CHAPTER 12
HYDRAULIC TRANSIENT DESIGN FOR PIPELINE SYSTEMS

C. Samuel Martin
School of Civil and Environmental Engineering
Georgia Institute of Technology
Atlanta, Georgia

12.1 INTRODUCTION TO WATERHAMMER AND SURGING

By definition, *waterhammer* is a pressure (acoustic) wave phenomenon created by relatively sudden changes in the liquid velocity. In pipelines, sudden changes in the flow (velocity) can occur as a result of (1) pump and valve operation in pipelines, (2) vapor pocket collapse, or (3) even the impact of water following the rapid expulsion of air out of a vent or a partially open valve. Although the name waterhammer may appear to be a misnomer in that it implies only water and the connotation of a "hammering" noise, it has become a generic term for pressure wave effects in liquids. Strictly speaking, waterhammer can be directly related to the compressibility of the liquid—primarily water in this handbook. For slow changes in pipeline flow for which pressure waves have little to no effect, the unsteady flow phenomenon is called *surging*.

Potentially, waterhammer can create serious consequences for pipeline designers if not properly recognized and addressed by analysis and design modifications. There have been numerous pipeline failures of varying degrees and resulting repercussions of loss of property and life. Three principal design tactics for mitigation of waterhammer are (1) alteration of pipeline properties such as profile and diameter, (2) implementation of improved valve and pump control procedures, and (3) design and installation of surge control devices.

In this chapter, waterhammer and surging are defined and discussed in detail with reference to the two dominant sources of waterhammer—pump and/or valve operation. Detailed discussion of the hydraulic aspects of both valves and pumps and their effect on hydraulic transients will be presented. The undesirable and unwanted, but often potentially possible, event of liquid column separation and rejoining are a common justification for surge protection devices. Both the beneficial and detrimental effects of free (entrained or entrapped) air in water pipelines will be discussed with reference to waterhammer and surging. Finally, the efficacy of various surge protection devices for mitigation of waterhammer is included.

12.2 FUNDAMENTALS OF WATERHAMMER AND SURGE

The fundamentals of waterhammer, an elastic process, and surging, an incompressible phenomenon, are both developed on the basis of the basic conservational relationships of physics or fluid mechanics. The acoustic velocity stems from mass balance (continuity), while the fundamental waterhammer equation of Joukowsky originates from the application of linear momentum [see Eq. (12.2)].

12.2.1 Definitions

Some of the terms frequently used in waterhammer are defined as follows.

- *Waterhammer*. A pressure wave phenomenon for which liquid compressibility plays a role.
- *Surging*. An unsteady phenomenon governed solely by inertia. Often termed *mass oscillation* or referred to as either *rigid column* or *inelastic effect*.
- *Liquid column separation.* The formation of vapor cavities and their subsequent collapse and associated waterhammer on rejoining.
- *Entrapped air.* Free air located in a pipeline as a result of incomplete filling, inadequate venting, leaks under vacuum, air entrained from pump intake vortexing, and other sources.
- *Acoustic velocity.* The speed of a waterhammer or pressure wave in a pipeline.
- *Joukowsky equation.* Fundamental relationship relating waterhammer pressure change with velocity change and acoustic velocity. Strictly speaking, this equation only valid for sudden flow changes.

12.2.2 Acoustic Velocity

For wave propagation in liquid-filled pipes the *acoustic (sonic) velocity* is modified by the pipe wall elasticity by varying degrees, depending upon the elastic properties of the wall material and the relative wall thickness. The expression for the wave speed is

$$a = \frac{\sqrt{K/\rho}}{\sqrt{1 + \dfrac{D}{e}\dfrac{K}{E}}} = \frac{a_o}{\sqrt{1 + \dfrac{D}{e}\dfrac{K}{E}}} \tag{12.1}$$

where E is the elastic modulus of the pipe wall, D is the inside diameter of the pipe, e is the wall thickness, and a_o is the acoustic velocity in the liquid medium. In a very rigid pipe or in a tank, or in large water bodies, the acoustic velocity a reduces to the well=known relationship $a = a_o = \sqrt{(K/\rho)}$. For water $K = 2.19$ GPa (318,000 psi) and $\rho = 998$ kg/m^3 (1.936 slug/ft^3), yielding a value of $a_o = 1483$ m/sec (4865 ft/sec), a value many times that of any liquid velocity V.

12.2.3 Joukowsky (Waterhammer) Equation

There is always a pressure change Δp associated with the rapid velocity change ΔV across a waterhammer (pressure) wave. The relationship between Δp and ΔV from the basic physics of linear momentum yields the well-known *Joukowsky equation*

$$\Delta p = -\rho a \Delta V \tag{12.2}$$

where ρ is the liquid mass density, and *a* is the sonic velocity of the pressure wave in the fluid medium in the conduit. Conveniently using the concept of head, the *Joukowsky head rise* for instantaneous valve closure is

$$\Delta H = \frac{\Delta p}{\rho g} = -\frac{\rho a \Delta V}{\rho g} = \frac{a V_o}{g} \tag{12.3}$$

The compliance of a conduit or pipe wall can have a significant effect on modification of (1) the acoustic velocity, and (2) any resultant waterhammer, as can be shown from Eq. (12.1) and Eq. (12.2), respectively. For simple waterhammer waves for which only radial pipe motion (*hoop stress*) effects are considered, the germane physical pipe properties are Young's elastic modulus (*E*) and Poisson ratio (μ). Table 12.1 summarizes appropriate values of these two physical properties for some common pipe materials.

The effect of the elastic modulus (*E*) on the acoustic velocity in water-filled circular pipes for a range of the ratio of internal pipe diameter to wall thickness (*D*/*e*) is shown in Fig. 12.1 for various pipe materials.

12.3 HYDRAULIC CHARACTERISTICS OF VALVES

Valves are integral elements of any piping system used for the handling and transport of liquids. Their primary purposes are flow control, energy dissipation, and isolation of portions of the piping system for maintenance. It is important for the purposes of design and final operation to understand the hydraulic characteristics of valves under both steady and unsteady flow conditions. Examples of dynamic conditions are direct opening or closing of valves by a motor, the response of a swing check valve under unsteady conditions, and the action of hydraulic servovalves. The hydraulic characteristics of valves under either noncavitating or cavitating conditions vary considerably from one type of valve design to another. Moreover, valve characteristics also depend upon particular valve design for a special function, upon absolute size, on manufacturer as well as the type of pipe fitting employed. In this section the fundamentals of valve hydraulics are presented in terms of pressure drop (headloss) characteristics. Typical flow characteristics of selected valve types of control—gate, ball, and butterfly, are presented.

TABLE 12.1 Physical Properties of Common Pipe Materials

Material	*Young's Modulus E (GPa)*	*Poisson's Ratio μ*
Asbestos cement	23–24	–
Cast iron	80–170	0.25–0.27
Concrete	14–30	0.10–0.15
Concrete (reinforced)	30–60	–
Ductile iron	172	0.30
Polyethylene	0.7–0.8	0.46
PVC (polyvinyl chloride)	2.4–3.5	0.46
Steel	200–207	0.30

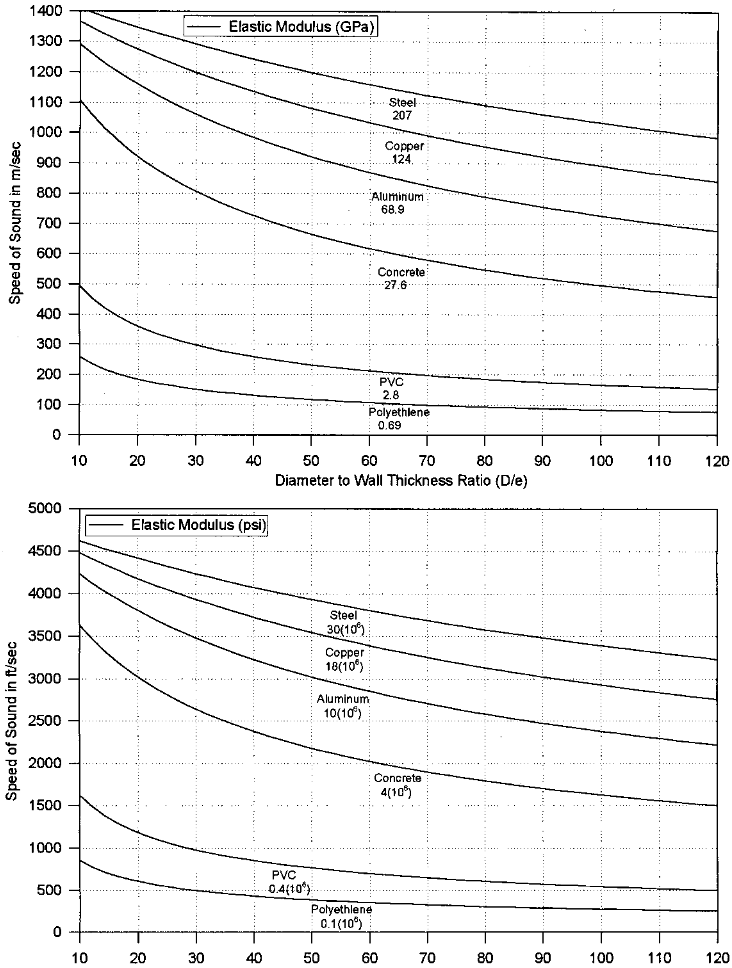


FIGURE 12.1 Effect of wall thickness of various pipe materials on acoustic velocity in water pipes.

12.3.1 Descriptions of Various Types of Valves

Valves used for the control of liquid flow vary widely in size, shape, and overall design due to vast differences in application. They can vary in size from a few millimeters in small tubing to many meters in hydroelectric installations, for which spherical and butterfly valves of very special design are built. The hydraulic characteristics of all types of valves, albeit different in design and size, can always be reduced to the same basic coefficients, notwithstanding fluid effects such as viscosity and cavitation. Figure 12.2 shows cross sections of some valve types to be discussed with relation to hydraulic performance.

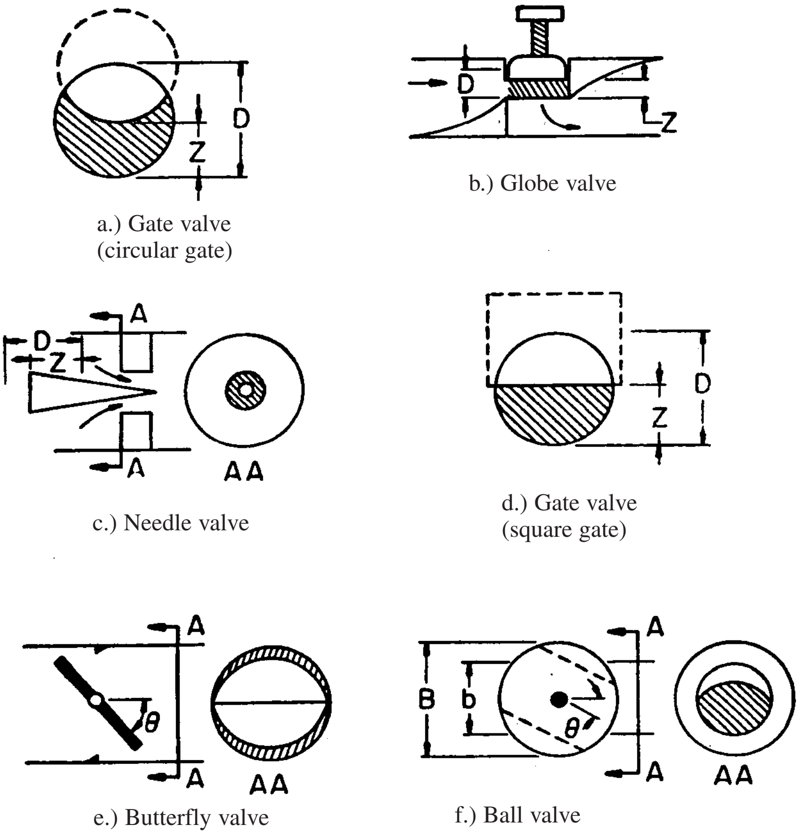


FIGURE 12.2 Cross sections of selected control valves: (From Wood and Jones, 1973).

