85	185	9.50	60.5	968	1.88	2172	315,000
90	194	9.47	60.3	965	1.87	2137	310,000
95	203	9.44	60.1	962	1.87	2103	305,000
100	212	9.40	59.8	958	1.86	2068	300,000

TABLE 10-8 Modulus of elasticity and Poisson's ratio for common pipe materials.

Pipe Material	Modulus of Elasticity (MPa)	Modulus of Elasticity (10^5 psi)	Poisson's Ratio
Asbestos-Cement	20,684	30	0.20
Cast Iron	103,421	150	0.29
Ductile Iron	165,474	240	0.29
Permastran	9,653	14	0.35
Polyvinyl Chloride	2,758	4	0.46
Polyethylene	689	1	0.40
Steel	206,843	300	0.30

Example Problem 10-4

A steel pipe 1500 m in length and 0.5 m in diameter has a wall thickness of 5.5 cm. The pipe carries water from a reservoir and discharges into the atmosphere 50 m below the water level of the reservoir. A valve installed at the downstream end of the pipe allows a flow rate of 0.4 m^3/s. If the valve is completely closed in 2.0 s, calculate the maximum water hammer pressure surge at the valve and the time for the pressure wave to return to the valve. Assume the pipe is anchored at one end.

Solution Assume standard temperature 20°C. From Tables 10-7 and 10-8,

$$K = 2.19 \times 10^9 \text{ N/m}^2$$
$$E = 2.07 \times 10^{11} \text{ N/m}^2$$
$$\rho = 998 \text{ kg/m}^3$$
$$\mu = 0.30$$

Based on units and pipe constraints,

$$k_1 = 1.0$$
$$C_1 = 1.25 - \mu = 0.95$$

Compute the celerity,

$$a = \frac{1.0\left[\dfrac{2.19 \times 10^9 \text{ N/m}^2}{998 \text{ kg/m}^3}\right]^{1/2}}{\left[1 + \left(\dfrac{2.19 \times 10^9}{2.07 \times 10^{11}}\right)\left(\dfrac{0.5 \text{ m}}{0.055 \text{ m}}\right)0.95\right]^{1/2}}$$

$$a = 1418 \text{ m/s}$$

Time required for pressure wave to return to valve:

$$t = \frac{2L}{a} = \frac{2(1500 \text{ m})}{1418 \text{ m/s}}$$

$$t = 2.12 \text{ s}$$

The valve is completely closed by this time. The change in velocity equals

$$\Delta v = \frac{0.4 \text{ m}^3/\text{s}}{\dfrac{\pi}{4}(0.5 \text{ m})^2}$$

$$\Delta v = 2.04 \text{ m/s}$$

Pressure surge:

$$\Delta H = \left(\frac{a}{g}\right)\Delta v$$

$$\Delta H = \left(\frac{1418 \text{ m/s}}{9.81 \text{ m/s}^2}\right)(2.04 \text{ m/s})$$

$$\Delta H = 295 \text{ m}$$

More gradual valve closure and water hammer protection devices are required on this pipeline to reduce the potential for damage due to such high pressure surges.

Example Problem 10-5

(a) Compute the pressure surge in psi in a 12-inch outside diameter class 150 PVC 1220 pipe carrying water at 60°F due to a sudden change in velocity of 2.0 ft/s. Assume that the pipe is fitted with expansion joints throughout its length.

(b) Compute the maximum allowable change in velocity for the given pipe if the design is based on a 2.8 to 1 safety factor.

Solution

(a) Pressure rating = PR = 150 psi

$$PR = \frac{2\,S}{\frac{D_0}{t} - 1}$$

From Table 10-1 for PVC 1220,

$$S = 2000 \text{ psi}$$

Substituting into PR equation,

$$150 \text{ psi} = \frac{4000 \text{ psi}}{(12 \text{ in.}/t) - 1}$$

$$t = 0.4337 \text{ in.}$$

From Tables 10-7 and 10-8,

$$\begin{aligned}
K &= 313{,}000 \text{ psi} \\
E &= 400{,}000 \text{ psi} \\
\rho &= 1.938 \text{ slugs/ft}^3 \\
D_i &= D_0 - 2\,t \\
&= 12 \text{ in.} - 2(0.4337 \text{ in.}) \\
&= 11.133 \text{ in.}
\end{aligned}$$

