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TUTORIAL: WATER HAMMER AND PIPING STRESSES

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ABSTRACT

This tutorial provides guidance to help understand the causes, and to prevent or mitigate the effects of water hammer, or fluid transients, in piping and pipeline systems. The text of this paper is based on early drafts of ASME B31D, which is still under consideration (The American Society of Mechanical Engineers, ASME Code for Pressure Piping, "ASME B31D, Design of Piping Systems for Dynamic Loads from Fluid Transients"). Drafts of B31D were, in turn, based on an ASME Press text book, titled "Fluid Mechanics, Water Hammer, Dynamic Stresses, and Piping Design" which was written by this author. The information presented here is not necessarily new, but it is provided in a format to provide an overview of fluid transient topics that are important to practicing engineers who work with piping systems. The essentials of water hammer are addressed in this tutorial, but piping failures are further complicated by fluid structure interactions. In particular, a dynamic load factor (DLF) expresses the fact that the maximum stresses due to water hammer are multiples of the stresses caused by statically applied loads. While of great importance, DLF"s are not considered in detail in this tutorial.

INTRODUCTION

Common system damages due to fluid transients, or water hammer, include piping fatigue, fracture of brittle components and piping, plastic deformation of ductile piping, and small to large leaks of piping, valves, flanges, and components. The extent of water hammer damages is immense, where more than 250,000 piping failures occur per year in U.S. and Canada water main piping alone (See Appendix A for a discussion of water main failures). The diversity of water hammer damages extends across many industries; from oil pipelines, to chemical plants, and to fires and explosions in nuclear power plants.

Applied loads on mechanical systems, such as piping systems are amplified, or magnified, when dynamically applied. The load amplification, or dynamic load factor, reflects the equivalent load on a piping system component when that load is applied in any manner other than a constant load. The DLF equals the ratio of the equivalent dynamic load divided by the static load that would occur under static load conditions. The use of DLF's with respect to water hammer induced loads is the study of this tutorial. Induced loads are caused by pressure changes in the system, which have different methods of application.

Sudden Pressure Changes Due to Fluid Transients

Sudden pressure changes on an elbow can double the bending stresses that would result from a constant pressure applied to that same elbow (DLF < 2). Multiple elbows in a piping system further complicate the determination of a DLF. Sudden pressure changes can quadruple the hoop stresses that would be caused by a constant pressure applied to the inside of a pipe (DLF < 4).

Gradual Pressure Changes

Gradual pressure changes reduce the dynamic effects of a suddenly applied pressure. If the pressure is applied slowly enough, the applied pressure is equivalent to a static pressure (DLF = 1).

Cyclic Pressure Changes, Resonance

Cyclic pressure changes may induce resonance, which may significantly increase the dynamic effects, where the DLF can be multiples of the expected static pressure condition. Resonance occurs when the frequency of the applied pressure is nearly equal to the frequency of any one of the piping natural frequencies or attached components, such as instrumentation or pumps. Cyclic pressures may be induced by some positive displacement pump designs, surge tank designs, and accumulator designs.

Acoustic Pressures

Acoustic pressure changes result in resonance, but the pressure magnitudes have values similar to sound wave pressures. Acoustic pressures and cyclic pressures are similar except for magnitudes.

Scope of This Tutorial

This tutorial consists of three parts to present an overview of water hammer (fluid transients), pipe stresses, and pipe damages, which are caused by pressure waves that travel at near sonic velocities along the bores of pipes when valves and pumps are operated to change flow rates in systems. To consider the different types of water hemmer, Part I provides definitions for this tutorial. Part II describes typical continuous operations. Part III of this tutorial presents different types of water hammer. Part IV presents simplified methods to calculate water hammer pressures. Finally, Part V presents simplified methods to determine pipe stresses.

PART I, TERMS AND DEFINITIONS

accumulator, a closed tank that typically contains a condensable gas and bladder, which maintains pressure and / or hinders vapor collapse in piping systems.

acoustic resonance, vibrations caused by pressure pulsations.

acoustic velocity, speed of sound in a fluid

cavitation, vapor bubbles form in the flow stream when velocities increase and pressures are consequently lowered to the vapor pressure of the fluid – low temperature boiling

cavitation erosion, all materials are pitted due when cavitation bubbles implode near the surface of a pipe, pump, or instrument.

centrifugal pump, a pump that converts energy to velocity using a rotating impeller. A typical example is a pump with rotating, radial vanes.

check valve, a valve which permits flow in only one direction.

choked flow, a condition where a maximum flow through a restriction that is caused when the flowing gas reaches its sonic velocity.

dead head pressure, static pressure caused by a pump when a downstream valve is closed to prevent flow through the operating pump.

design pressure, the maximum permitted normal operating pressure as defined in the applicable B31 Code.

dynamic load factor, a factor used to multiply static loads so they may be used in a structural piping system analysis to approximate the maximum forces, strains and stresses that result due to the application of time varying loads.

fluid transient, flow rate, density, and pressure changes in a piping system caused by changes in operating conditions.

head, specific energy at any point in a piping system typically measured in feet (or meters) of water.

hydraulic grade line, the height of fluid located at each point in a piping system.

load, applied force, pressure, or displacement.

method of characteristics, a computer simulation which uses a finite difference technique by dividing the pipe system into a number of reaches, and then solving equations of motion and continuity simultaneously.

modulus of elasticity, linear relationship between stress and strain in a mechanical body such as piping or valves, which is typically measured in a tensile test.

operating pressure, system operating pressure at the onset of a fluid transient.

pipe losses, See system resistance

plastic deformation, permanent deformation due to an applied load when the yield strength is exceeded.

pipe losses, See system resistance

positive displacement pump, a pump that converts energy to pressure by compressing a liquid. A typical example is a piston pump.

pump shut-down, de-energizing the pump.

relief valve, valve used to limit pressure in liquid filled systems.

run-out, free flow condition of a pump without discharge piping.

safety valve, valve used to limit pressure in gas or vapor filled systems.

surge tank, a tank open to atmosphere, which maintains pressures and / or hinders vapor collapse in piping systems.

system resistance, frictional resistance to flow through valves, piping, and other components.

water hammer, a term used to describe fluid transients, when banging sounds are sometimes heard as check valves slam shut or piping strikes adjacent structures. The terms water hammer and fluid transient are frequently used interchangeably, but fluid transient is a generic technical term, since it describes any pressure or flow change in a system.

wave speed, the speed of a pressure wave inside fluid filled piping.

yield strength (yield stress), onset of plastic deformation in a material as defined by the applicable Code.

PART II, CONTINUOUS OPERATIONS

Many, but not all, types of water hammer are briefly considered here, where all pressure changes are transmitted throughout piping systems at speeds less than the acoustic velocity of the fluid in the piping. To introduce water hammer types, continuous operations are first considered.