12.3.2 Definition of Geometric Characteristics of Valves

The valve geometry, expressed in terms of cross-sectional area at any opening, sharpness of edges, type of passage, and valve shape, has a considerable influence on the eventual hydraulic characteristics. To understand the hydraulic characteristics of valves it is useful, however, to express the projected area of the valve in terms of geometric quantities. With reference to Fig. 12.2 the ratio of the projected open area of the valve A_v to the full open valve A_{vo} can be related to the valve opening, either a linear measure for a gate valve, or an angular one for rotary valves such as ball, cone, plug, and butterfly types. It should be noted that this geometric feature of the valve clearly has a bearing on the valve hydraulic performance, but should not be used directly for prediction of hydraulic performance—either steady state or transient. The actual hydraulic performance to be used in transient calculations should originate from experiment.

12.3.3 Definition of Hydraulic Performance of Valves

The hydraulic performance of a valve depends upon the flow passage through the valve opening and the subsequent recovery of pressure. The hydraulic characteristics of a valve under partial to fully opened conditions typically relate the volumetric flow rate to a characteristic valve area and the head loss ΔH across the valve. The principal fluid properties that can affect the flow characteristics are fluid density ρ, fluid viscosity μ, and liquid vapor pressure p_v if cavitation occurs. Except for small valves and/or viscous liquids or both, Reynolds number effects are usually not important, and will be neglected with reference to water. A valve in a pipeline acts as an obstruction, disturbs the flow, and in general causes a loss in energy as well as affecting the pressure distribution both upstream and downstream. The characteristics are expressed either in terms of (1) flow capacity as a function of a defined pressure drop or (2) energy dissipation (headloss) as a function of pipe velocity. In both instances the pressure or head drop is usually the difference in total head caused by the presence of the valve itself, minus any loss caused by regular pipe friction between measuring stations.

The proper manner in determining ΔH experimentally is to measure the hydraulic grade line (*HGL*) far enough both upstream and downstream of the valve so that uniform flow sections to the left of and to the right of the valve can be established, allowing for the extrapolation of the energy grade lines (*EGL*) to the plane of the valve. Otherwise, the valve headloss is not properly defined. It is common to express the hydraulic characteristics either in terms of a headloss coefficient K_L or as a discharge coefficient C_f where A_v is the area of the valve at any opening, and ΔH is the headloss defined for the valve. Frequently a discharge coefficient is defined in terms of the fully open valve area. The hydraulic coefficients embody not only the geometric features of the valve through A_v but also the flow characteristics.

Unless uniform flow is established far upstream and downstream of a valve in a pipeline the value of any of the coefficients can be affected by effects of nonuniform flow. It is not unusual for investigators to use only two pressure taps—one upstream and one downstream, frequently 1 and 10 diameters, respectively. The flow characteristics of valves in terms of pressure drop or headloss have been determined for numerous valves by many investigators and countless manufacturers. Only a few sets of data and typical curves will be presented here for ball, butterfly, and gate, ball, butterfly, and gate valves C_D. For a valve located in the interior of a long continuous pipe, as shown in Fig. 12.3, the presence of the valve disturbs the flow both upstream and downstream of the obstruction as reflected by the velocity distribution, and the pressure variation, which will be non—hydrostatic in the regions of nonuniform flow. Accounting for the pipe friction between

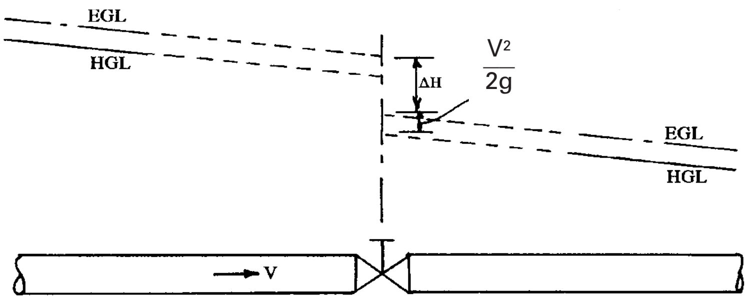


FIGURE 12.3 Definition of headloss characteristics of a valve.

upstream and downstream uniform flow sections, the headloss across the valve is expressed in terms of the pipe velocity and a headloss coefficient K_L

$$\Delta H = K_L \frac{V^2}{2g} \tag{12.4}$$

Often manufacturers represent the hydraulic characteristics in terms of discharge coefficients

$$Q = C_f A_{vo} \sqrt{2g\Delta H} = C_F A_{vo} \sqrt{2gH}\,, \tag{12.5}$$

where

$$H = \Delta H + \frac{V^2}{2g} \tag{12.6}$$

Both discharge coefficients are defined in terms of the nominal full-open valve area A_{vo} and a representative head, ΔH for C_f and H for C_Q, the latter definition generally reserved for large valves employed in the hydroelectric industry. The interrelationship between C_f, C_F, and K_L is

$$K_L = \frac{1}{C_f^2} = \frac{1 - C_F^2}{C_F^2} \tag{12.7}$$

Frequently valve characteristics are expressed in terms of a dimensional flow coefficient C_v from the valve industry

$$Q = C_v \sqrt{\Delta p} \tag{12.8}$$

where Q is in American flow units of gallons per minute (gpm) and Δp is the pressure loss in pounds per square inch (psi). In transient analysis it is convenient to relate either the loss coefficient or the discharge coefficient to the corresponding value at the fully open valve position, for which $C_f = C_{fo}$. Hence,

$$\frac{Q}{Q_o} = \frac{C_f}{C_{fo}} \sqrt{\frac{\Delta H}{\Delta H_o}} = \tau \sqrt{\frac{\Delta H}{\Delta H_o}} \tag{12.9}$$

Traditionally the dimensionless valve discharge coefficient is termed τ and defined by

$$\tau = \frac{C_f}{C_{fo}} = \frac{C_v}{C_{vo}} = \frac{C_f}{C_{fo}} = \sqrt{\frac{K_{Lo}}{K_L}} \tag{12.10}$$

12.3.4 Typical Geometric and Hydraulic Valve Characteristics

The geometric projected area of valves shown in Fig. 12.2 can be calculated for ball, butterfly, and gate valves using simple expressions. The dimensionless hydraulic flow coefficient τ is plotted in Fig. 12.4 for various valve openings for the three selected valves along with the area ratio for comparison. The lower diagram, which is based on hydraulic measurements, should be used for transient calculations rather than the upper one, which is strictly geometric.

12.3.5 Valve Operation

The instantaneous closure of a valve at the end of a pipe will yield a pressure rise satisfy-

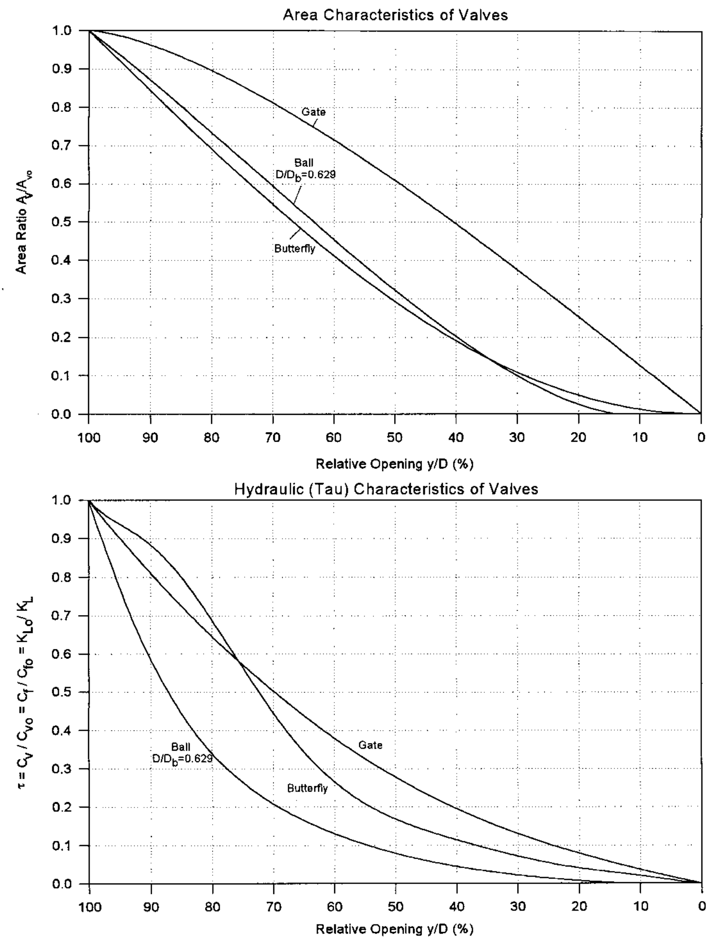


FIGURE 12.4 Geometric and hydraulic characteristics of typical control valves

ing Joukowsky's equation—Eq. (12.2) or Eq. (12.3). In this case the velocity difference $\Delta V = 0 - V_o$, where V_o is the initial velocity of liquid in the pipe. Although Eq. (12.2) applies across every wavelet, the effect of complete valve closure over a period of time greater than $2L/a$, where L is the distance along the pipe from the point of wave creation to the location of the first pipe area change, can be beneficial. Actually, for a simple pipeline the maximum head rise remains that from Eq. (12.3) for times of valve closure $t_c \leq 2L/a$, where L is the length of pipe. If the value of $t_c > 2L/a$, then there can be a considerable reduction of the peak pressure resulting from beneficial effects of negative wave reflections from the open end or reservoir considered in the analysis. The phenomenon can still be classified as waterhammer until the time of closure $t_c >> 2L/a$, beyond which time there are only inertial or incompressible deceleration effects, referred to as *surging*, also known as *rigid column analysis*. Table 12.2 classifies four types of valve closure, independent of type of valve.

TABLE 12.2 Classification of Valve Closure

Time of Closure t_c	*Type of Closure*	*Maximum Head* ΔH_{max}	*Phenomenon*
0	Instantaneous	aV_o/g	Waterhammer
$\leq 2L/a$	Rapid	aV_o/g	Waterhammer
$> 2L/a$	Gradual	$< aV_o/g$	Waterhammer
$>> 2L/a$	Slow	$<< aV_o/g$	Surging

Using standard waterhammer programs, parametric analyses can be conducted for the preparation of charts to demonstrate the effect of time of closure, type of valve, and an indication of the physical process—waterhammer or simply inertia effects of deceleration. The charts are based on analysis of valve closure for a simple reservoir-pipe-valve arrangement. For simplicity fluid friction is often neglected, a reasonable assumption for pipes on the order of hundreds of feet in length.

12.4 HYDRAULIC CHARACTERISTICS OF PUMPS

Transient analyses of piping systems involving centrifugal, mixed-flow, and axial-flow pumps require detailed information describing the characteristics of the respective turbomachine, which may pass through unusual, indeed abnormal, flow regimes. Since little if any information is available regarding the dynamic behavior of the pump in question, invariably the decision must be made to use the steady-flow characteristics of the machine gathered from laboratory tests. Moreover, complete steady-flow characteristics of the machine may not be available for all possible modes of operation that may be encountered in practice.