Calculate celerity:

$$a = \frac{12\left[\dfrac{313{,}000 \text{ psi}}{(1.938 \text{ slugs/ft}^3)}\right]^{1/2}}{\left[1 + \left(\dfrac{313{,}000 \text{ psi}}{400{,}000 \text{ psi}}\right)\left(\dfrac{11.133 \text{ in.}}{0.4337 \text{ in.}}\right)1.0\right]^{1/2}}$$

$$a = 1052 \text{ ft/s}$$

Calculate the pressure surge,

$$\Delta H = \frac{a}{g}\Delta v$$

$$\Delta H = \left(\frac{1050 \text{ ft/s}}{32.17 \text{ ft/s}^2}\right)(2.0 \text{ ft/s})$$

$$\Delta H = 65.29 \text{ ft} = 28.3 \text{ psi}$$

(b) Maximum operating pressure:

$$P_{max} = \frac{\dfrac{2.0\,S_{lt}}{S_f}}{\dfrac{D_0}{t} - 1}$$

From Table 10-1,

$$S_{lt} = 2.0(S)$$
$$= 2.0(2000 \text{ psi})$$
$$S_{lt} = 4000 \text{ psi}$$

Applying 2.8 safety factor,

$$P_{max} = \frac{\dfrac{2.0(4000 \text{ psi})}{2.8}}{\left(\dfrac{12 \text{ in.}}{0.4337 \text{ in.}}\right) - 1}$$

$$P_{max} = 107 \text{ psi}$$

Maximum surge pressure magnitude:

$$\Delta H_{max} = PR - P_{max}$$
$$\Delta H_{max} = 150 \text{ psi} - 107 \text{ psi}$$
$$\Delta H_{max} = 43 \text{ psi}$$

Rearranging the surge pressure equation,

$$\Delta v_{max} = \Delta H_{max}(g/a)$$
$$= 43 \text{ psi}(2.31 \text{ ft/psi}) \left[\frac{(32.17 \text{ ft/s}^2)}{(1052 \text{ ft/s})}\right]$$
$$\Delta v_{max} = 3.04 \text{ ft/s}$$

Table 10-9 indicates the maximum operating pressure (maximum working pressure) for thermoplastic pipe from the ASAE standards in the case when the surge pressure is unknown. As long as the operating pressure is less than the maximum indicated in Table 10-9, the pipeline should be able to sustain surge pressures developed in normal operations without failure and without additional pipeline protection devices. Even if pipeline protection devices for water hammer are installed, it is prudent to operate the pipe at pressures below the maximum indicated in case of failure of the protection devices.

Standard pipeline design requires specification of protection devices which alleviate the effects of water hammer and other damaging conditions on the pipeline. These devices are key elements to the safe and continuous operation of the pipeline. They are described in the following section.

10.4 Pipeline System Components

Various devices need to be specified in the plans for a pipeline to insure safe and efficient operation of the system. These devices are required to control entrapped air, to allow the pipeline to drain after operations are completed, and to protect the line from pressure surges caused by hydraulic transients. Without these devices, the pipeline may not only operate inefficiently, it is subject to failure. This section describes the type of devices required to overcome the most common pipeline operational problems and their placement on the line.

TABLE 10-9 Maximum allowable pressure for nonthreaded thermoplastic pipes when surge pressures are not known.*† (Taken from ASAE Standard S376.1.)

SDR		PVC materials (all pipes OD based)								PE materials (pipes made to both OD & ID basis)						ABS materials (all pipes OD based)					
		PVC 1120 PVC 1220 PVC 2120		PVC 2116		PVC 2112		PVC 2110		PE 3408		PE 3406 PE 3306 PE 2306		PE 2305		ABS 1316		ABS 1212		ABS 1210	
OD based pipe	ID based pipe	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa
	5.3									180	1240	144	995	115	795						
	7.0									144	995	115	795	90	620						
11.0	9.0									115	795	90	620	72	495						
13.5	11.5	227	1565	180	1240	144	995	115	795							180	1240	144	995	115	795
17.0	15.0	180	1240	144	995	115	795	90	620	72	495	58	400	45	310	144	995	115	795	90	620
21.0		144	995	115	795	90	620	72	495	58	400	45	310			115	795	90	620	72	495
26.0		115	795	90	620	72	495	58	400	45	310	36	250			90	620	72	495	58	400
32.5		90	620	72	495	58	400	45	310	36	250	29	200			72	495	58	400	46	315
41.0		72	495	58	400	45	310	36	250	29	200	22	150			58	400	46	315	36	250
51.0		58	400	45	310	36	250	29	200							46	315	36	250	29	200
64.0		45	310	36	250	29	200	22	150												
81.0		36	250	29	200	22	150	18	125							29	200	22	150	18	125
93.5		31	215																		
50 ft head		21	145																		

*Maximum allowable working pressure = pressure rating (PR) × 0.782 for SDR and DR pipe.