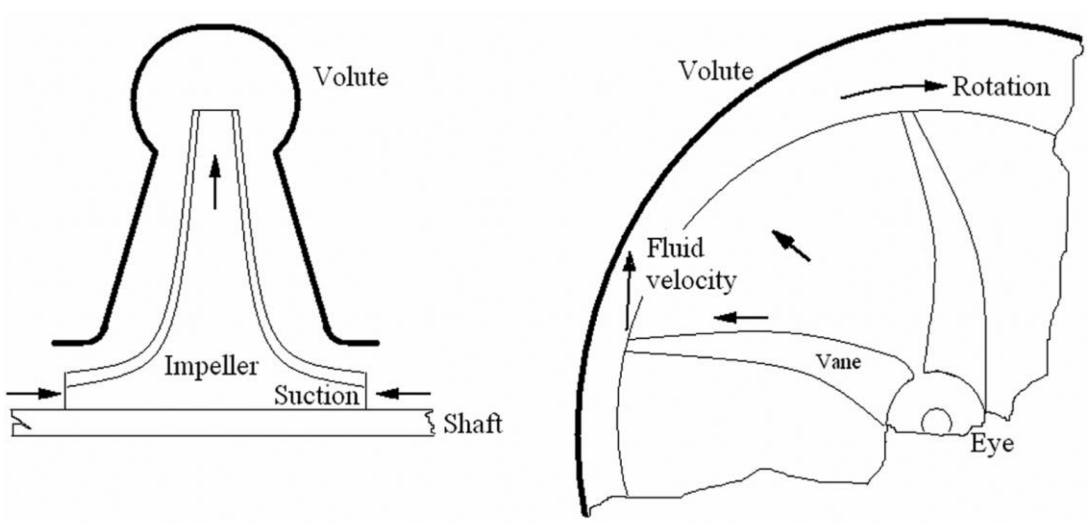

FIGURE 1: RADIAL PUMP DESIGN (Leishear [1 and 2])

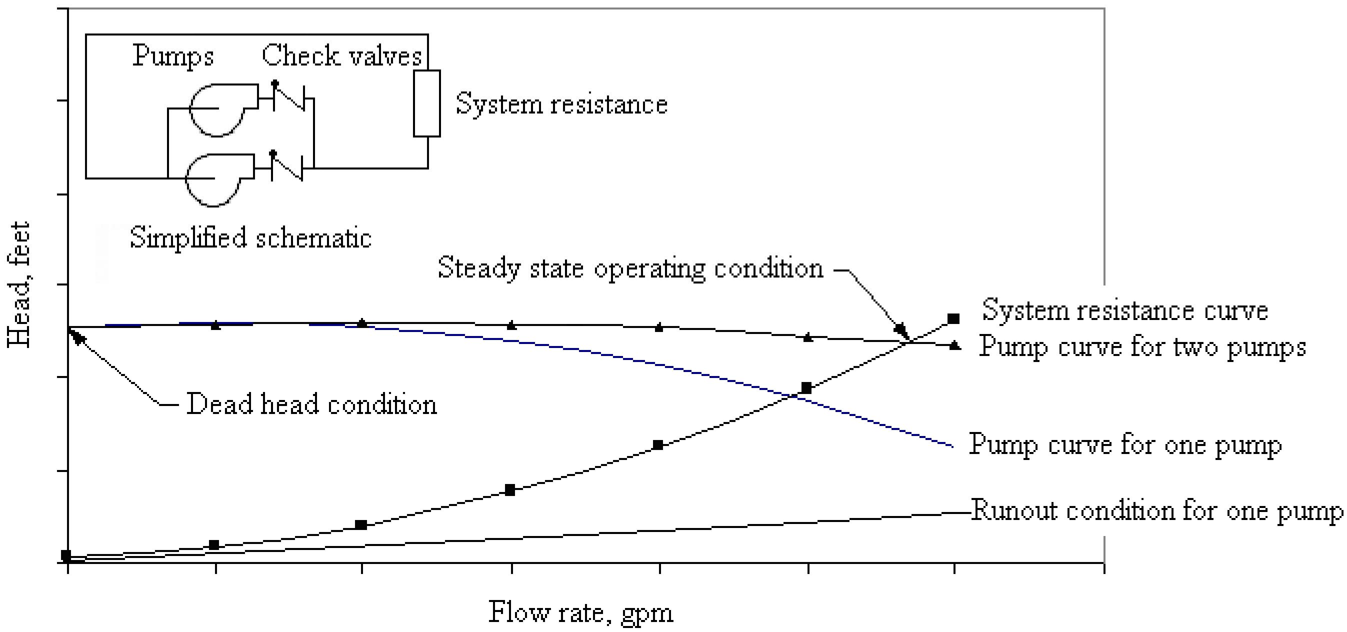

FIGURE 2: TYPICAL PUMP OPERATIONS (Leishear [1 and 2])

PART II, CONTINUOUS OPERATIONS

During routine, or continuous, operations numerous processes occur that affect piping stresses, where these processes may not be typically considered as fluid transients. However, these processes and supporting equipment are fundamental with respect to fluid flow and piping stresses.

Dynamic Pumps

Dynamic pumps require continuous energy during operations to overcome system head. These pumps include radial, axial and mixed flow designs.

Centrifugal Pumps

Centrifugal pumps provide required flow rates, while overcoming piping system resistance and changes in elevation. These pumps may be radial, axial or mixed flow designs.

Radial Flow Pumps

Radial flow pumps contain rotating vanes that accelerate the fluid in the pump, so that the energy from this acceleration may be transformed into pressure heaad for system operations. See Fig. 1 and 2.

Axial and Mixed Flow Pumps

An axial flow pump is similar to an aircraft propeller, and mixed flow pumps are hybrids between radial and axial flow pumps. Axial flow pumps have much higher dead head pressures than radial pumps, and occasionally require a bypass line around the pump to prevent piping overpressure.

Jet Pumps

Jet pumps operate due to applied steam or gas (frequently air) motive forces. See Fig. 3.

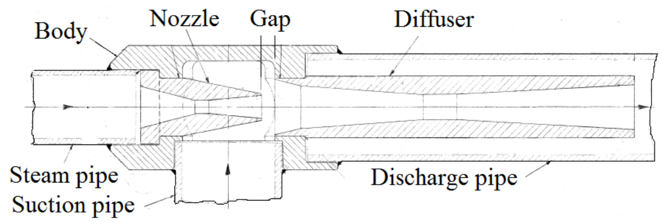

FIGURE 3: TYPICAL JET PUMP DESIGN (Leishear [1 and 2])

Positive Displacement Pumps

Positive displacement pumps are operated by periodic energy sources applied to their internal components. There are numerous designs, which are separated into two categories: reciprocating and rotating.

Reciprocating Pumps

Reciprocating pumps may be piston-cylinder designs or may contain internal diaphragms and valve ports to control flow. With respect to fluid transients a common design problem is the fact that pressure pulsations and resonance frequently result from their use.

Rotating Pumps

There are many rotating designs, i.e., vane, piston, peristaltic, gear, lobe, and screw. Cyclic pressure surge is not a problem in many designs, and where cyclic pressures occur in lobe, gear, and peristaltic pumps the pressures are typically low enough to not cause piping failures.

Compressors

Compressor designs are divided into basic types: reciprocating, piston-cylinder designs and centrifugal.

Reciprocating Compressors

Reciprocating compressors may cause pressure pulsations, which may lead to resonant damages to compressor components, piping, and system components. Additionally, resonant compressor vibrations may cause damages as well.

Centrifugal Compressors

High frequency rotations of centrifugal compressors generally do not cause detrimental pressure pulsations, but resonant vibrations may damage system components. In parallel compressor installations, gas flow may surge back and forth between compressors and affect gas delivery to the system.

Control Valves

Control valves stop, start, or adjust fluid flow. There are many control valve designs, e.g., gate, globe, ball, diaphragm, butterfly, check, needle, and pinch valves. With respect to water hammer, primary valve characteristics concern valve closure performance (See Fig. 4). Valve closure performance may affect the magnitude of fluid transient pressures when valves are closed.