In this section steady-flow characteristics of pumps in all possible zones of operation are defined. The importance of geometric and dynamic similitude is first discussed with respect to both (1) homologous relationships for steady flow and (2) the importance of the assumption of similarity for transient analysis. The significance of the eight zones of operation within each of the four quadrants is presented in detail with reference to three possible modes of data representation. The steady-flow characteristics of pumps are discussed in detail with regard to the complete range of possible operation. The loss of driving power to a pump is usually the most critical transient case to consider for pumps, because of the possibility of low pipeline pressures which may lead to (1) pipe collapse due to buckling, or (2) the formation of a vapor cavity and its subsequent collapse. Other waterhammer problems may occur due to slam of a swing check valve, or from a discharge valve closing either too quickly (column separation), or too slowly (surging from reverse flow). For radial-flow pumps for which the reverse flow reaches a maximum just subsequent to passing through zero speed (locked rotor point), and then is decelerated as the shaft runs faster in the turbine zone, the head will usually rise above the nominal operating value. As reported by Donsky (1961) mixed-flow and axial-flow pumps may not even experience an upsurge in the turbine zone because the maximum flow tends to occur closer to runaway conditions.

12.4.1 Definition of Pump Characteristics

The essential parameters for definition of hydraulic performance of pumps are defined as

- *Impeller diameter.* Exit diameter of pump rotor D_I.
- *Rotational speed.* The angular velocity (rad/s) is ω, while $N = 2\ \pi\omega/60$ is in rpm.

- *Flow rate.* Capacity Q at operating point in chosen units.
- *Total dynamic head* (*TDH*). The total energy gain (or loss) H across pump, defined as

$$H = \left(\frac{P_d}{\gamma} + z_d\right) - \left(\frac{P_s}{\gamma} + z_s\right) + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} \tag{12.11}$$

where subscripts s and d refer to suction and discharge sides of the pump, respectively,

12.4.2 Homologous (Affinity) Laws

Dynamic similitude, or dimensionless representation of test results, has been applied with perhaps more success in the area of hydraulic machinery than in any other field involving fluid mechanics. Due to the sheer magnitude of the problem of data handling it is imperative that dimensionless parameters be employed for transient analysis of hydraulic machines that are continually experiencing changes in speed as well as passing through several zones of normal and abnormal operation. For liquids for which thermal effects may be neglected, the remaining fluid-related forces are pressure (head), fluid inertia, resistance, phase change (cavitation), surface tension, compressibility, and gravity. If the discussion is limited to single-phase liquid flow, three of the above fluid effects—cavitation, surface tension, and gravity (no interfaces within machine)—can be eliminated, leaving the forces of pressure, inertia, viscous resistance, and compressibility. For the steady or even transient behavior of hydraulic machinery conducting liquids the effect of compressibility may be neglected.

In terms of dimensionless ratios the three forces yield an *Euler number* (ratio of inertia force to pressure force), which is dependent upon geometry, and a Reynolds number. For all flowing situations, the viscous force, as represented by the Reynolds number, is definitely present. If water is the fluid medium, the effect of the Reynolds number on the characteristics of hydraulic machinery can usually be neglected, the major exception being the prediction of the performance of a large hydraulic turbine on the basis of model data. For the transient behavior of a given machine the actual change in the value of the Reynolds number is usually inconsequential anyway. The elimination of the viscous force from the original list reduces the number of fluid-type forces from seven to two–pressure (head) and inertia, as exemplified by the Euler number. The appellation geometry in the functional relationship in the above equation embodies primarily, first, the shape of the rotating impeller, the entrance and exit flow passages, including effects of vanes, diffusers, and so on; second, the effect of surface roughness; and lastly the geometry of the streamline pattern, better known as kinematic similitude in contrast to the first two, which are related to geometric similarity. *Kinematic similarity* is invoked on the assumption that similar flow patterns can be specified by congruent velocity triangles composed of peripheral speed U and absolute fluid velocity V at inlet or exit to the vanes. This allows for the definition of a *flow coefficien*t, expressed in terms of impeller diameter D_I and angular velocity ω:

$$C_Q = \frac{Q}{\omega D_I^3} \tag{12.12}$$

The reciprocal of the Euler number (ratio of pressure force to inertia force) is the *head coefficient,* defined as

$$C_H = \frac{gH}{\omega^2 D_I^2} \tag{12.13}$$

A *power coefficient* can be defined

$$C_P = \frac{P}{\rho\omega^3 D_I^5} \tag{12.14}$$

For transient analysis, the desired parameter for the continuous prediction of pump speed is the unbalanced torque T. Since $T = P/\omega$, the *torque coefficient* becomes

$$C_T = \frac{T}{\rho\omega^2 D_I^5} \tag{12.15}$$

Traditionally in hydraulic transient analysis to refer pump characteristics to so-called *rated conditions*—which preferably should be the optimum *or best efficiency point* (BEP), but sometimes defined as the *duty, nameplate*, or *design point*. Nevertheless, in terms of *rated conditions*, for which the subscript R is employed, the following ratios are defined;

$$\text{Flow: } v = \frac{Q}{Q_R} \qquad \text{speed: } \alpha = \frac{\omega}{\omega_R} = \frac{N}{N_R} \qquad \text{head: } h = \frac{H}{H_R} \qquad \text{torque: } \beta = \frac{T}{T_R}$$

Next, for a given pump undergoing a transient, for which D_I is a constant, Eqs. (12.12–12.15) can be written in terms of the above ratios

$$\frac{v}{\alpha} = \frac{C_Q}{C_{QR}} = \frac{Q_R}{Q}\frac{\omega}{\omega_R} \qquad \frac{h}{\alpha^2} = \frac{C_H}{C_{HR}} = \frac{H}{H_R}\frac{\omega_R^2}{\omega^2} \qquad \frac{\beta}{\alpha^2} = \frac{C_T}{C_{TR}} = \frac{T}{T_R}\frac{\omega_R^2}{\omega^2}$$

12.4.3 Abnormal Pump (Four–Quadrant) Characteristics

The performance characteristics discussed up to this point correspond to pumps operating normally. During a transient, however, the machine may experience either a reversal in flow, or rotational speed, or both, depending on the situation. It is also possible that the torque and head may reverse in sign during passage of the machine through abnormal zones of performance. The need for characteristics of a pump in abnormal zones of operation can best be described with reference to Fig. 12.5, which is a simulated pump power failure transient. A centrifugal pump is delivering water at a constant rate when there is a sudden loss of power from the prime move—rin this case an electric motor. For the postulated case of no discharge valves, or other means of controlling the flow, the loss of driving torque leads to an immediate deceleration of the shaft speed, and in turn the flow. The three curves are dimensionless head (h), flow (v), and speed (α). With no additional means of controlling the flow, the higher head at the final delivery point (another reservoir) will eventually cause the flow to reverse ($v < 0$) while the inertia of the rotating parts has maintained positive rotation ($\alpha > 0$). Up until the time of flow reversal the pump has been operating in the normal zone, albeit at a number of off-peak flows.

To predict system performance in regions of negative rotation and/or negative flow the analyst requires characteristics in these regions for the machine in question. Indeed, any peculiar characteristic of the pump in these regions could be expected to have an influence on the hydraulic transients. It is important to stress that the results of such analyses are critically governed by the following three factors: (1) availability of complete pump characteristics in zones the pump will operate, (2) complete reliance on dynamic similitude (homologous) laws during transients, and (3) assumption that steady-flow derived pumpcharacteristics are valid for transient analysis.

In vestigations by Kittredge (1956) and Knapp (1937) facilitated the understanding of

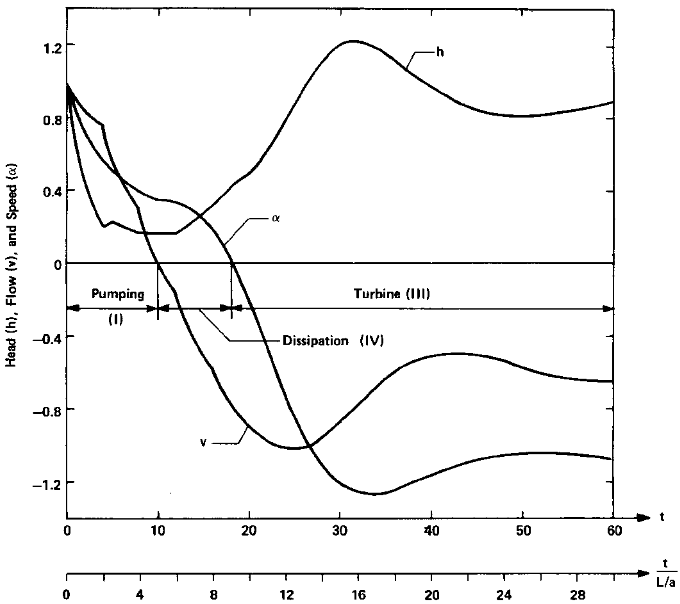


FIGURE 12.5 Simulated pump trip without valves in a single-pipeline system.

abnormal operation, as well as served to reinforce the need for test data. Following the work by Knapp (1941) and Swanson (1953), and a summary of their results by Donsky (1961), eight possible zones of operation, four normal and four abnormal, will be discussed here with reference to Fig. 12.6, developed by Martin (1983). In Fig. 12.6 the head H is shown as the difference in the two reservoir elevations to simplify the illustration. The effect of pipe friction may be ignored for this discussion by assuming that the pipe is short and of relatively large diameter. The regions referred to on Fig. 12.6 are termed zones and quadrants, the latter definition originating from plots of lines of constant head and constant torque on a flow-speed plane ($v - \alpha$ axes). Quadrants I ($v > 0$, $\alpha > 0$) and III ($v < 0$, $\alpha < 0$) are defined in general as regions of pump or turbine operation, respectively. It will be seen, however, that abnormal operation (neither pump nor turbine mode) may occur in either of these two quadrants. A very detailed description of each of the eight zones of operation is in order. It should be noted that all of the conditions shown schematically in Fig. 12.6 can be contrived in a laboratory test loop using an additional pump (or two) as the master and the test pump as a slave. Most, if not all, of the zones shown can also be experienced by a pump during a transient under the appropriate set of circumstances.