†For water at 23°C (73.4°F).

Air in Pipelines

Occurence

Air in pipelines may occur due to absorption at free water surfaces or entrainment in turbulent flow at pipeline entrances. Air may be present in solution or in the form of pockets or bubbles. Air in solution does not generally cause a problem unless pressure is reduced sufficiently that the dissolved air forms bubbles. The formation of bubbles may lead to cavitation damage on pump impellers or the bubbles may coalesce to form air pockets along the pipeline. The occurence of air in pipelines causes a reduction in effective flow area resulting in increased flow velocity and increased headloss. Air intermittently drawn into pumps and pipelines can cause vibration due to unsteady flow.

Pump inlet arrangements which promote either intermittent air entry or bubbles in pipelines and pumps can be avoided by (a) providing adequate submergence of the pump intake, (b) providing for minimum flow turbulence at the pump intake—that is, by assuring that the intake velocity is less than 1 m/s (3 ft/s) and that the nearest obstruction to flow is at a distance greater than four pipe diameters from the pump inlet, and (c) minimizing the opportunity for entrapping air on the suction side of the pump—that is, by using an eccentric reducer and checking that the pump intake is the highest point in the suction line. These points are more fully discussed in the Pump Installation section of the Pump System chapter.

Air release—vacuum relief valves

Air which does enter pipelines requires venting to the atmosphere by air release valves. These valves vent air which may collect at high points in the pipeline due to entrainment at the pump intake and release air during pipe filling. Vacuum relief valves vent into the pipe and draw air from the atmosphere when the pipeline is draining.

Venting of entrapped air is accomplished by small orifice valves—that is, valves with orifices up to 3 mm (0.12 inch) in diameter. These are activated by the accumulated air forcing a ball to be released from the seat around an orifice or forcing a ball to activate a lever which vents air to the atmosphere. Figure 10-4 demonstrates the design of such a valve.

Relatively rapid purging of air from or induction of air into pipelines, as occurs during pipeline filling or draining, is accomplished by large orifice valves. These

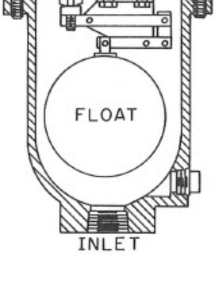

Figure 10-4 Graphic of air release valve.

valves are normally activated by releasing a ball from the orifice seat. Small and large orifice valves may be installed independently. However, because the specified location of such valves on the pipeline is often the same, dual or combination type valves are common. Such a valve is depicted in Fig. 10-5.

Referring to Fig. 10-5 we observe that during pipeline filling the large orifice ball is at the bottom of the valve—that is, unseated—and air is free to escape to the atmosphere. When the pipeline is filled with water, the large orifice ball becomes seated. If entrained air collects at the valve site, the small orifice ball is depressed and air is released to the atmosphere. When the pipeline is drained, atmospheric pressure exceeds the internal pressure in the pipeline and the large ball becomes unseated. This allows air at atmospheric pressure to freely enter the pipeline and promotes complete drainage of the line. Without a vacuum relief valve, drainage may not be complete and in fact the vacuum developed could collapse the pipeline.

Valve placement

Large orifice valves should be installed at high points in the physical layout of the pipeline and at high points along the hydraulic gradient. Together with small orifice valves, or more commonly as a combination valve, they should also be installed at the ends and intermediate points along the length of pipeline which is parallel to the hydraulic grade line. It is recommended that combination valves be placed every 0.5 to 1.0 km (0.3 to 0.6 mi) along descending pipeline sections, especially at points where the descending pipeline grade is steep.

Air may be released from solution if there is adequate drop in pressure. Such conditions may occur along a mainline which rises in elevation or where velocity is increased through a flow restriction such as a partially closed valve. This air should be bled off by appropriately placed small orifice air release valves to avoid blockage of the pipeline. Air release valves are therefore required along all ascending lengths of pipeline, particularly at points where there is a decrease in the upward gradient. Other reasonable points of air release valve placement are on the discharge side of pumps, at high points on large flow control valves, and upstream of orifice plates and reducing tapers if they are located in relatively high velocity flow streams.