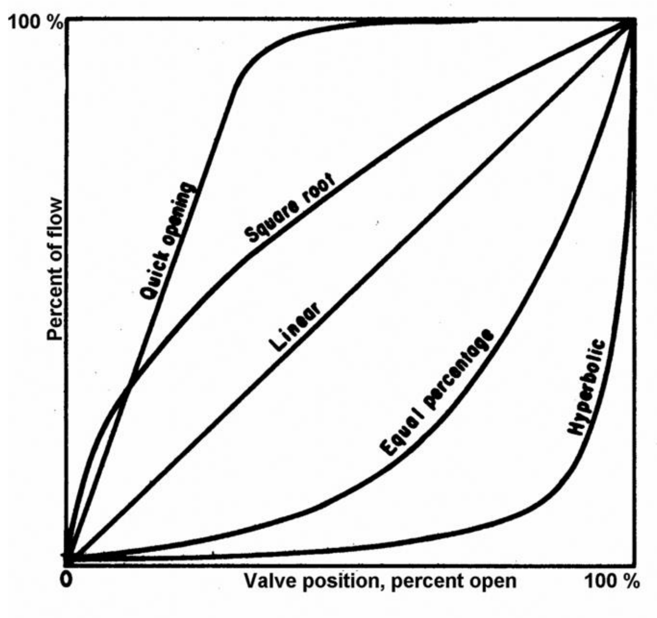

FIGURE 4: VALVE CLOSURE PERFORMANCE (ISA [3])

Pressure Regulators

Pressure regulators cause fluid transients as system pressures are changed by these valves. High pressure let down is an example of regulator use in a gas filled system to reduce system pressures. At low flow rates in both liquid and gas systems, pressure regulator operation becomes unstable, and downstream pressures oscillate to create resonant pressure conditions in the system. Pilot valves or small diameter bypass lines around the regulator are used to prevent this condition and better control pressure transients during pressure adjustments.

Relief Valves and Safety Valves

Relief valves prevent overpressures in liquid filled systems, and safety valves prevent overpressures in gas or vapor filled systems, where the high energies in vapor (such as steam) and gas systems pose safety risks. Relief valves and safety valves do not inhibit sudden pressure surges during their first pass through the piping.

Orifices

Orifices are frequently installed in piping for flow control or measurements. High velocities through the orifice opening, or openings, can cause cavitation, vibrations, or whistling in the piping.

Cavitation

Cavitation occurs at locations in the system where local velocities are increased, e.g., pumps, valves, reducers, and elbows.

Cavitation Erosion

Cavitation erosion occurs downstream of the locations where cavitation bubbles form. These bubbles collapse and damage piping and components where the fluid velocity reduces in the piping at locations such as on the surfaces of pump impellers and pump components, and downstream of valves, orifices, reducers, elbows, increasing pipe diameters, and instrumentation.

Acoustic Fatigue, High Frequency Vibration

During operation of choked-flow gas valves, significant levels of high-frequency noise can be generated, leading to excessive vibration and ultimately, to an acoustic fatigue failure of the associated downstream pipework.
Typical sources of acoustic noise include pressure relief and safety valves, blowdown (orifice plate) valves, pressure regulators, choke valves, gas compressor control valves. The consequences of an acoustic fatigue failure (also known as acoustic-induced vibration) are severe, from both a safety standpoint (gas explosion) and production viewpoint (loss of systems).

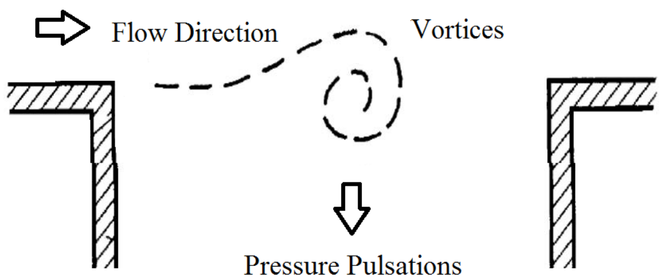

FIGURE 5: VORTEX SHEDDING

Vortex Shedding

Pressure pulsations result from vortex shedding inside piping. Vortex shedding may result in pressure pulsations that cause acoustic resonance. See Fig. 5.

Two Phase Flow

Two phase flow, thermal-fluid transients, are complicated by the fact that the flow rate and fluid properties dictate the occurrence of stratified flow, wavy flow, bubbly flow, slug flow, or annular flow where the liquid phase attaches to the pipe wall.

Pat III, TYPES OF WATER HAMMER

Pressure surges during transient operations occur when flow rates are changed in any piping system containing vapors, gasses, liquids, or combinations of these fluids.

Gas and Vapor Fluid Transients

As valves are opened in pipelines containing pressurized gasses or vapors upstream of those valves, pressure transients occur in the downstream piping. These transients are further complicated by the fact that sonic velocities may occur at the valve or in the piping, and fluid densities and temperatures vary significantly during these transients.

Fluid Transients in Liquid Filled Systems

During all fluid transients, pressure waves are transmitted through the fluids. In liquid filled systems, these transients typically occur when valves and pumps are operated.

Control Valve Operations

Fluid transients due to valve operations cause pressure surges of varying magnitudes, which depend on flow rates, piping dimensions, piping materials, the types of fluid, and the opening or closing speeds of valves.

Rapid Valve Closures or Openings

An example of sudden pressure changes, suddenly closed valves may cause high system pressures. Note that the resultant pressure due to this applied pressure increase is multiplied times the DLF to determine the pressure effects on maximum piping stresses. Suddenly opening valves causes pressures lower than those caused by sudden closing. See Fig. 6. Sudden valve openings also cause increased pressures in the system. Note that a finite difference computer codes, referred to as the Method of Characteristics (MOC) are typically used for fluid transient calculations.

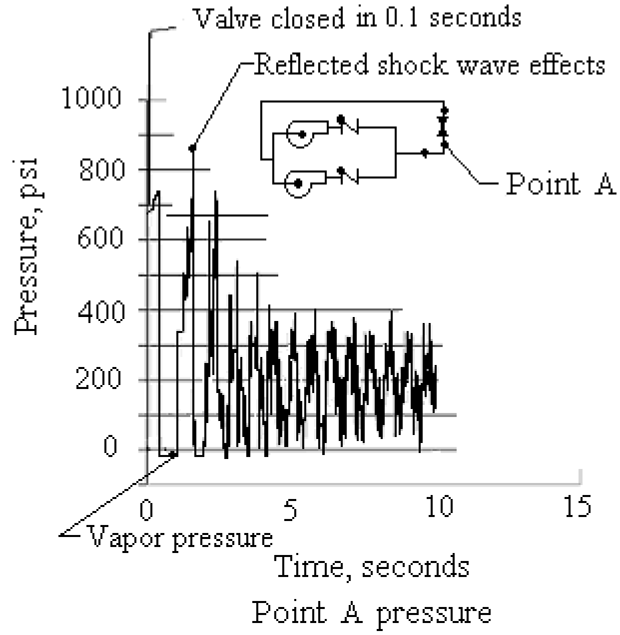

FIGURE 6: FLUID TRANSIENT DUE TO A SUDDEN VALVE CLOSURE (Leishear [1 and 2])

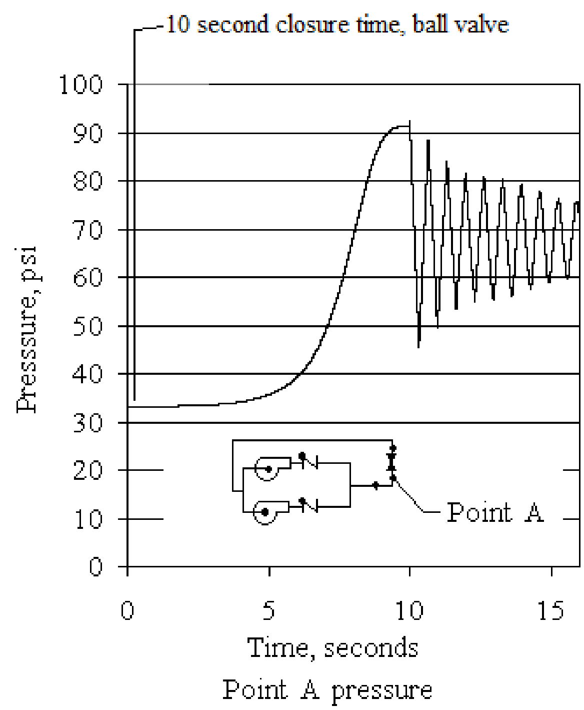

FIGURE 7: FLUID TRANSIENT DUE TO A SLOW CLOSING VALVE (Leishear [1 and 2])

Slow Valve Closures

Slow valve closure speeds result in lower pressures than those that would be caused by sudden valve closures. See Figs. 6 and 7. In fact, pressure surges may be eliminated if the closure speed is low enough, and the pressure will slowly rise to the operating pressure when a pump is started.