Quadrant I. Zone A (normal pumping) in Fig. 12.6 depicts a pump under normal operation for which all four quantities— Q, N, H, and T are regarded as positive. In this case $Q > 0$, indicating useful application of energy. Zone B (energy dissipation) is a condition of positive flow, positive rotation, and positive torque, but negative head—quite an abnormal condition. A machine could operate in Zone B by (1) being overpowered by another pump or by a reservoir during steady operation, or (2) by a sudden drop in head during a

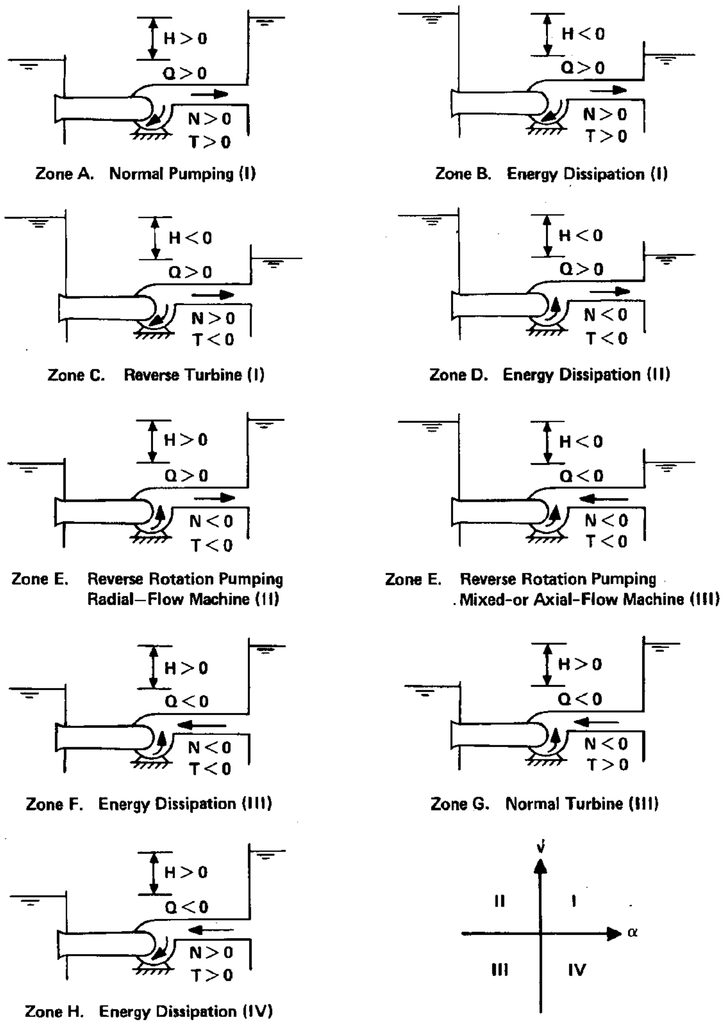


FIGURE 12.6 Four quadrants and eight zones of possible pump operation. (From Martin, 1983)

transient caused by power failure. It is possible, but not desirable, for a pump to generate power with both the flow and rotation in the normal positive direction for a pump, Zone C (reverse turbine), whichis caused by a negative head, resulting in a positive efficiency because of the negative torque. The maximum efficiency would be quite low due to the bad entrance flow condition and unusual exit velocity triangle.

Quadrant IV. Zone H, labeled energy dissipation, is often encountered shortly after a tripout or power failure of a pump, as illustrated in Fig. 12.5. In this instance the combined inertia of all the rotating elements—motor, pump and its entrained liquid, and shaft—has maintained pump rotation positive but at a reduced value at the time of flow reversal caused by the positive head on the machine. This purely dissipative mode results in a negative or zero efficiency. It is important to note that both the head and fluid torque are positive in Zone H, the only zone in Quadrant IV.

Quadrant III. A machine that passes through Zone H during a pump power failure will then enter Zone G (normal turbining) provided that reverse shaft rotation is not precluded by a mechanical ratchet. Although a runaway machine rotating freely is not generating power, Zone G is the precise mode of operation for a hydraulic turbine. Note that the head and torque are positive, as for a pump but that the flow and speed are negative, opposite to that for a pump under normal operation (Zone A).

Subsequent to the tripout or load rejection of a hydraulic turbine or the continual operation of a machine that failed earlier as a pump, Zone F (energy dissipation) can be encountered. The difference between Zones F and G is that the torque has changed sign for Zone F, resulting in a braking effect, which tends to slow the free=wheeling machine down. In fact the real runaway condition is attained at the boundary of the two zones, for which torque $T = 0$.

Quadrant II. The two remaining zones—D and E—are very unusual and infrequently encountered in operation, with the exception of pump/turbines entering Zone E during transient operation. Again it should be emphasized that both zones can be experienced by a pump in a test loop, or in practice in the event a machine is inadvertently rotated in the wrong direction by improper wiring of an electric motor. Zone D is a purely dissipative mode that normally would not occur in practice unless a pump, which was designed to increase the flow from a higher to lower reservoir, was rotated in reverse, but did not have the capacity to reverse the flow (Zone E, mixed or axial flow), resulting in $Q > 0$, $N < 0$, $T < 0$, for $H < 0$. Zone E, for which the pump efficiency > 0, could occur in practice under steady flow if the preferred rotation as a pump was reversed. There is always the question regarding the eventual direction of the flow. A radial-flow machine will produce positive flow at a much reduced capacity and efficiency compared to $N > 0$ (normal pumping), yielding of course $H > 0$. On the other hand, mixed and axial-flow machines create flow in the opposite direction (Quadrant III), and $H < 0$, which corresponds still to an increase in head across the machine in the direction of flow.

12.4.4 Representation of Pump Data for Numerical Analysis

It is conventional in transient analyses to represent h/α^2 and β/α^2 as functions of v/α, as shown in Fig. 12.7 and 12.8 for a radial-flow pump. The curves on Fig. 12.7 are only for positive rotation ($\alpha > 0$), and constitute pump Zones A, B, and C for $v > 0$ and the region of energy dissipation subsequent to pump power failure (Zone H), for which $v < 0$. The remainder of the pump characteris-tics are plotted in Fig. 12.8 for $\alpha < 0$. The complete characteristics of the pump plotted in Figs. 12.7 and 12.8 can also be correlated on what is known as a *Karman-Knapp circle diagram*, a plot of lines of constant head (h) and torque (β) on the coordinates of dimensionless flow (v) and speed (α). Fig. 12.9 is such a correlation for the same pump. The complete characteristics of the pump require six curves, three each for head and torque. For example, the h/α^2 curves from Figs. 12.7 and 12.8 can be represented by continuous lines for $h = 1$ and $h = -1$, and two straight lines through the origin for $h = 0$. A similar pattern exists for the torque (β) lines. In addition to the eight zones A–H illustrated in Fig. 12.6, the four Karman-Knapp quadrants in terms of v and, are well defined. Radial lines in Fig. 12.9 correspond to constant values for v/α in Figs. 12.7 and 12.8, allowing for relatively easy transformation from one form of presentation to the other.

In computer analysis of pump transients, Figs. 12.7 and 12.8, while meaningful from

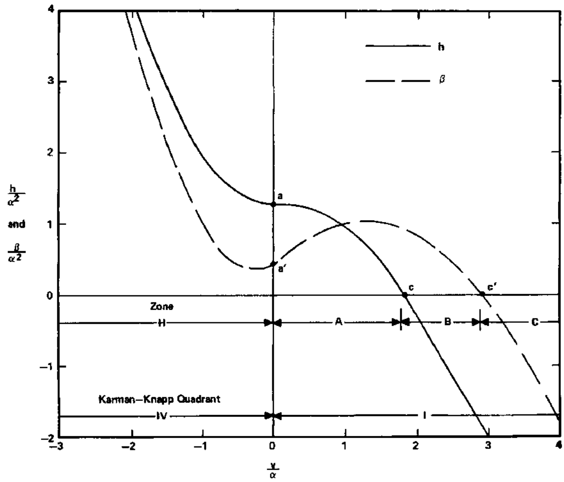


FIGURE 12.7 Complete head and torque characteristics of a radial-flow pump for positive rotation. (From Martin, 1983)

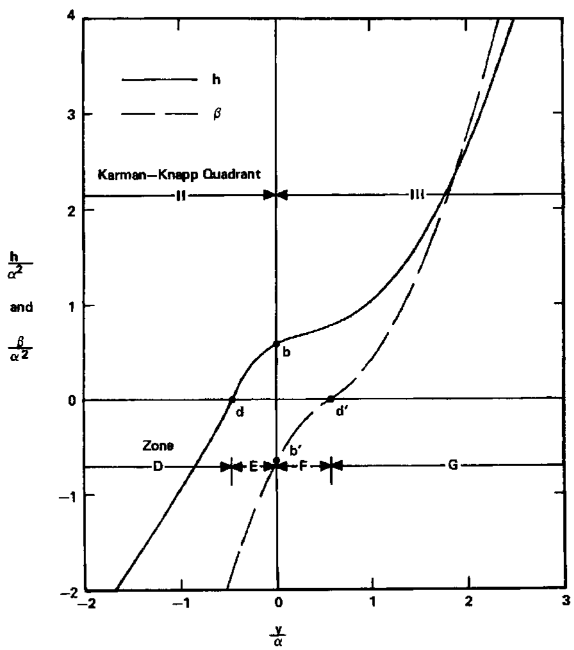


FIGURE 12.8 Complete head and torque characteristics of a radial-flow pump for negative rotation. (From Martin, 1983)

the standpoint of physical understanding, are fraught with the difficulty of $|v/\alpha|$ becoming infinite as the unit passes through, or remains at, zero speed ($\alpha = 0$). Some have solved that problem by switching from h/α^2 versus v/α to h/v^2 versus α/v, and likewise for β, for $|v/\alpha| > 1$. This technique doubles the number of curves on Figs. 12.7 and 12.8, and thereby creates discontinuities in the slopes of the lines at $|v/\alpha| = 1$, in addition to complicating the storing and interpolation of data. Marchal et al. (1965) devised a useful transformation which allowed the complete pump characteris-tics to be represented by two single curves, as shown for the same pump in Fig. 12.10. The difficulty of v/α becoming infinite was eliminated by utilizing the function $\tan^{-1}(v/\alpha)$ as the abscissa. The eight zones, or four quadrants can then be connected by the continuous functions. Although some of the physical interpretation of pump data has been lost in the transformation, Fig. 12.10 is now a preferred correlation for transient analysis using a digital computer because of function continuity and ease of numerical interpolation. The singularities in Figs. 12.7 and 12.8 and the asymptotes in Fig. 12.9 have now been avoided.

12.4.5 Critical Data Required for Hydraulic Analysis of Systems with Pumps

Regarding data from manufacturers such as pump curves (normal and abnormal), pump and motor inertia, motor torque-speed curves, and valve curves, probably the most critical for pumping stations are pump-motor inertia and valve closure time. Normal pump curves are

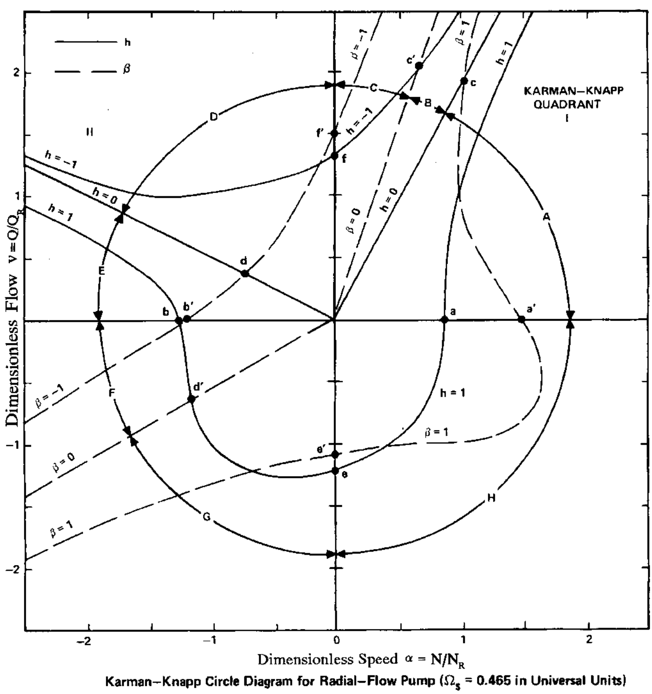


FIGURE 12.9 Complete four-quadrant head and torque characteristics of radial-flow pump. (From Martin, 1983)

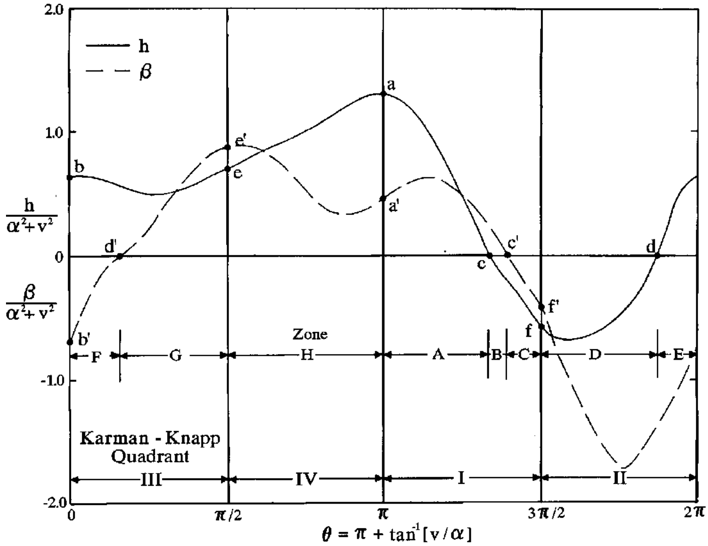


FIGURE 12.10 Complete head and torque characteristics of a radial-flow pump in Suter diagram. (From Martin, 1983)

usually available and adequate. Motor torque-speed curves are only needed when evaluating pump startup. For pump trip the inertia of the combined pump and motor is important.