Valve sizing

Air release/vacuum relief valves are sized by the diameter of the connection between the pipeline and the valve. The actual orifice diameter is generally smaller

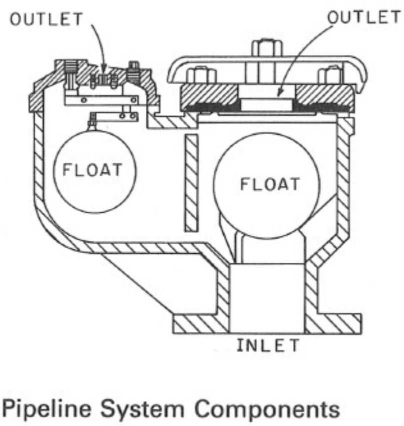

Figure 10-5 Graphic of combination air release/vacuum relief valve.

and can be obtained by referring to an equipment manufacturer's catalog. The ratio of valve diameter to pipeline diameter for high pressure systems should be 0.1 or greater—that is, valve diameter should be $\geq$10 percent of pipeline diameter. Table 10-10 indicates minimum recommended valve sizing for high and low pressure systems taken from the ASAE Standards (1987).

Pipeline Protection

The need to protect pipelines from surge pressure that develops during sudden valve closure has been previously described in the section on water hammer. An inverse condition exists if a pump is suddenly shut off. In this case the flow stops and the sudden deceleration of the water column causes a pressure drop at the point of pump discharge. This negative pressure surge moves along the pipeline in an analagous manner to the positive pressure surge demonstrated for a sudden valve closure in Fig. 10-3. The sudden pressure drop may cause vaporization, pipeline collapse, or induction of air at atmospheric pressure into the pipeline through combination air release/vacuum relief valves.

High pressure surges can be controlled through installation of pressure relief valves at those points in a pipeline where it is anticipated pressure waves will be generated or reflected. Relatively simple valves of this type suitable for on-farm distribution systems operate using a spring-loaded discharge valve. Under normal operating pressures the valve remains in the closed position. If water hammer causes a pressure surge in the line, the excess pressure depresses the spring thereby opening the valve and causing water to be discharged from the line. The result is that the magnitude of the pressure surge is considerably reduced and the reflected shock wave does not damage the pipeline. The level of pressure at which the valve will be activated can be preset by the manufacturer or set in the field to correspond to the operating pressure of a particular system.

The opposite problem is encountered when sudden downsurges in pressure are caused by the stopping of pumps. The objective of pipeline protection in this case is

TABLE 10-10 Air release and vacuum relief valve standards. (Adapted from ASAE Standard S376.1.)

High Pressure Systems			
Pipe Diameter		Minimum Valve Outlet Diameter	
(mm)	(in.)	(mm)	(in.)
102	4 or less	13	0.5
127–203	5–8	25	1.0
254–500	10–20	51	2.0
530	21 or larger	0.1 × pipe diameter	
Low Pressure Systems			
Pipe Diameter		Minimum Valve Outlet Diameter	
(mm)	(in.)	(mm)	(in.)
152	6 or less	51	2.0
178–254	7–10	76	3.0
305	12 or larger	102	4

to reduce the magnitude of the downsurge so the reflected high pressure wave is also reduced. The most common method of limiting downsurge is to feed water into the pipeline as soon as the pressure drops. Different methods of accomplishing this objective are applicable depending on the operating pressure of the system and anticipated magnitude of the pressure surge.

Pump bypass

Figure 10-6 indicates a pump bypass which is coupled with a check valve to allow water to enter the pipeline after the pump is shut off. This system is only operational if the pressure in the pipeline drops below the net positive suction head available to induce flow through the check valve. The minimum pressure the pipeline will experience with this type of installation is the net positive suction head available minus friction headloss through the bypass. This method is not applicable if the reflected surge pressure wave, equal in magnitude to the initial pressure drop, is damaging to the pipeline. This method is only recommended when the operational discharge pressure head is considerably less than the product involving pressure wave celerity and design flow velocity as given by the equation

$$H_O \ll a\left(\frac{v_0}{g}\right) \tag{10-21}$$

where

H_0 = operating pressure head, m
a = celerity of pressure wave, m/s
v_0 = design flow velocity, m/s
g = acceleration of gravity, m/s^2

The celerity of the pressure wave is computed using Eq. (10-19). Typical ranges of celerity are approximately 200 m/s for thermoplastic pipe and 850 m/s for thin wall steel pipes.