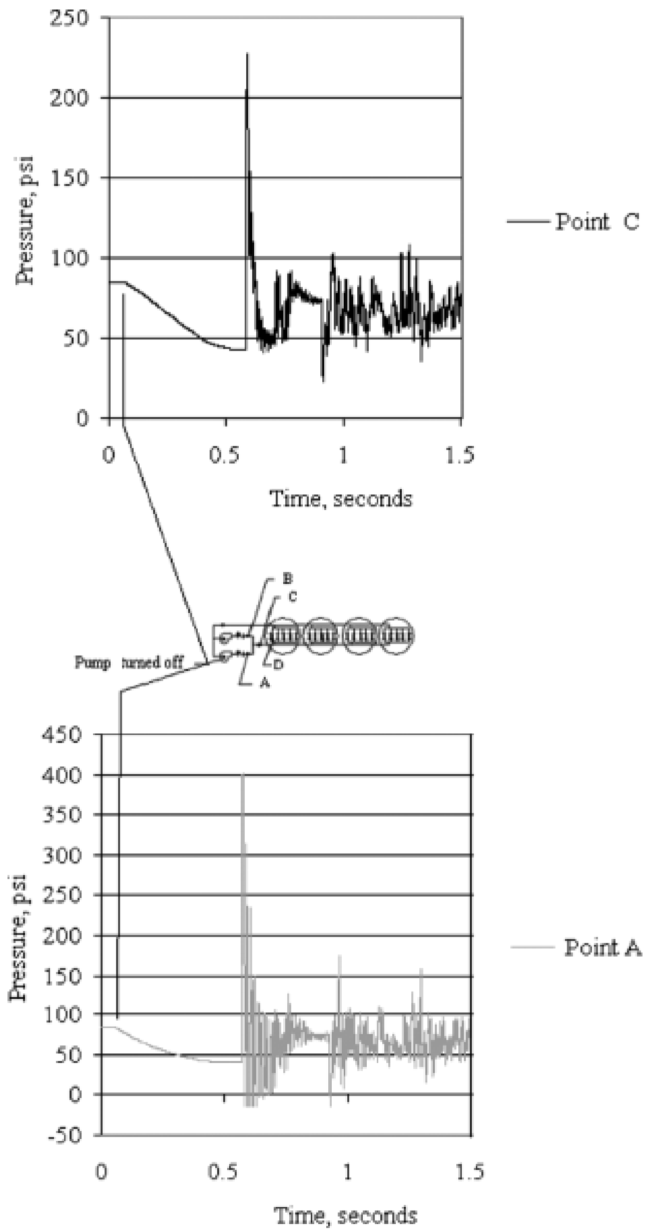

FIGURE 8: PRESSURES DUE TO STOPPING ONE OF TWO PUMPS (Leishear [1 and 2])

Pump Operations

Pump operations are complicated by their installation and pump performance. In a long pipeline, a pump can act like a suddenly opened or closed valve.

Pump Startups and Shutdowns

Fluid transients occur when pumps are started or stopped. In a closed loop system, the flow will coast to a stop, and reverse flow through a pump will not occur.

Reverse Flow Through A Closing Check Valve During Single Pump Operation

In a system pumping uphill using a single pump, a check valve is required to prevent reverse flow through a pump. Pressure surges will occur when the check valve closes.

If pumping downhill, there will be flow separation in the piping. The piping will be completely drained if the lower end of the piping is open to atmosphere. Otherwise fluid transients will occur due to vapor collapse on restart of the pump or opening of the valve, when upstream liquid is present.

Reverse Flow Through A Closing Check Valve During Parallel Pump Operations

When one of two pumps shuts down, the flow from the operating pump will reverse the flow through the stopping pump and hammer the check valve to a closed position. Resulting high pressures will occur, which are affected by the closing characteristics of the check valve. See Fig. 8.

Reflected Pressure Waves

Reflected pressure waves occur at all tees, changes in pipe diameter, and changes in pipe wall material. The maximum reflected pressure waves occur at the closed ends of piping, where the magnitude of a pressure wave is doubled as the impinging wave reflects back into the piping.

Pressurizing or Depressurizing a System with Noncondensable Gas Pockets

In general, trapped gas, or air, collects at high points in piping systems. These noncondensable gas pockets act to reduce the pressures caused by fluid transients when valves are closed or systems are pressurized. See Figs. 6 and 9.
When systems are depressurized, noncondensable gasses expand, and system surge tanks or cooling tower levels open to atmosphere will suddenly increase, which may cause overflows.

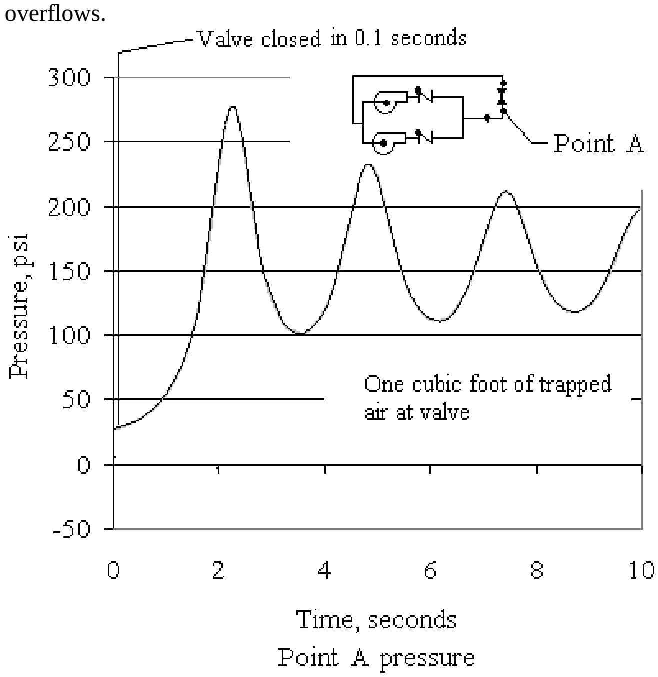

FIGURE 9: AIR POCKET EFFECTS ON A FLUID TRANSIENT (Leishear [1 and 2])

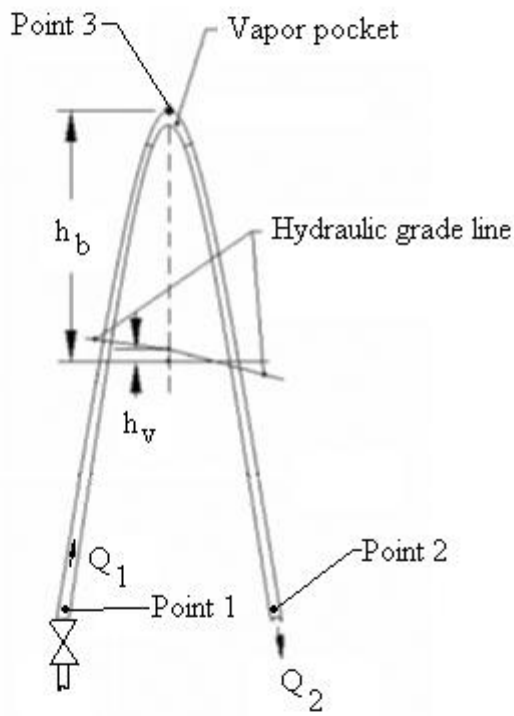

FIGURE 10: COLUMN SEPARATION, VOID FORMATION, AND VAPOR COLLAPSE (Wiley [4])

Vapor Cavity Collapse

Vapor cavity collapse and increased pressures occur in many different circumstances. Only some of those conditions are presented here. Any time that a vapor pocket is present in a system, valve or pump operations will cause pressure increases, and additionally valve and pump operations may form vapor pockets that create pressure increases.