12.5 SURGE PROTECTION AND SURGE CONTROL DEVICES

There are numerous techniques for controlling transients and waterhammer, some involving design considerations and others the consideration of surge protection devices. There must be a complete design and operational strategy devised to combat potential waterhammer in a system. The transient event may either initiate a low-pressure event (*downsurge*) as in the case of a pump power failure, or a high pressure event (*upsurge*) caused by the closure of a downstream valve. It is well known that a downsurge can lead to the undesirable occurrence of water-column separation, which itself can result in severe pressure rises following the collapse of a vapor cavity. In some systems negative pressures are not even allowed because of (1) possible pipe collapse or (2) ingress of outside water or air.

The means of controlling the transient will in general vary, depending upon whether the initiating event results in an upsurge or downsurge. For pumping plants the major cause of unwanted transients is typically the complete outage of pumps due to loss of electricity to the motor. For full pipelines, pump startup, usually against a closed pump discharge valve for centrifugal pumps, does not normally result in significant pressure transients. The majority of transient problems in pumping installations are associated with the potential (or realized) occurrence of *water-column separation* and *vapor-pocket collapse*, resulting from the tripout of one or more pumps, with or without valve action. The pump-discharge valve, if actuated too suddenly, can even aggravate the downsurge problem. To

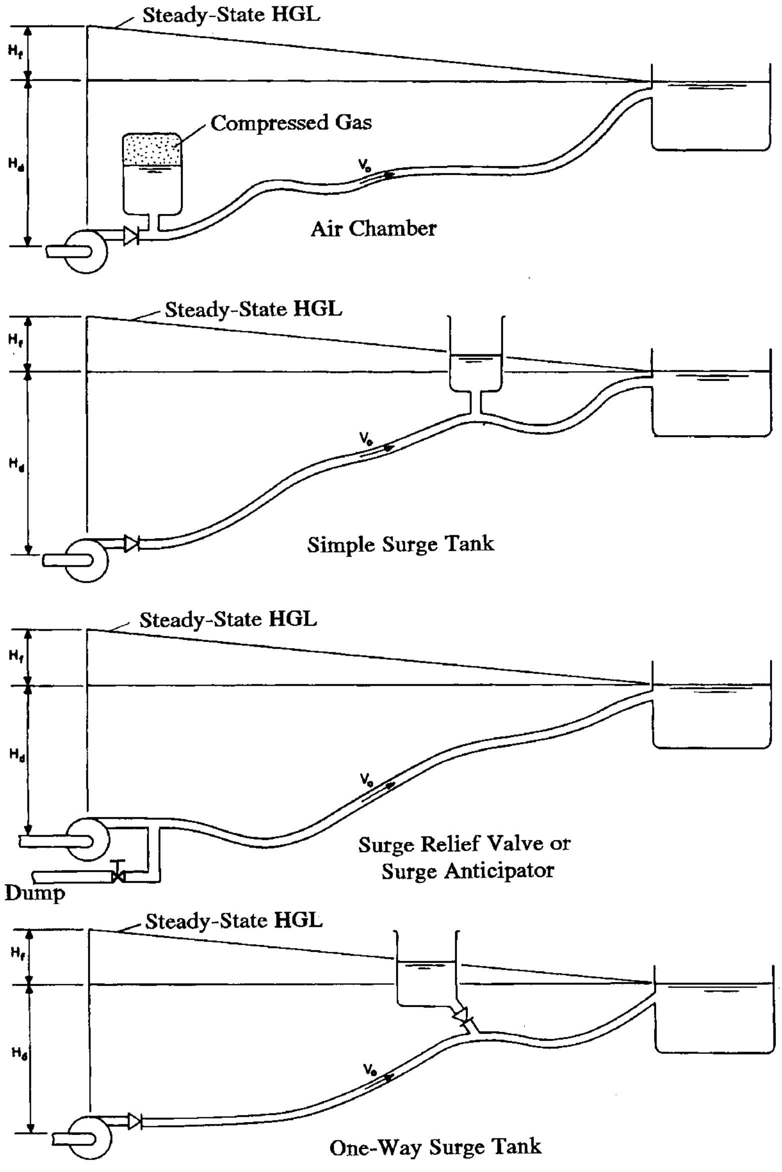


FIGURE 12.11 Schematic of various surge protection devices for pumping installations

combat the downsurge problem there are a number of options, mostly involving the design and installation of one or more surge protection devices. In this section various surge protection techniques will be discussed, followed by an assessment of the virtue of each with respect to pumping systems in general. The lift systems shown in Fig. 12.11 depict various surge protection schemes.

12.5.1 Critical Parameters for Transients

Before discussing surge protection devices, some comments will be made regarding the various pipeline, pump and motor, control valve, flow rate, and other parameters that affect the magnitude of the transient. For a pumping system the four main parameters are (1) pump flow rate, (2) pump and motor WR^2, (3) any valve motion, and (4) pipeline characteristics. The pipeline characteristics include piping layout—both plan and profile—pipe size and material, and the acoustic velocity. So-called short systems respond differently than long systems. Likewise, valve motion and its effect, whether controlled valves or check valves, will have different effects on the two types of systems.

The pipeline characteristics—item number (4)—relate to the response of the system to a transient such as pump power failure. Clearly, the response will be altered by the addition of one or more surge protection device or the change of (1) the flow rate, or (2) the WR^2, or (3) the valve motion. Obviously, for a given pipe network and flow distribution there are limited means of controlling transients by (2) WR^2 and (3) valve actuation. If these two parameters can not alleviate the problem than the pipeline response needs to be altered by means of surge protection devices.

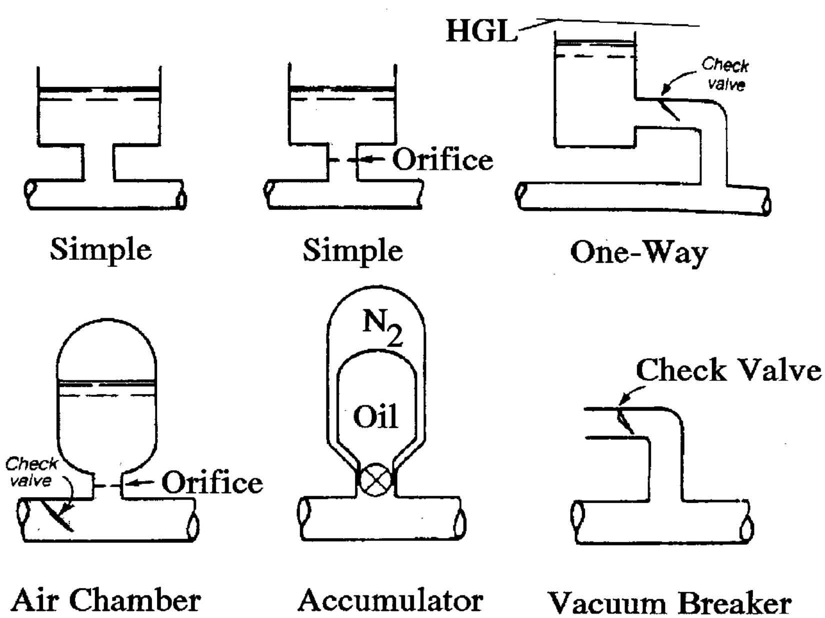
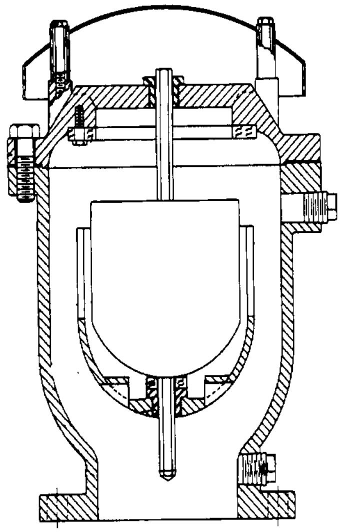
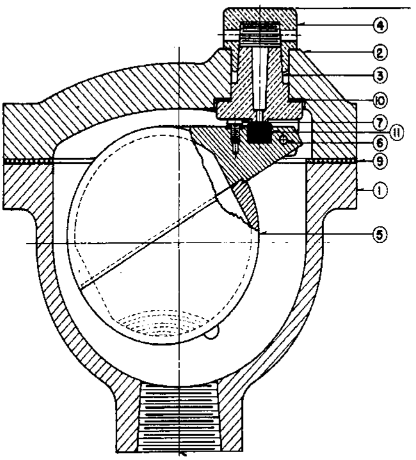


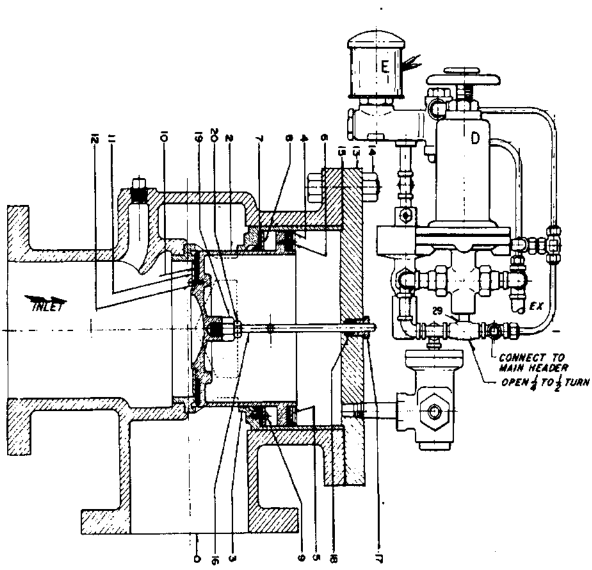
FIGURE 12.12 Cross sectional view of surge tanks and gas–n related surge protection devices



a. Vacuum Breaker Valve



a. Air Release Valve



c. Surge Relief or Surge Anticipator Valve

FIGURE 12.13 Cross sections of vacuum breaker, air release and surge relief valves.