Surge tank

Surge tanks are connected to the pipeline and may be vented to the atmosphere. Tanks which are vented to the atmosphere are often called stand pipes. The height of water in the stand pipe is at the hydraulic grade line during normal operating conditions. Water in the tank dampens both decompression and compression waves generated by water hammer. The hydraulic transients between the pump and stand pipe are high frequency pressure waves which cause only slight changes in the

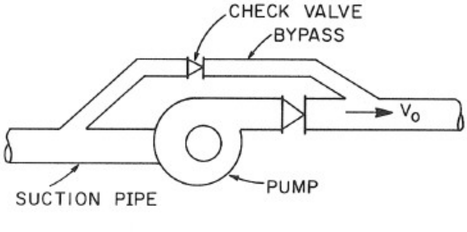

Figure 10-6 Schematic of pump bypass with check valve.

water level in the tank. However, the installation of the tank causes dampening of the pressure waves between the tank and the delivery end of the pipeline.

Closed or pneumatic surge tanks are used in high pressure systems where stand pipe heights are uneconomical. The tank contains air which serves as a pressure buffer. An air compressor is often necessary to replace air absorbed by the pressurized water. The discussion which follows is with reference to Fig. 10-7 which indicates possible locations for a number of water hammer protection devices along a pipeline.

The rate of deceleration of the flow stream between the stand pipe and end of the pipeline is given by

$$\frac{dv}{dt} = \frac{-g\Delta h}{l} \tag{10-22}$$

where

Δh = difference in height between hydraulic grade line and depressed water level in stand pipe following passage of pressure wave, m

l = length of pipeline between stand pipe and delivery end, m

By continuity,

$$A_p(v) = A_t\frac{dh}{dt} \tag{10-23}$$

where

A_p = cross-sectional area of pipe, m^2

A_t = cross-sectional area of stand pipe, m^2

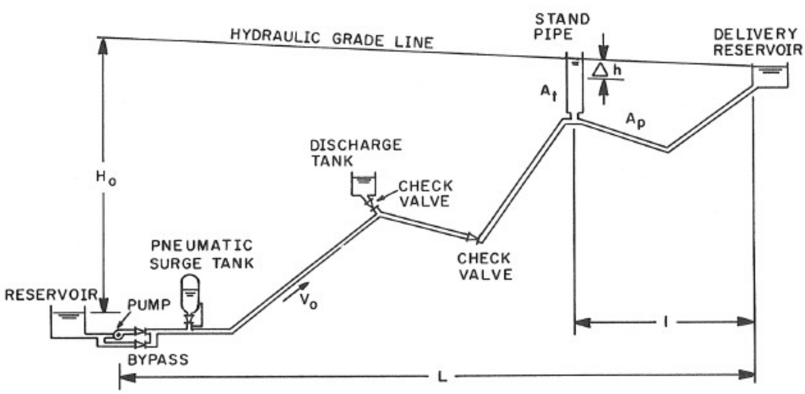

Figure 10-7 Pipeline profile with practical locations for various pipeline protection devices.

Combining Eqs. (10-22) and (10-23) and noting that at initial conditions $t = 0$, $\Delta h = 0$, and $v = v_0$, the following equation is derived for the change of water level in the tank (Stephenson, 1981):

$$\Delta h = v_0 \left[\frac{A_p(1)}{A_t g}\right]^{0.5} \sin\left\{t\left[\frac{A_p(g)}{A_t 1}\right]^{0.5}\right\} \qquad (10\text{-}24)$$

where

t = time required for flow stoppage following pump shut off, s

The term in brackets following the sin is in radians. Application of Eq. (10-24) to estimate the expected depression of water in a surge tank is indicated in the following example problem.

Example Problem 10-6

A stand pipe 20 cm in diameter is installed on a pipeline for water hammer protection. The pipeline diameter is 30 cm and the flow velocity is 1.5 m/s under normal operating conditions. Assume that 0.5 seconds is required for flow stoppage after the pump is shut off. The distance from the surge tank to the end of the pipeline is 300 m. Using a safety factor of 1.5 to avoid underestimation, compute the expected change in height of water in the tank.

Solution Let h equal the height of the tank calculated considering pipe hydraulics and h_{sf} the height adjusted for the safety factor.

$$h_{sf} = 1.5\,\Delta h$$

From Eq. (10-24),

$$\Delta h = v_0 \left[\frac{A_p(1)}{A_t g}\right]^{0.5} \sin\left\{t\left[\frac{A_p(g)}{A_t 1}\right]^{0.5}\right\}$$