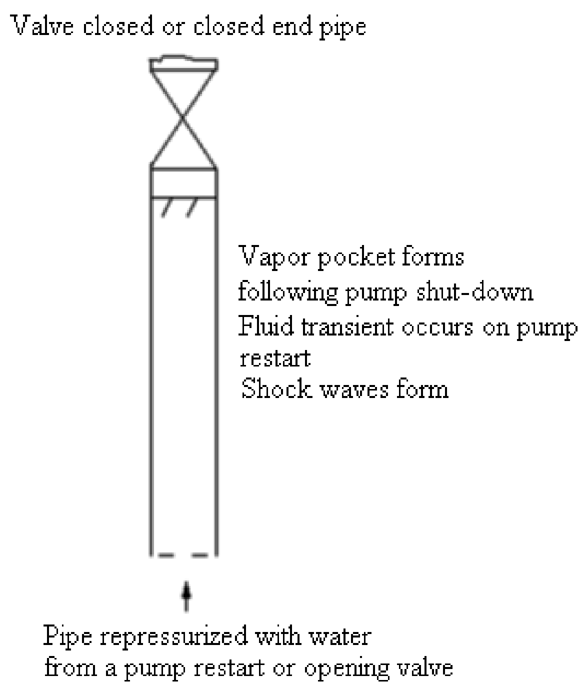

FIGURE 11: FILLING A VERTICAL PIPE CONTAINING A TRAPPED VAPOR SPACE (Merilo [5])

Vapor Collapse During Valve and Pump Operations

When valves are suddenly closed, or pumps are shut down, vapor collapse may occur at high points in the system, where this vapor collapse is dependent on the hydraulic grade line of the system. See Fig. 10. High pressures may occur near the pump or valve as well as at the high points of the piping system.

When the pressure in a pipe reduces to the vapor pressure of the liquid contained in the pipe, the liquid vaporizes. During void formation, this low pressure occurs when the liquid column in the pipe separates due to motion. When the column rejoins, the vapor collapses.
Waves are induced throughout the liquid filled system. Again, computer simulations are recommended, but when computer simulations are unavailable systems should be designed to prevent occurrence of vapor collapse.
Where elbows occur near a starting pump, high forces and stresses may occur, since the pump may be operating near runout and the resultant flow rates that induce piping stresses will cause off-normal design conditions.

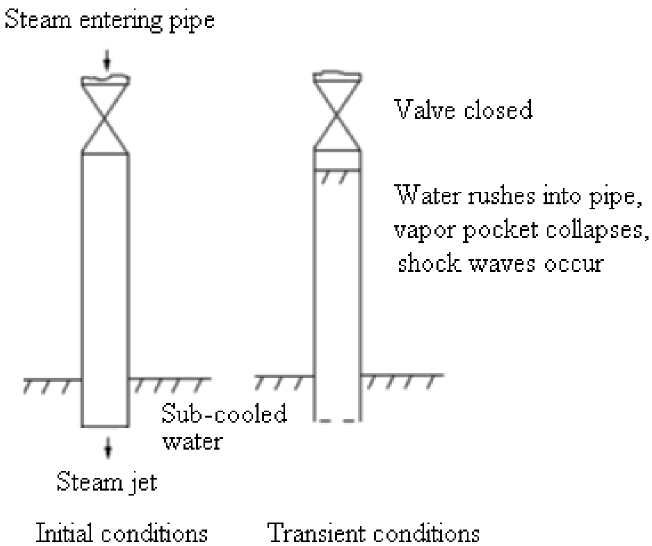

FIGURE 12: WATER CANNON, SUB-COOLED WATER WITH CONDENSING STEAM IN A VERTICAL PIPE (Merilo [5])

Vapor Collapse During Pump Start-Ups / Filling a Voided Pipeline

Vapor pockets may form at system high points when pumps are shut down or valves are closed. These pockets collapse when the system is re-pressurized by an upstream valve opening or pump restart, as shown in Figure 11. Resultant pressure surges occur.

For piping systems where vapor accumulates at high points in the system after pump shut-downs systems, Equation 20 or 21 governs the pressure surge on re-pressurization. Computer simulations are frequently inadequate for this type of transient.

Water Cannon

The water cannon mechanism is described using Fig. 12, and is described as follows.
Steam enters a cooler fluid
A valve is closed to throttle or stop the flow.
Steam condenses as water rushes into the pipe.
Vapor and liquid mix to form vapor pockets, which then collapse and cause pressure waves.

Steam / Water Counter Flow

Commonly called condensate induced water hammer, counter flow causes pressure increases in piping.

Condensate induced water hammer occurs in a horizontal pipe when liquid accumulates below a vapor flow in a pipe. The water may initially be moving or stationary in the pipe. One cause of condensate accumulation occurs during steam system shut-down, where the closed system cools to form vapor and liquid. Another source of condensate is considered in Fig. 12. Both causes result in condensate induced water hammer.

For Fig. 13:
Condensate slowly fills a pipe supplied with steam.

Unstable wave growth occurs.
Steam voids and slugs are formed.
Slugs accelerate.
Steam voids collapse due to slug impacts.
Waves and / or liquid pulses are formed.

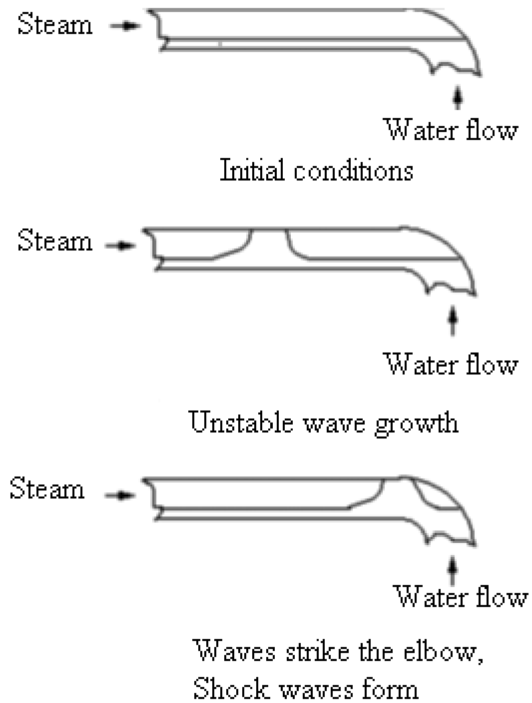

FIGURE 13: CONDENSATE INDUCED WATER HAMMER IN A HORIZONTAL PIPE (Merilo [5])

Slug Flow Transients

Slug flow transients are another example of vapor cavity collapse, and occur when a volume of water moves through a pipe due to an applied pressure. The slug will cause high pressures when it strikes elbows or the closed end of a pipe. For example, when a condensate slug moves through a system, it will impact every elbow as it travels and induce bending stresses at each of those elbows. When the slug reaches a closed end of the piping, the vapor in the piping will condense and collapse with negligible resistance to the moving slug of condensate, where increased pressures and reflected pressure waves will then occur.