12.5.2 Critique of Surge Protection

For pumping systems, downsurge problems have been solved by various combinations of the procedures and devices mentioned above. Details of typical surge protection devices are illustrated in Figs. 12.12 and 12.13. In many instances local conditions and preferences of engineers have dictated the choice of methods and/or devices. Online devices such as accumulators and simple surge tanks are quite effective, albeit expensive, solutions. One-way surge tanks can also be effective when judiciously sized and sited. Surge anticipation valves should not be used when there is already a negative pressure problem. Indeed, there are installations where surge anticipation functions of such valves have been deactivated, leaving only the surge relief feature. Moreover, there have been occasions for which the surge anticipation feature aggravated the low pressure situation by an additional downsurge caused by premature opening of the valve.

Regarding the consideration and ultimate choice of surge protection devices, subsequent to calibration of analysis with test results, evaluation should be given to simple surge tanks or standpipes, one-way surge tanks, and hydropneumatic tanks or air chambers. A combination of devices may prove to be the most desirable and most economical.

The admittance of air into a piping system can be effective, but the design of air vacuum-valve location and size is critical. If air may be permitted into pipelines careful analysis would have to be done to ensure effective results. The consideration of air-vacuum breakers is a moot point if specifications such as the Ten State Standards limit the pressures to positive values.

12.5.3 Surge Protection Control and Devices

Pump discharge valve operation. In gravity systems the upsurge transient can be controlled by an optimum valve closure—perhaps two stage, as mentioned by Wylie and Streeter (1993). As shown by Fleming (1990), an optimized closing can solve a waterhammer problem caused by pump power failure if coupled with the selection of a surge protection device. For pump power failure a control valve on the pump discharge can often be of only limited value in controlling the downsurge, as mentioned by Sanks (1989). Indeed, the valve closure can be too sudden, aggravating the downsurge and potentially causing column separation, or too slow, allowing a substantial reverse flow through the pump. It should also be emphasized that an optimum controlled motion for single-pump power failure is most likely not optimum for multiple-pump failure. The use of microprocessors and servomechanisms with feedback systems can be a general solution to optimum control of valves in conjunction with the pump and pipe system. For pump discharge valves the closure should not be too quick to exacerbate downsurge, nor too slow to create a substantial flow back through the valve and pump before closure.

Check valves. Swing check valves or other designs are frequently employed in pump discharge lines, often in conjunction with slow acting control valves. As indicated by Tullis (1989), a check valve should open easily, have a low head loss for normal positive flow, and create no undesirable transients by its own action. For short systems, a slow=responding check valve can lead to waterhammer because of the high reverse flow generated before closure. A spring or counterweight loaded valve with a dashpot can (1) give the initial fast response followed by (2) slow closure to alleviate the unwanted transient. The proper selection of the load and the degree of damping is important, however, for proper performance.

Check valve slam is also a possibility from stoppage or failure of one pump of several in a parallel system, or resulting from the action of an air chamber close to a pump undergoing power failure. Check valve slam can be reduced by the proper selection of a dashpot.

Surge anticipator valves and surge relief valves. A surge anticipation valve, Fig. 12.13c frequently installed at the manifold of the pump station, is designed to open initially under (1) pump power failure, or (2) the sensing of underpressure, or (3) the sensing of overpressure, as described by Lescovitch (1967). On the other hand, the usual type of surge relief valve opens quickly on sensing an overpressure, then closes slowly, as controlled by pilot valves. The surge anticipation valve is more complicated than a surge relief valve in that it not only embodies the relief function at the end of the cycle, but also has the element of anticipation. For systems for which water-column separation will not occur, the surge anticipation valve can solve the problem of upsurge at the pump due to reverse flow or wave reflection, as reported in an example by White (1942). An example of a surge relief valve only is provided by Weaver (1972). For systems for which water-column separation will not occur, Lundgren (1961) provides charts for simple pipeline systems.

As reported by Parmakian (1968,1982a-b) surge anticipation valves can exacerbate the downsurge problem inasmuch as the opening of the relief valve aggravates the negative pressure problem. Incidents have occurred involving the malfunctioning of a surge anticipation valve, leading to extreme pressures because the relief valve did not open.

Pump bypass. In shorter low-head systems a pump bypass line (Fig. 12.11) can be installed in order to allow water to be drawn into the pump discharge line following power failure and a downsurge. As explained by Wylie and Streeter (1993), there are two possible bypass configurations. The first involves a control valve on the discharge line and a check valve on the bypass line between the pump suction or wet well and the main line. The check valve is designed to open subsequent to the downsurge, possibly alleviating column separation down the main line. The second geometry would reverse the valve locations, having a control valve in the bypass and a check valve in the main line downstream of the pump. The control valve would open on power failure, again allowing water to bypass the pump into the main line.

Open (simple) surge tank. A simple on-line surge tank or standpipe (Fig. 12.11) can be an excellent solution to both upsurge and downsurge problems, These devices are quite common in hydroelectric systems where suitable topography usually exists. They are practically maintenance free, available for immediate response as they are on line. For pumping installations open simple surge tanks are rare because of height considerations and the absence of high points near most pumping stations. As mentioned by Parmakian (1968) simple surge tanks are the most dependable of all surge protection devices. One disadvantage is the additional height to allow for pump shutoff head. Overflowing and spilling must be considered, as well as the inclusion of some damping to reduce oscillations. As stated by Kroon et al. (1984) the major drawback to simple surge tanks is their capital expense.

One-way surge tank. The purpose of a one-way surge tank is to prevent initial low pressures and potential water-column separation by admitting water into the pipeline subsequent to a downsurge. The tank is normally isolated from the pipeline by one or more lateral pipes in which there ok one or more check valves to allow flow into the pipe if the HGL is lower in the pipe than the elevation of the water in the open tank. Under normal operating conditions the higher pressure in the pipeline keeps the check valve closed. The major advantage of a one-way surge tank over a simple surge tank is that it does not have to be at the HGL elevation as required by the latter. It has the disadvantage, however, on only combatting initial downsurges, and not initial upsurges. One-way surge tanks have been employed extensively by the U.S. Bureau of Reclamation in pump discharge lines, principally by the instigation of Parmakian (1968), the originator of the concept. Another example of the effective application of one-way surge tanks in a pumping system was reported by Martin (1992), to be discussed in section 12.9.1.

Considerations for design are: (1) location of high points or knees of the piping, (2) check valve and lateral piping redundancy, (3) float control refilling valves and water supply, and other appurtenances. Maintenance is critical to ensure the operation of the check valve(s) and tank when needed.

Air chamber (hydropneumatic surge tank). If properly designed and maintained, an air chamber can alleviate both negative and positive pressure problems in pumping systems. They are normally located within or near the pumping station where they would have the greatest effect. As stated by Fox (1977) and others, an air chamber solution may be extremely effective in solving the transient problem, but highly expensive. Air chambers have the advantage that the tank–sometimes multiple–can be mounted either vertically or horizontally. The principal criteria are available water volume and air volume for the task at hand.

For design, consideration must be given to compressed air supply, water level sensing, sight glass, drains, pressure regulators, and possible freezing. Frequently, a check valve is installed between the pump and the air chamber. Since the line length between the pump and air chamber is usually quite short, check valve slamming may occur, necessitating the consideration of a dashpot on the check valve to cushion closure.

The assurance of the maintenance of air in the tank is essential—usually 50 percent of tank volume, otherwise the air chamber can be ineffective. An incident occurred at a raw water pumping plant where an air chamber became waterlogged due to the malfunctioning of the compressed air system. Unfortunately, pump power failure occurred at the same time, causing water column separation and waterhammer, leading to pipe rupture.

Air vacuum and air release valves. Another method for preventing subatmospheric pressures and vapor cavity formation is the admittance of air from air—vacuum valves (vacuum breakers) at selected points along the piping system. Proper location and size of air—vacuum valves can prevent water-column separation and reduce waterhammer effects, as calculated and measured by Martin (1980). The sizing and location of the valves are critical, as stated by Kroon et al. (1984). In fact, as reported by Parmakian (1982a,—b) the inclusion of air-vacuum valves in a pipeline did not eliminate failures. Unless the air-vacuum system is properly chosen, substantial pressures can still occur due to the compression of the air during resurge, especially if the air is at extremely low pressures within the pipeline when admitted. Moreover, the air must be admitted quickly enough to be effective. Typical designs are shown in Fig. 12.13

As shown by Fleming (1990) vacuum breakers can be a viable solution. The advantage of an air-vacuum breaker system, which is typically less expensive than other measures such as air chambers, must be weighed against the disadvantages of air accumulation along the pipeline and its subsequent removal. Maintenance and operation of valves is critical in order for assurance of valve opening when needed. Air removal is often accomplished with a combined air—release air-vacuum valve. For finished water systems the admittance of air is not a normal solution and must be evaluated carefully. Moreover, air must be carefully released so that no additional transient is created.

Flywheel. Theoretically, a substantial increase in the rotating inertia (WR^2) of a pump-motor unit can greatly reduce the downsurge inasmuch as the machine will not decelerate as rapidly. Typically, the motor may constitute from 75 to 90 percent of the total WR^2. Additional WR^2 by the attachment of a flywheel will reduce the downsurge. As stated by Parmakian (1968), a 100 percent increase in WR^2 by the addition of a flywheel may add up to 20 percent to the motor cost. He further states that a flywheel solution is only economical in some marginal cases. Flywheels are usually an expensive solution, mainly useful only for short systems. A flywheel has the advantage of practically no maintenance, but the increased torque requirements for starting must be considered.

Uninterrupted power supply (UPS). The availability of large uninterrupted power supply systems are of potential value in preventing the primary source of waterhammer in pumping; that is, the generation of low pressures due to pump power failure. For pumping stations with multiple parallel pumps, a UPS system could be devised to maintain one or more motors while allowing the rest to fail, inasmuch as there is a possibility of maintaining sufficient pressure with the remaining operating pump(s). The solution usually is expensive, however, with few systems installed.

12.6 DESIGN CONSIDERATIONS

Any surge or hydraulic transient analysis is subject to inaccuracies due to incomplete information regarding the systems and its components. This is particularly true for a water distribution system with its complexity, presence of pumps, valves, tanks, and so forth, and some uncertainty with respect to initial flow distribution. The ultimate question is how all of the uncertainties combine in the analysis to yield the final solution. There will be offsetting effects and a variation in accuracy in terms of percentage error throughout the system. Some of the uncertainties are as follows.

The simplification of a pipe system, in particular a complex network, by the exclusion of pipes below a certain size and the generation of equivalent pipes surely introduces some error, as well as the accuracy of the steady-state solution. However, if the major flow rates are reasonably well known, then deviation for the smaller pipes is probably not too critical. As mentioned above incomplete pump characteristics, especially during reverse flow and reverse rotation, introduce calculation errors. Valve characteristics that must be assumed rather than actual are sources of errors, in particular the response of swing check valves and pressure reducing valves. The analysis is enhanced if the response of valves and pumps from recordings can be put in the computer model.

For complex pipe network systems it is difficult to assess uncertainties until much of the available information is known. Under more ideal conditions that occur with simpler systems and laboratory experiments, one can expect accuracies when compared to measurement on the order of 5 to 10 percent, sometimes even better. The element of judgment does enter into accuracy. Indeed, two analyses could even differ by this range because of different assumptions with respect to wave speeds, pump characteristics, valve motions, system schematization, and so forth. It is possible to have good analysis and poorer analysis, depending upon experience and expertise of the user of the computer code. This element is quite critical in hydraulic transients. Indeed, there can be quite different results using the same code.

Computer codes, which are normally based on the *method of characteristics* (MOC), are invaluable tools for assessing the response based of systems to changes in surge protection devices and their characteristics. Obviously, the efficacy of such an approach is enhanced if the input data and network schematization is improved via calibration. Computer codes have the advantage of investigating a number of options as well as optimizing the sizing of surge protection devices. The ability to calibrate a numerical analysis code to a system certainly improves the determination of the proper surge protection. Otherwise, if the code does not reasonably well represent a system, surge protection devices can either be inappropriate or under- or oversized.