Substituting the problem values,

$$\Delta h = 1.5\text{ m/s}\left\{\frac{(0.30\text{ m})^2\,300\text{ m}}{(0.20\text{ m})^2\,9.81\text{ m/s}^2}\right\}^{0.5} \sin\left\{0.5\text{ s}\left[\frac{(0.30\text{ m})^2\,(9.81\text{ m/s}^2)}{(0.20\text{ m})^2\,300\text{ m}}\right]^{0.5}\right\}$$

$$\Delta h = 1.68\text{ m}$$

$$h_{sf} = 1.5(\Delta h) = 2.5\text{ m}$$

Discharge tank

Discharge tanks are applied where the pipeline profile is considerably lower than the hydraulic grade line. It is also recommended that the pipeline be concave downwards at the location of the discharge tank. The discharge tank must be separated from the pipeline by a check valve which allows water to flow one way from the tank to the pipeline. This valve isolates the tank from positive pressure surges in the pipeline. The discharge tank protects the pipeline from negative pressure surges following pump shut down. The analysis of surge protection for a discharge tank is similar to that indicated for the stand pipe.

Not all of the pipeline protection devices indicated in this section are required on irrigation system mainlines. The extent to which surge protection must be installed depends on the flow velocities and operating pressures of the system. For many moderate size irrigation systems without high operating pressures, the basic spring-loaded pressure relief valve previously discussed will adequately protect the pipeline.

10.5 Pipeline Installation

This section indicates specifications recommended for installation of underground pipelines. The recommendations are taken for the most part from the ASAE standards for underground irrigation lines and are considered adequate for a broad range of irrigation systems. Additional specifications may be required for very high capacity lines or lines expected to operate at very high pressures (Stephenson, 1981).

Trenching Requirements

Trench width

Trench widths above the top of the pipe should not be greater than 0.6 m (2 ft) wider than the pipe diameter. This standard may be exceeded in unstable soils where sloughing or caving may occur and it is required to slope the sidewalls above the pipe. Minimum and maximum trench widths for low pressure lines as a function of pipe diameter are indicated in Table 10-11.

Trench depth

The normal required minimum depth of cover as a function of pipe diameter is given in Table 10-12. If extra fill must be placed above the soil surface to provide the minimum depth of cover, the top width of the fill should be no less than 3 m (10 ft) and the side-slope ratio no less than 4 Horizontal: 1 Vertical. The minimum depth of cover for locations in which vehicular wheel loads will be applied to the

TABLE 10-11 Minimum and maximum trench widths. (Adapted from ASAE Standard S376.1.)

Pipe Diameter		Approximate Trench Width			
		Minimum		Maximum	
(in.)	(mm)	(in.)	(mm)	(in.)	(mm)
4	102	16	400	30	760
6	152	18	450	30	760
8	203	20	510	30	760
10	254	22	560	30	760
12	305	24	610	30	760
14	356	26	660	30	760
15	381	27	690	30	760
18	457–475	30	760	36	910
20	508	32	810	36	910
24	610–630	36	910	42	1070
27	710	40	1020	46	1170

TABLE 10-12 Minimum depth of cover.

Pipe Diameter		Minimum Depth	
(mm)	(in.)	(mm)	(in.)
13–64	0.5–2.5	460	18
76–102	3–4	610	24
102	4	760	30

trench is 0.76 m (30 in.) above the top of the pipeline. This depth of cover is required before wheel loads are applied to the pipeline for the first time. This minimum depth of cover standard applies to both low and high pressure lines.

Thrust Blocking

Thrust blocking prevents the pipeline from shifting in the trench and is required to increase the capability of the line to resist deformation and failure at the joints. Thrust loading results from unequal forces developed by a change in direction of water in the pipeline. The thrust block serves to transfer the load from the pipe to a large load bearing surface.

Thrust blocks are required at the following locations:

(a) Where the pipeline changes the direction of water flow (i.e., at ties, elbows, crosses, wyes, and tees).

(b) Where pipe size changes (i.e., at reducers, reducing tees and crosses).

(c) At the end of a pipeline (i.e., at caps and plugs).

(d) Where there is an in-line flow control valve.

Thrust blocks are formed against a solid trench wall that has been excavated by hand. The pipe couplings themselves are not to be thrust blocked. This is indicated in Fig. 10-8 which demonstrates the proper location and placement of thrust-block material. The following steps to calculate the required load-bearing area are given with reference to Tables 10-13 and 10-14:

(a) Multiply the pipeline operating pressure by the appropriate value in Table 10-13 to obtain an estimate of the total thrust.

(b) Determine an estimation of the bearing strength of the soil from Table 10-14.

(c) Divide the total thrust determined in step (a) by the bearing strength to calculate the required load bearing area.