Relief Valve and Safety Valve Operations

Forces on relief valve and safety valve piping depend on the designs of the connected piping (See ASME B31.1 [6]).

Valves Open to Atmosphere

When relief valves or safety valves open to atmosphere through their c piping, significant forces occur on the piping. ASME B31.1 [6] provides simplified methods to calculate these forces.

Flare Headers

Flare headers connect system piping to safety valves. Depending on design, forces and stresses can be minimized.

PART IV, PRESSURE TRANSIENTS DUE TO WATER HAMMER

Of the many types of water hammer, some types may be described by available calculation methods, which are described below. In general, computer solutions provide more accurate results.

Pressure Surges Due to Sudden Valve Openings and Closures

For a suddenly (instantly) closed valve, the maximum upstream pressure surge magnitude (ΔP), or head (Δh) with respect to the operating pressure (P_0), is described approximately by Leishear [1 and 2], such that

$$\Delta h(ft) = \pm \frac{a(ft/\sec)\cdot \Delta V(ft/\sec)}{g_c(ft/\sec^2)} \\ = \pm a \cdot \Delta V / g \tag{1}$$

$$\Delta h(m) = \pm \frac{a(m/\sec)\cdot \Delta V(m/\sec)}{g_c(m/\sec^2)} \\ = \pm a \cdot \Delta V / g \tag{2}$$

$$\Delta P(lbf / ft^2) = \pm \frac{\rho \cdot a \cdot \Delta V}{g_c} \\ = \pm \frac{\rho(lbm/ft^3)\cdot a(ft/\sec)\cdot \Delta V(ft/\sec)}{g_c(ft \cdot lbm/lbf \cdot \sec^2)} \tag{3}$$

$$\Delta P(N/m^2) = \pm \rho \cdot a \cdot \Delta V \\ = \pm \rho(kg/m^3)\cdot a(m/\sec)\cdot \Delta V(m/\sec) \\ \cdot (N \cdot \sec^2 / kg \cdot m) \tag{4}$$

where the minimum pressure is limited by the vapor pressure of the liquid in the pipe under vacuum conditions. Refer to Figure 4.1A (Valve 1). The surge upstream of the closed valve travels at a sub-sonic wave velocity, *a*. Pressure surges on the downstream side of the closing valve may also be affected by flow separation and vapor collapse.

The pressures, or heads, in a pipe following a valve closure are more complicated than indicated by Equations 1 - 4, as shown in the simplified system of Figure 14. Waves travel back and forth in the pipe at a velocity, *a*, while the pressure surge behind the wave changes from positive to negative as the flow rate in the pipe stops and changes direction. The cycle repeats itself until system resistance stops fluid motion in the pipe (Fig. 15).

Using Equations 3 or 4, the maximum pressure surge for Schedule 40 pipe is described by Fig. 16. It is emphasized that these pressures assume an instantaneous flow stoppage, which may be conservative but provide a bounding limit for pressure surge estimates.

There is also a gradual pressure increase in the wake of the pressure surge, referred to as line pack. Figure 15 depicts line pack at a point in a piping system, as well as a comparison of experimental results to a method of characteristics calculation for the fluid transient.

Pressures downstream of a closing valve in a pipeline may cause different maximum pressures than those observed upstream of a closing valve. For example, consider the system shown in Figure 4.1A, but consider only Valve 2 while assuming that Valve 1 remains open. The maximum downstream head rise due to the instantaneous closure of Valve 2 equals (Parmakian [7])

$$\Delta h = 0.22 \cdot h_0 \tag{5}$$

The maximum head drop downstream of the closing Valve 2 equals (Parmakian [7])

$$\Delta h = 0.53 \cdot h_0 \tag{6}$$

In other words, the maximum head rise and drop upstream of a closing valve is described by Equations 1 and 2, while the maximum and minimum head downstream of a closing valve may be described by Equations 3 and 4.

Higher pressure surges may also be initiated on the downstream side of a closing valve due to flow separation and vapor collapse (see para. 6.1). Downstream pressures may also be reduced by system venting or draining.

Referring to Fig. 14, the maximum head rise upstream of Valve 1 due to opening the valve equals (Parmakian [7])

$$\Delta h = 0.23 \cdot h_0 \tag{7}$$

Maximum pressure surge magnitudes initiated by valve operations in recirculating systems are also governed by Equations 3 – 7, but reflected waves throughout the system may further increase pressure surges.

Planned valve closures are to be expected in operation, but off-normal conditions should also be anticipated. For example, unplanned transients may be caused by fail-to close or fail-to-open valves on loss of electric power, or loss of air supply, depending on whether electric power or air pressure is used to actuate the valve.

The definition of an instantaneous valve closure depends on specific systems. Equations 3 and 4 provide the approximate the maximum pressure surges due to valve closures, where the pressures may be reduced by valve closing characteristics. In particular, the magnitude of the pressure surge is affected by the closure rates of the valve, the valve design, pipe diameters, pipe lengths, number and types of fittings, and system friction. Equations presented in this Appendix may be used in some case, but in other cases computer simulations may be required to evaluate decreases in the maximum pressure caused by valve operations.