Computer codes that do not properly model the formation of vapor pockets and subsequent collapse can cause considerable errors. Moreover, there is also uncertainty regarding any free or evolved gas coming out of solution. The effect on wave speed is known, but this influence can not be easily addressed in an analysis of the system. It is simply another possible uncertainty.

Even for complicated systems such as water distribution networks, hydraulic transient calculations can yield reasonable results when compared to actual measurements provided that the entire system can be properly characterized. In addition to the pump, motor, and valve characteristics there has to be sufficient knowledge regarding the piping and flow demands. An especially critical factor for a network is the schematization of the network; that is, how is a network of thousands of pipes simplified to one suitable for computer analysis, say hundreds of pipes, some actual and some equivalent. According to Thorley (1991), a network with loops tends to be more forgiving regarding waterhammer because of the dispersive effect of many pipes and the associated reflections. On the other hand, Karney and McInnis (1990) show by a simple example that wave superposition can cause amplification of transients. Since water distribution networks themselves have not been known to be prone to waterhammer as a rule, there is meager information as to simplification and means of establishing equivalent pipes for analysis purposes. Large municipal pipe networks are good examples wherein the schematization and the selection of pipes characterizing the networks need to be improved in orde to represent the system better.

12.7 NEGATIVE PRESSURES AND WATER COLUMN SEPARATION IN NETWORKS

For finished water transmission and distribution systems the application of 138 kPa (20 psig) as a minimum pressure to be maintained under all conditions should prevent column separation from occurring provided analytical models have sufficient accuracy. Although water column separation and collapse is not common in large networks, it does not mean that the event is not possible. The modeling of water column separation is clearly difficult for a complicated network system. Water column separation has been analytically modeled with moderate success for numerous operating pipelines. Clearly, not only negative pressures, but also water column separation, are unwanted in pipeline systems, and should be eliminated by installation of properly designed surge protection devices.

If the criterion of a minimum pressure of 138 kPa (20 psig) is imposed then the issue of column separation and air-vacuum breakers are irrelevant, except for prediction by computer codes. Aside from research considerations, column separation is simulated for engineering situations mainly to assess the potential consequences. If the consequences are serious, as they often are in general, either operational changes or more likely surge protection devices are designed to alleviate column separation. For marginal cases of column separation the accuracy of pressure prediction becomes difficult. If column separation is not to be allowed and the occurrence of vapor pressure can be adequately predicted, then the simulation of column separation itself is not necessary.

Some codes do not simulate water column separation, but instead only maintain the pressure at cavity location at vapor pressure. The results of such an analysis are invalid, if indeed an actual cavity occurred, at some time subsequent to cavity formation. This technique is only useful to know if a cavity could have occurred, as there can be no assessment of the consequences of column separation. The inability of any code to model water column separation has the following implications: (1) the seriousness of any column separation event, if any, can not be determined, and (2) once vapor pressure is attained, the computation model loses its ability to predict adequately system transients. If negative pressures below 138 kPa (20 psig) are not to be allowed the inability of a code to assess the consequences of column separation and its attendant collapse is admittedly not so serious. The code need only flag pressures below 138 kPa (20) psig and negative pressures, indicating if there is a need for surge protection devices.

The ability of any model to properly simulate water column separation depends upon a number of factors. The principal ones are

- Accurate knowledge of initial flow rates
- Proper representation of pumps, valves, and piping system
- A vapor pocket allowed to form, grow, and collapse
- Maintenance of vapor pressure within cavity while it exists
- Determination of volume of cavity at each time step
- Collapse of cavity at the instant the cavity volume is reduced to zero

12.8 TIME CONSTANTS FOR HYDRAULIC SYSTEMS

- Elastic time constant

$$t_e = \frac{2L}{a} \tag{12.16}$$

- Flow time constant

$$t_f = \frac{LV_o}{gH^o} \tag{12.17}$$

- Pump and motor inertia time constant

$$t_m = \frac{I\omega_R}{T_R} = \frac{I\omega_R^2}{\rho g Q_R H_R \eta_{RT}} \tag{12.18}$$

- Surge tank oscillation inelastic time constant

$$t_s = 2p\sqrt{\frac{L_T}{g}\frac{A_t}{A_T}} \tag{12.19}$$

12.9 CASE STUDIES

For three large water pumping systems with various surge protection devices waterhammer analyses and site measurements have been conducted. The surge protection systems in question are (1) one-way and simple surge tanks, (2) an air chamber, and (3) air-vacuum breakers.

12.9.1 Case Study with One-way and Simple Surge Tanks

A very large pumping station has been installed and commissioned to deliver water over a distance of over 30 kilometers. Three three-stage centrifugal pumps run at a synchronous speed of 720 rpm, with individual rated capacities of 1.14 m^3/sec, rated heads of 165 m, and rated power of 2090 kw. Initial surge analysis indicated potential water-column separation. The surge protection system was then designed with one-way and simple surge tanks as well as air-vacuum valves strategically located.

The efficacy of these various surge protection devices was assessed from site measurements. Measurements of pump speed, discharge valve position, pump flow rate, and pressure at seven locations were conducted under various transient test conditions. The site measurements under three-pump operation allowed for improvement of hydraulic transient calculations for future expansion to four and five pumps. Figure 12.14 illustrates the profile of the ground and the location of the three pairs of surge tanks. The first and second pair of surge tanks are of the one-way (feed tank) variety, while the third pair are simple open on-line tanks.

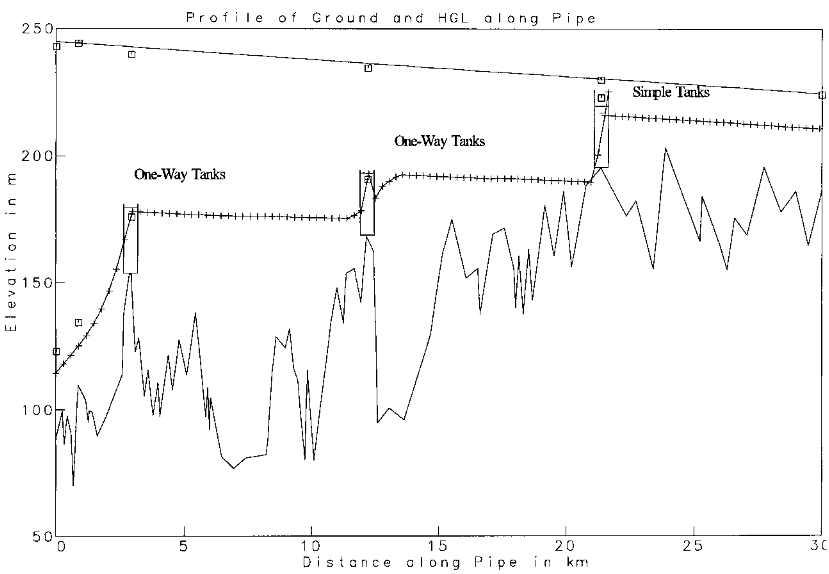
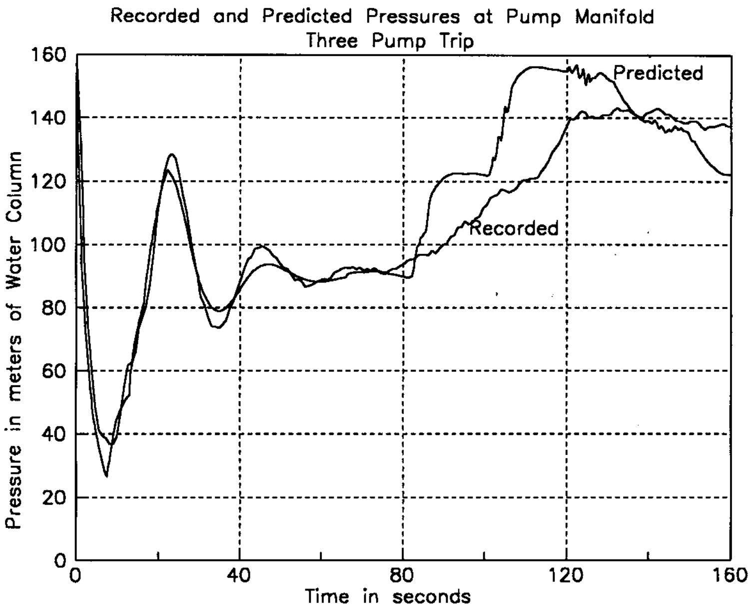


FIGURE 12.14 Case study of pump power failure at pumping station with three pair of surge tanks—two pair one way and one pair simple surge tanks Martin(1992).



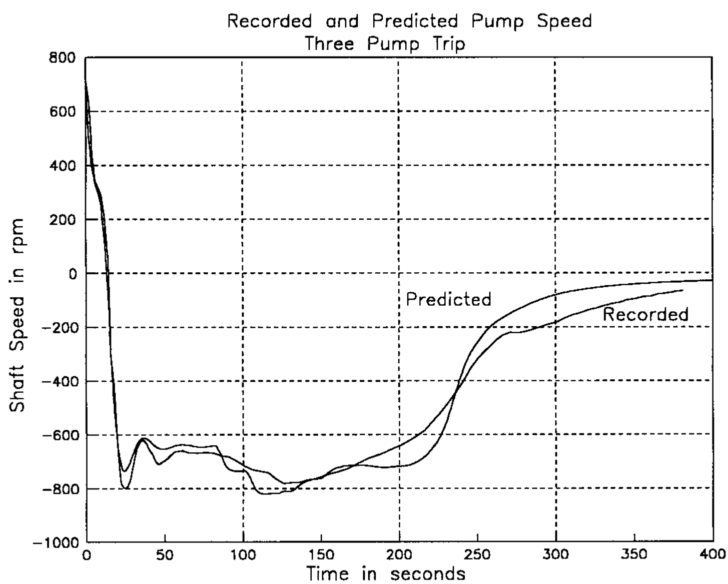
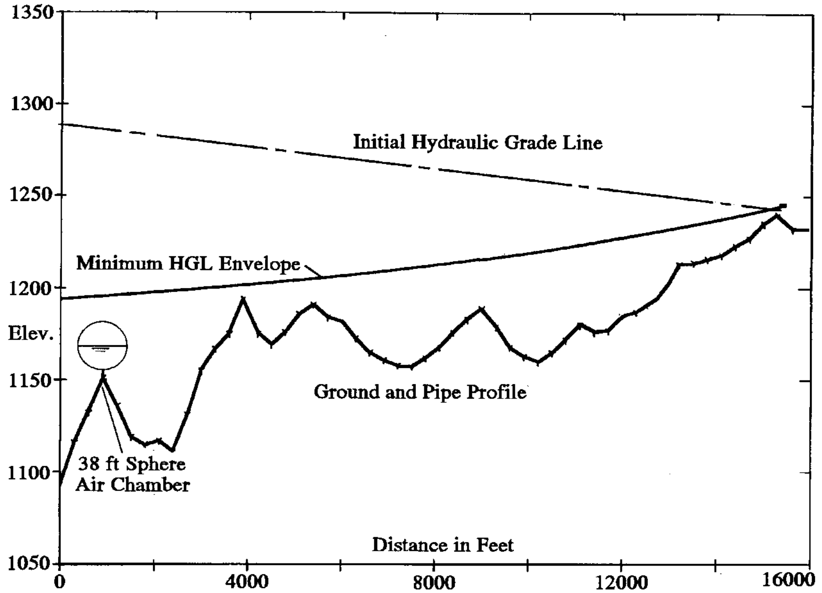


Figure 12.15. The test program and transient analysis clearly indicated that the piping system was adequately protected by the array of surge tanks inasmuch as there were no negative pressures.