An alternative procedure applies the magnitude of the deflection of the flow stream in degrees. The amount of deflection is multiplied times the value tabulated in Table 10-15 as a function of pipe diameter. The result is multiplied by the operating pressure divided by 100 to obtain the total side thrust. To compute the total side thrust in newtons (N), the operating pressure is required in kPa. To obtain the side thrust in lb, the operating pressure is required in psi. The resulting total thrust is divided by the bearing strength of the soil from Table 10-14 to obtain the required

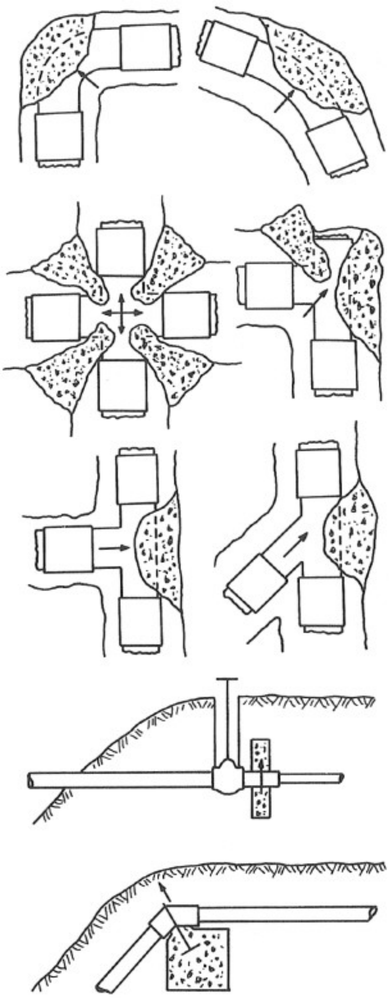

Figure 10-8 Examples of various configurations for installation of thrust blocks. (Adapted from ASAE Standard S376.1.)

load-bearing area as was indicated in step (c) of the preceding. The procedure to calculate the load-bearing area is demonstrated in the following example problem.

Example Problem 10-7

Compute the required load-bearing area for a thrust block at a 90 degree elbow in a 273 mm diameter pipeline which has an operating pressure of 586 kPa. The pipeline is installed in a trench made up of a coarse sand.

Solution From Table 10-13 for 90° elbow,

$$\text{total thrust} = T = 130.0(586\text{ kPa})$$
$$T = 76{,}180\text{ N}$$

From Table 10–14 for coarse sand,

$$\text{soil bearing strength} = s_b = 150\text{ kPa}$$

Area required:

$$A_r = \frac{T}{s_b} = \frac{76{,}180 \text{ N}}{150 \text{ kPa}}$$

$$A_r = 0.51 \text{ m}^2$$

Thrust load is therefore to be transferred to approximately 0.51 m^2 bearing area of thrust block.

TABLE 10-13 Pipeline thrust factors. (Adapted from ASAE Standard S376.1.)

Pipe Diameter					
(in.)	(mm)	Dead End or Tee	90 Degree Elbow	45 Degree Elbow	22.5 Degree Elbow
1.5	48	2.94	4.16	2.25	1.15
2	60	4.56	6.45	3.50	1.78
2.5	73	6.65	9.40	5.10	2.60
3	89	9.80	13.9	7.51	3.82
3.5	102	12.8	18.1	9.81	4.99
4	114	16.2	23.0	12.4	6.31
5	141	24.7	35.0	18.9	9.63
6	168	34.8	49.2	26.7	13.6
8	219	59.0	83.5	45.2	23.0
10	273	91.5	130.0	70.0	35.8
12	324	129.0	182.0	98.5	50.3
14	363	160.0	226.5	122.6	62.6
15	389	183.9	260.0	140.7	71.9
16	406	201.1	284.4	153.8	78.6
18	475	274.7	388.4	210.1	107.4
20	518	326.9	462.2	250.1	127.8
21	560	381.8	539.9	292.1	149.3
24	630	483.2	683.2	369.6	188.9
27	710	613.7	867.8	469.5	239.9

TABLE 10-14 Bearing strength of soils. (Adapted from ASAE Standard S376.1.)

	Safe Bearing Load	
Soil Type	(kPa)	(lb/ft^2)
Sound shale	500	10,000
Cemented to gravel and sand difficult to pick	200	4,000
Coarse and fine compact sand	150	3,000
Medium clay—can be spaded	100	2,000
Soft clay	50	1,000
Muck	0	0

TABLE 10–15 Data for alternative side thrust procedure. (Adapted from ASAE Standard S376.1.)