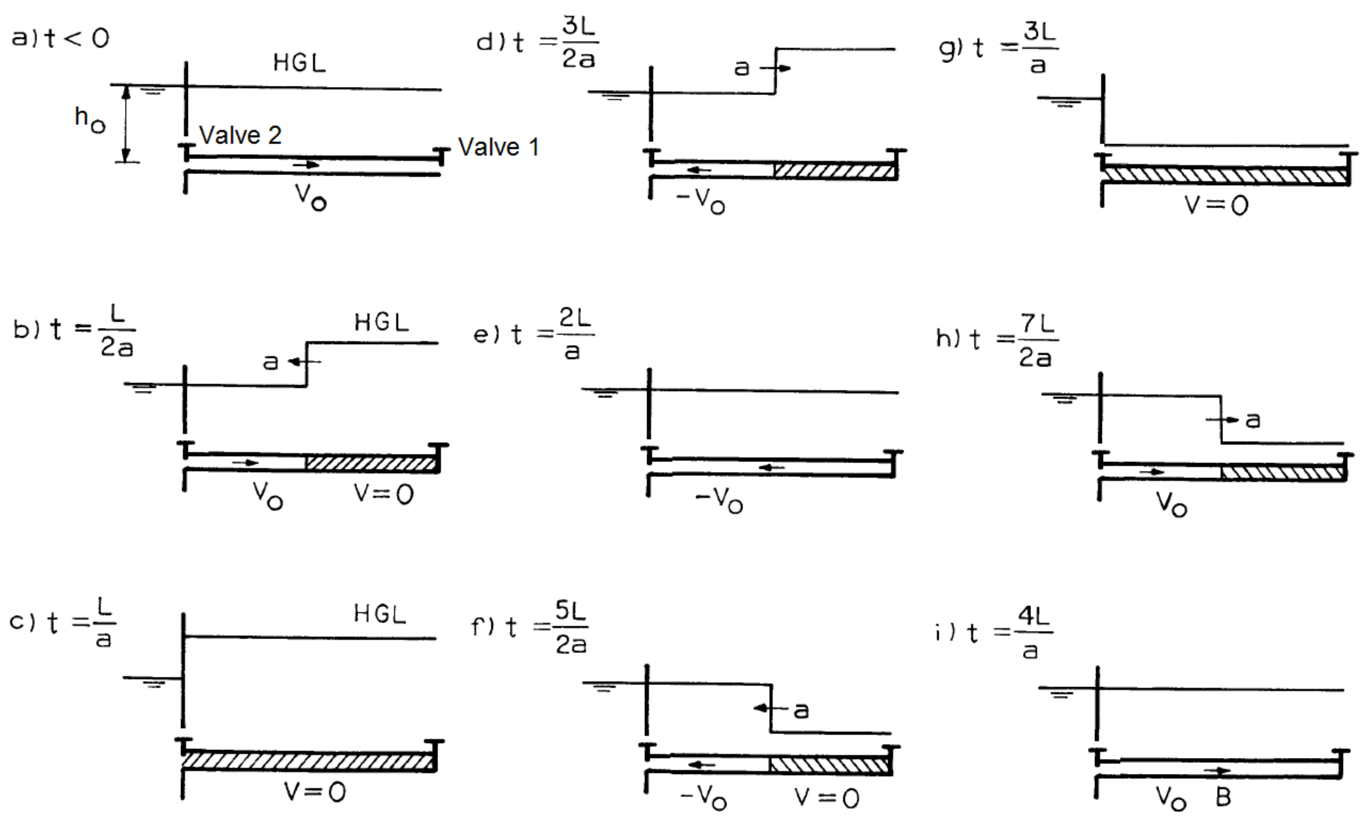

FIGURE 14: FLUID TRANSIENTS DUE TO A SUDDEN VALVE CLOSURE IN A PIPE ATTACHED TO A RESERVOIR OR TANK (PIPE LOSSES NEGLECTED) (Martin [8])

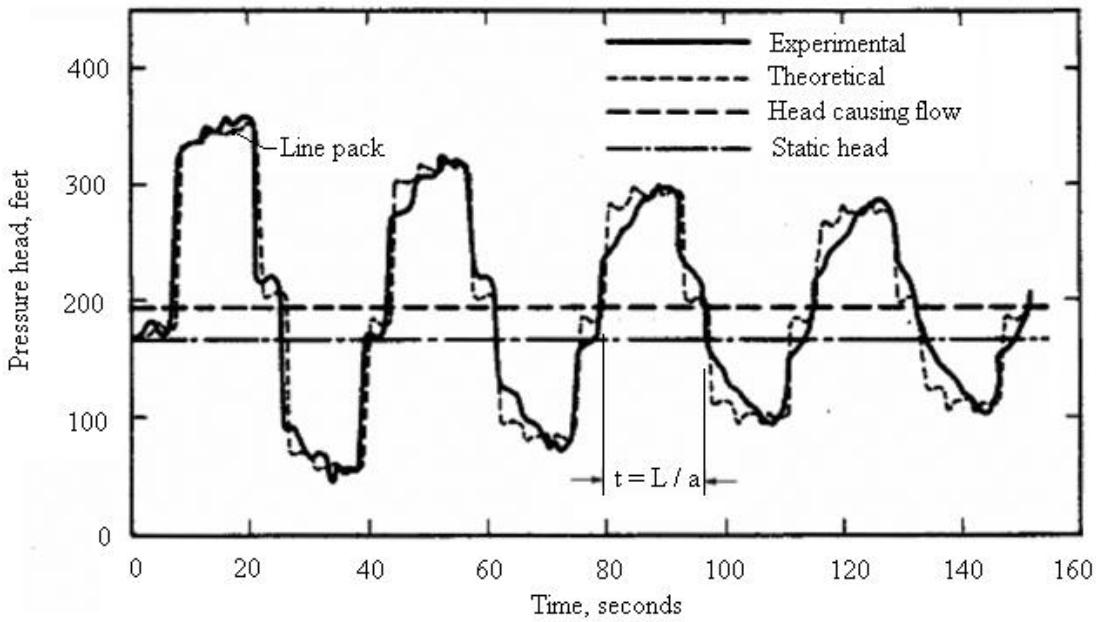

FIGURE 15: COMPARISON OF THEORY TO EXPERIMENT, PIPE LOSSES CONSIDERED (Wiley [4])

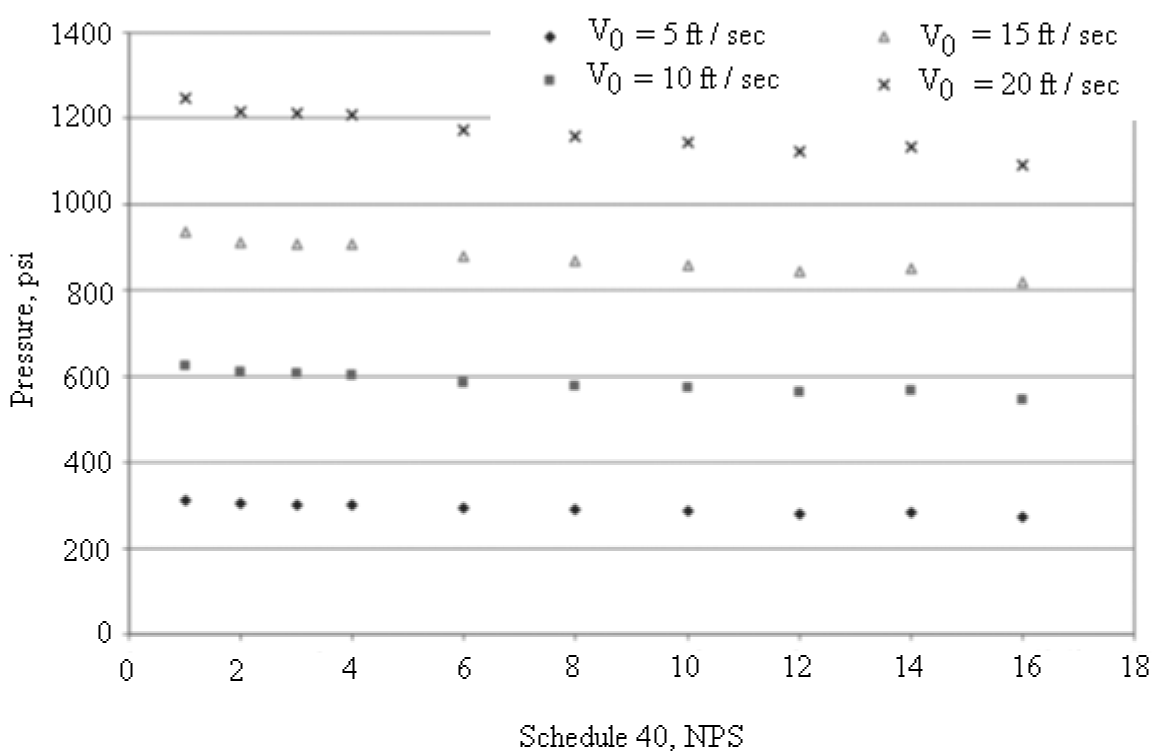

FIGURE 16: MAXIMUM PRESSURES DUE TO SUDDEN VALVE CLOSURES IN SCHEDULE 40 PIPE CONTAINING WATER, ANCHORED, THICK WALL PIPE APPROXIMATION (Leishear [1])

Pressure Wave Velocities in Liquid Filled Piping

Wave velocities, *a*, in a pipe are described by Equations 8 - 12, where the constant, *c*, describes limiting restraint conditions for piping. The actual value of the wave velocity in a pipe occurs between these two limiting values, and other values of *c* for different restraint conditions are available in the literature (Leishear [1 and 2]).

$$a = \sqrt{\frac{\dfrac{k\left(lbf\ /\ ft^2\right)\cdot g\left(ft/\sec^2\right)}{\gamma\left(lbf\ /\ ft^3\right)}}{1+\left(\dfrac{k\left(lbf\ /\ ft^2\right)}{E\left(lbf\ /\ ft^2\right)}\right)\cdot\left(\dfrac{ID}{\bar{T}}\right)\cdot c}} \tag{8}$$

$$a = \sqrt{\frac{\dfrac{k\left(N/m^2\right)}{\rho\left(kg/m^3\right)}}{1+\left(\dfrac{k\left(N/m^2\right)}{E\left(N/m^2\right)}\right)\cdot\left(\dfrac{ID}{\bar{T}}\right)\cdot c}} \tag{9}$$

Note that this velocity, *a*, is less than the speed of sound, or acoustic velocity, of the enclosed fluid, and that the acoustic velocity is affected by materials, pressures and temperatures, as shown in Figs. 17 through 19. Also, entrained air in the system can significantly change the wave speed.