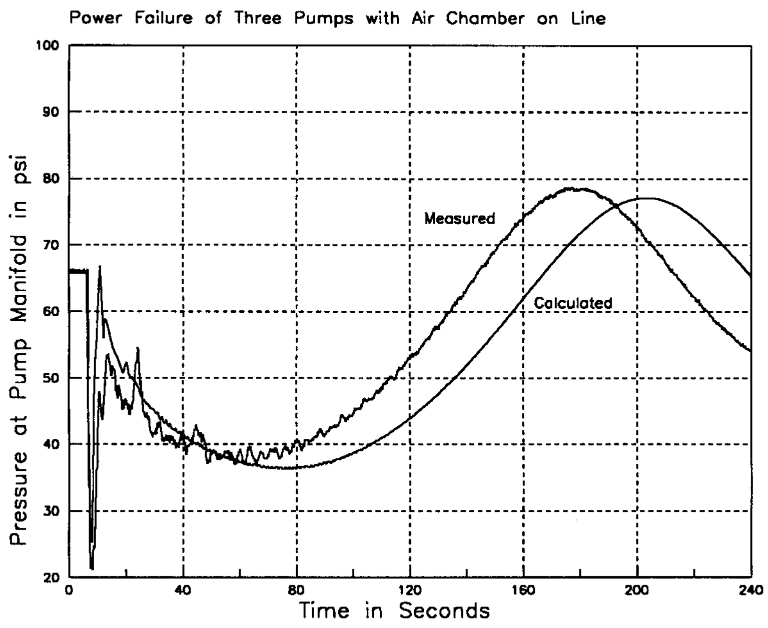


FIGURE 12.16 Case study of air chamber performance for raw water supply.

Pump trip tests were conducted for three-pump operation with cone valves actuated by the loss of motor power. For numerical analysis a standard computer program applying the method of characteristics was employed to simulate the transient events. Figure 12.15 shows the transient pressures for three pump power failure. The transient pressures agree reasonably well for the first 80 seconds. The minimum HGL's in Fig. 12.14 also show good agreement, as well as the comparison of measured and calculated pump speeds in

12.9.2 Case Study with Air chamber

Hydraulic transients caused by simultaneous tripping of pumps at the pumping station depicted on Fig. 12.16 were evaluated to assess the necessity of surge protection. Without the presence of any protective devices such as accumulators, vacuum breakers, or surge suppressors, water hammer with serious consequences was shown to occur due to depressurization caused by the loss of pumping pressure following sudden electrical outage. In the case of no protection a large vapor cavity would occur at the first high point above the pumping station, subsequently collapsing after the water column between it and the reservoir stops and reverses. This phenomenon, called water-column separation, can be mitigated by maintaining the pressures above vapor pressure.

The efficacy of the 11.6 m (38 ft) diameter air chamber shown in Fig. 12.16 was investigated analytically and validated by site measurements for three-pump operation. The envelope of the minimum HGL drawn on Fig. 12.16 shows that all pressures remained positive. The lower graph compares the site measurement with the calculated pressures obtained by a standard waterhammer program utilizing MOC.

12.9.3 Case Study with Air-vacuum Breaker

Air-inlet valves or air-vacuum breakers are frequently installed on liquid piping systems and cooling water circuits for the purpose of (1) eliminating the potential of water-column separation and any associated waterhammer subsequent to vapor pocket collapse; (2) protecting the piping from an external pressure of nearly a complete vacuum; and (3) providing an elastic cushion to absorb the transient pressures.

A schematic of the pumping and piping system subject to the field test program is shown in Fig. 12.17. This system provides the cooling water to a power plant by pumping water from the lower level to the upper reservoir level. There are five identical vertical pumps in parallel connected to a steel discharge pipe 1524 mm (60 in) in diameter. On the discharge piping of each pump there are 460 mm (18 in) diameter swing check valves. Mounted on top of the 1524 mm (60 in) diameter discharge manifold is a 200 mm (8 in) diameter pipe, in which is installed a swing check valve with a counter weight. Air enters the vacuum breaker through the tall riser, which extends to the outside of the pump house.

Transient pressures were measured in the discharge header for simultaneous tripout of three, four, and five pumps. The initial prediction of the downsurge caused by pump power failure was based on the method of characteristics with a left end boundary condition at the pumps, junction boundary condition at the change in diameter of the piping, and a constant pressure boundary condition at the right end of the system.

The predicted pressure head variation in the pump discharge line is shown in Fig. 12.17 for a simulated five pump tripout. The predicted peak pressure for the five pump

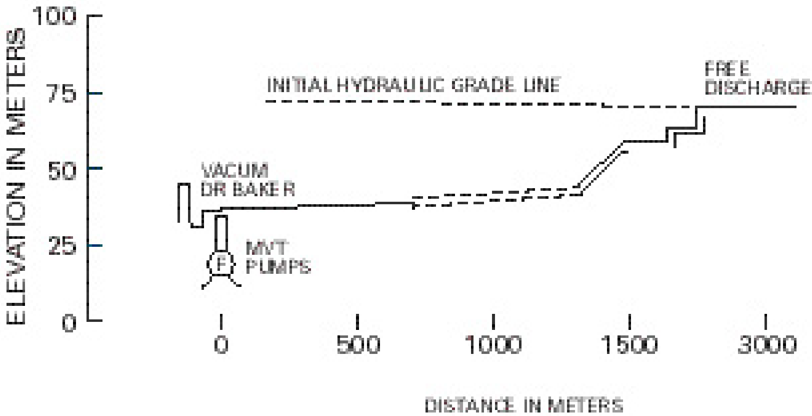


Figure 12.17. Case Study of Vacuum Breaker Performance for River Water System of Nuclear Plant, Martin (1980).

tripout compares favorably with the corresponding measured peak, but the time of occurrence of the peaks and the subsequent phasing vary considerably. Analysis without a vacuum breaker or other protective device in the system predicted waterhammer pressure caused by collapse of a vapor pocket to exceed 2450 kPa (355 psi). The vacuum breaker effectively reduced the peak pressure by 60 per cent. Peak pressures can be adequately predicted by a simplified liquid column, orifice, and air spring system. Water-column separation can be eliminated by air-vacuum breakers of adequate size.

REFERENCES

Chaudhry, M. H., *Applied Hydraulic Transients*, 2d ed., Van Nostrand Reinhold, New Yorle 1987.

Donsky, B., "Complete Pump Characteristics and the Effects of Specific Speeds on Hydraulic Transients," *Journal of Basic Engineering*, *Transactions,* American Society of Mechanical Engineers, 83: 685–699, 1961.

Fleming, A. J., "Cost-Effective Solution to a Waterhammer Problem," P*ublic Works*, 42—44, 1990.

Fox, J. A., *Hydraulic Analysis of Unsteady Flow in Pipe Networks*, John Wiley & Sons, New Yorle 1977.

Karney, B. W., and McInnis, D., "Transient Analysis of Water Distribution Systems," *Journal American Water Works Assoriation*, 82: 62–70, 1990.

Kittredge, C. P., "Hydraulic Transients in Centrifugal Pump Systems," *Transactions,* American Society of Mechanical Engineers, 78: 1307–1322, 1956.

Knapp, R. T., "Complete Characteristics of Centrifugal Pumps and Their Use in Prediction of Transient Behavior," *Transactions,* American Society of Mechanical Engineers , 59:683–689, 1937.

Knapp, R. T., "Centrifugal-Pump Performance Affected by Design Features," *Transactions,* American Societiy of Mecharnal Engineers, 63:251–260, 1941.

Kroon, J. R., Stoner, M. A., and Hunt, W. A., "Water Hammer: Causes and Effects," *Journal American Water Works Assoriation*, 76:39–45, 1984.

Lescovitch, J. E., "Surge Control of Waterhammer by Automatic Valves," *Journal American Water Works Assoriation*, 59:632-644, 1967.

Lundgren, C. W., "Charts for Determining Size of Surge Suppressor for Pump-Discharge Lines," *Journal of Engineering for Power*, *Transactions,* American Society of Mechanical Engineers, 93:43–47, 1961.

Marchal, M., Flesh, G., and Suter, P., "The Calculation of Waterhammer Problems by Means of the Digital Computer," *Proceedings, International Symposium on Waterhammer in Pumped Storage Projects*, American Society of Mechanical Engineers (ASME), Chicago, 1965.

Martin, C. S., "Entrapped Air in Pipelines," Paper F2 Second BHRA *International Conference on Pressure Surges*, The City University, London, September 22–24, 1976.

Martin, C. S., "Transient Performance of Air Vacuum Breakers," Fourth International Conference on Water Column Separation, Cagliari, November 11–13, 1979. *"Transient Performance Air Vacuum Breakere," L'Energia Elettrica*, *Proceedings No. 382*, 1980, pp. 174–184.

Martin, C. S., "Representation of Pump Characteristics for Transient Analysis," *ASME Symposium on Performance Characteristics of Hydraulic Turbines and Pumps*, Winter Annual Meeting, Boston, November 13–18, pp. 1–13,1983.

Martin, C. S., "Experience with Surge Protection Devices," *BHr Group International Conference on Pipelines*, Manchester, England, March pp. 171–178 24–26, 1992.

Martin, C. S., "Hydraulics of Valves," in J. A. Schetz and A. E. Fuhs, eds. *Handbook of Fluid Dynamics and Fluid Machinery*, Vol. III, McGraw-Hill, pp. 2043–2064. 1996,

Parmakian, J., *Water Hammer Analysi*s, Prentice-Hall, New York, 1955.

Parmakian, J., "Unusual Aspects of Hydraulic Transients in Pumping Plants," *Journal of the Boston Society of Civil Engineers*, 55:30–47, 1968.

Parmakian, J., "Surge Control," in M. H. Chaudhry, ed., *Proceedings*, *Unsteady Flow in Conduits*, Colorado State University, pp. 193–207 1982,

Parmakian, J., "Incidents, Accidents and Failures Due to Pressure Surges," in M. H. Chaudhry ed., *Proceedings*, *Unsteady Flow in Conduits*, Colorado State University, pp. 301–311 1982.

Sanks, R. L., *Pumping Station Design*, Butterworths, Bestar, 1989.

Stepanoff, A. I., *Centrifugal and Axial Flow Pumps*, John Wiley & Sons, New York, 1957.

Swanson, W.M., "Complete Characteristic Circle Diagrams for Turbomachinery," *Transactions,* American Society of Mechanical Engineers, 75:819–826, 1953.

Thorley, A. R. D., *Fluid Transients in Pipeline Systems*, D. & L. George Ltd., 1991.

Tullis, J. P., *Hydraulics of Pipelines*, John Wiley & Sons, New Yorh1989.

Watters, G. Z., *Modern Analysis and Control of Unsteady Flow in Pipelines*, Ann Arbor Science, Ann Arbor, MI, 1980.

Weaver, D. L., "Surge Control," *Journal American Water Works Association*, 64: 462–466, 1972.

White, I. M., "Application of the Surge Suppressor in Water Systems," *Water Works Engineering*, 45,304-306, 1942.

Wood, D.J., and Jones, S.E., "Waterhammer Charts for Various Types of Valves", ASCE, *Journal of Hydraulics Division*, HY1, 99:167-178, 1973.

Wylie, E. B., and Streeter, V. L., *Fluid Transients in System*s, Prentice-Hall, 1993.