Pipe Diameter		Side Thrust per Degree	
(in.)	(mm)	(lb)	(N)
1.5	48	5.1	22.7
2	60	7.9	35.1
2.5	73	11.6	51.6
3	89	17.1	76.1
3.5	102	22.4	99.6
4	114	28.3	125.9
5	141	43.1	191.7
6	168	60.8	270.5
8	219	103.0	458.2
10	273	160.0	711.7
12	324	225.0	1000.8
14	363	278.2	1237.4
15	389	319.6	1421.6
16	406	349.3	1553.7
18	475	477.3	2123.0
20	518	568.0	2526.5
21	560	663.6	2951.7
24	630	839.6	3734.5
27	710	1066.2	4742.5

Thrust block material

The recommended thrust block material is concrete which has a compressive strength of at least 13.8 MPa (2000 psi). The concrete mixture should be one part cement, two parts washed sand, and four parts gravel. The blocks should be constructed so the bearing surface is to the greatest extent possible in a direct line with the maximum force exerted by the pipeline. The direction of these forces is indicated by the arrows in Fig. 10-8.

REFERENCES

American Society of Agricultural Engineers, "Design, Installation and Performance of Underground Thermoplastic Irrigation Pipelines," ASAE Standard S376.1, ASAE, St. Joseph, Michigan, 1987, pp. 501–511.

Colebrook, C. F., "Turbulent Flows in Pipes with Particular Reference to the Transition Region between Smooth and Rough Pipe Laws," *Journal Institute of Civil Engineers,* London, 1939.

Moody, L. F., "Friction Factors for Pipe Flow," Transactions of the American Society of Mechanical Engineers, vol. 66, 1944, p. 671.

Mott, R. L., *Applied Fluid Mechanics,* second edition. Columbus, Ohio: Charles E. Merrill Publishing, 1979.

Murdock, J. W., *Fluid Mechanics and Its Application*. Boston: Houghton Mifflin, 1976.

STEPHENSON, D., *Pipeline Design for Water Engineers,* second edition. Amsterdam: Elsevier, 1981.

WATTERS, G. Z., *Analysis and Control of Unsteady Flow in Pipelines,* second edition. Boston: Butterworths, 1984.

PROBLEMS

10-1. The pressure rating of a PE pipe according to the ASAE standards is 160 kPa. The pipe is inside diameter based and has an inside diameter of 500 mm and a wall thickness of 12 mm. Determine the standard code designation for the pipe.

10-2. The annual power cost of a section of new PVC pipeline 1 km in length is $12,000. The pipeline has a diameter of 400 m and carries a flow of 180 L/s. Calculate the annual cost of energy for the same section of pipeline if it became badly corroded but everything else remained the same.

10-3. Compute the average annual total cost over a period of 50 years for a section of galvanized steel pipe 40.0 cm in diameter, 2 km in length, with a continuous flow rate of 440 L/s. Cost of new pipe is $1740 per 100 m including installation, power cost is 6 cents per kWh, and the interest rate is 9 percent. Power cost and interest rate will be assumed constant over the 50-year period. Assume the pipeline is horizontal and the pumping plant efficiency is 65 percent.

10-4. A 183 m long aluminum mainline with couplers of 15 cm diameter has a friction headloss of 1.93 m computed using the Hazen-Williams equation with C = 120. What is the volumetric flow rate through the pipe in m^3/s?

10-5. Water is discharged through a 90° flanged elbow which has a ratio of radius of curvature (r) to diameter (d) of 10. Discharge through the line is 12 L/s and friction loss is estimated as 0.0555 meters. What is the nominal diameter of the pipe in cm?

10-6. A 1600 ft long steel pipe discharges water from a reservoir at the upstream end and is fitted with an instantaneously closing valve on the downstream end. The pipe has an outside diameter of 12.750 inch and a wall thickness of 0.375 inch. Design discharge through the pipe is 3500 gpm with water at 50°F. The pipe is fitted with expansion joints throughout its length. Compute the time in seconds for a pressure wave to reach the reservoir if the valve is instantaneously closed.

10-7. A 203 mm inside diameter steel pipe set on the horizontal has a wall thickness of 8 mm and carries 65.56 degrees C water at a flow rate of 56 L/s. The pipeline is 457 m long, securely anchored at both ends only, and subject to an operating pressure of 3000 kPa. At t = 0 a gate valve is closed at the downstream end of the pipeline. Neglecting friction, compute the pressure in kPa at the middle of the pipeline at t = 1.0 s.

10-8. List the three main purposes of orifice type air release vacuum relief valves.