Figure 18depicts wave speeds for thin wall pipes. For thin wall pipe anchored along the pipe length,

$$c = 1-\nu^2 \tag{10}$$

The effects of piping material on wave speeds are shown in Fig. 19.

Thick wall approximations provide better accuracy for piping calculations. Simplifying assumptions are required to approximate wave speeds. For a thick wall pipe, assuming that one end is fixed and the other is free to move at the downstream end of the piping,

$$c = \frac{2\cdot\bar{T}}{ID}\cdot(1+\nu)+\frac{ID}{ID+\bar{T}}\cdot\left(1-\frac{\nu}{2}\right) \tag{11}$$

For a thick wall pipe, and assuming that it is restrained along its entire length (anchored),

$$c = \frac{2\cdot\bar{T}}{ID}\cdot(1+\nu)+\frac{ID}{ID+\bar{T}}\cdot\left(1-\nu^2\right) \tag{12}$$

Wave velocities in liquids are reduced when vapors or gasses are entrained in solution, due to the increase in compressibility, and are increased when solids are entrained in solution. In most cases, the entrapped air in solution has little effect, but the effects on wave speed where air is forced into the system should be evaluated.

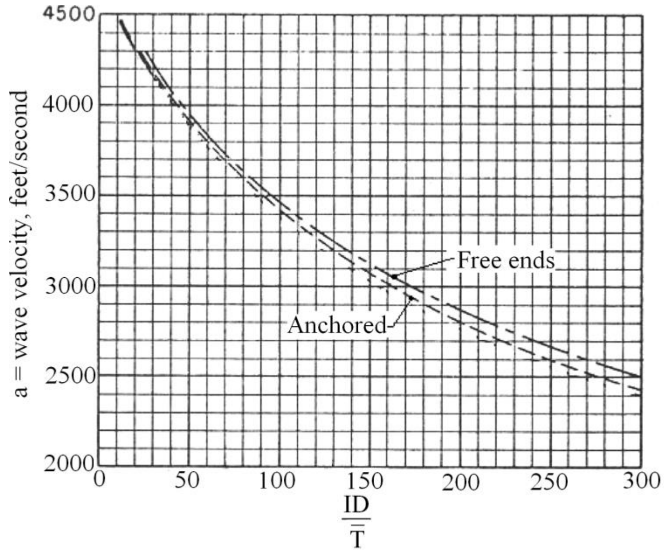

FIGURE 17: WAVE VELOCITIES FOR WATER FILLED, THIN WALL PIPING AT STANDARD CONDITIONS (Parmakian [7])

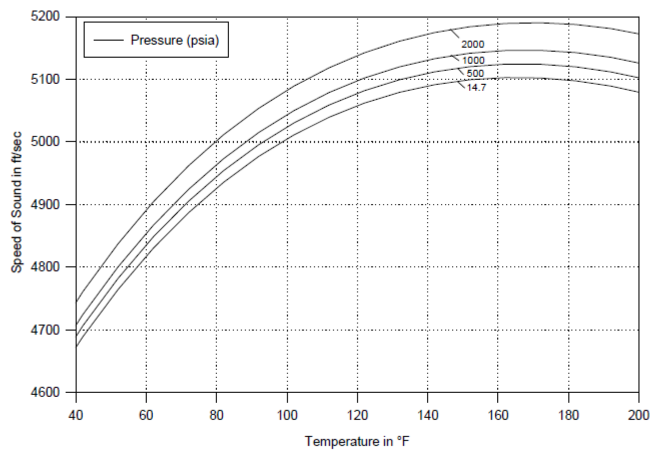

FIGURE 18: SPEED OF SOUND FOR WATER AT DIFFERENT PRESSURES AND TEMPERATURES IN THIN WALL PIPING (Martin [8])

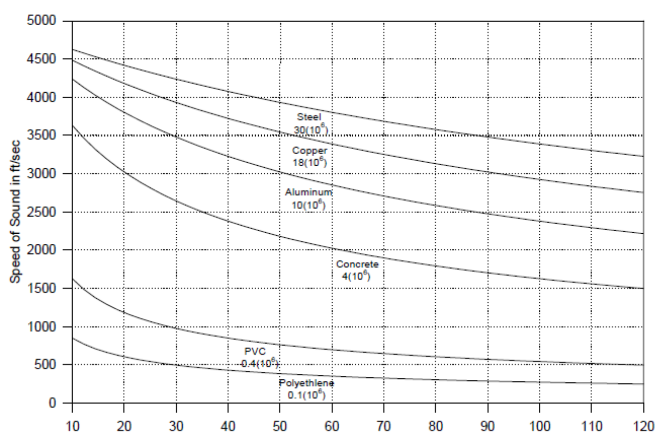

FIGURE 19: WAVE VELOCITIES FOR DIFFERENT MATERIALS IN ATHIN WALL ANCHORED PIPELINE (Martin [8])

Reflected and Transmitted Waves in Liquid Filled Piping

Waves initiated in piping systems are further complicated by reflected and transmitted waves, which occur at all transitions in diameter (reducers) or transitions in pipe material where densities or elastic moduli vary significantly at the transition (Figs. 19 – 21) Additionally, fully reflected waves occur at closed ends in pipelines. Reflected waves may increase the pressure in the pipeline, and failures caused by reflected waves may occur close to the end of a pipe or near a transition in material or pipe diameter. The equations presented here provide simplified results, which do not include pressure reductions due to friction or energy losses due to pipe deformations, where pressures will be reduced due to friction.

A reflected wave at a closed end doubles the incident pressure wave magnitude at the closed end of a pipe. The magnitude of a reflected wave may be reduced if trapped air is present at the end of the pipe. Neglecting pipe losses, the maximum total pressure, P_{total}, near the end of pipe caused by an instantaneous valve closure in the system equals the sum of two pressure wave magnitudes ($2 \cdot \Delta P$) added to the initial pressure (P_1) in the piping, such that (Leishear [1 and 2])

$$\begin{aligned} P_{total} &= P_1 + 2 \cdot \Delta P \\ &= P_1\left(lbf / ft^2\right) + 2 \cdot \frac{\rho\left(lbm / ft^3\right) \cdot a\left(ft / \sec\right) \cdot \Delta V\left(ft / \sec\right)}{g_c\left(ft \cdot lbm / lbf \cdot \sec^2\right)} \end{aligned} \tag{13}$$

$$\begin{aligned} P_{total} &= P_1 + 2 \cdot \Delta P \\ &= P_1\left(N / m^2\right) + 2 \cdot \rho\left(kg / m^3\right) \cdot a\left(m / \sec\right) \cdot \Delta V\left(m / \sec\right) \end{aligned} \tag{14}$$

When a valve is opened into a closed end pipe filled with liquid, the change in pressure due to the valve's sudden opening is doubled in the pipe due to wave reflections at the pipe end, where the incident wave of pressure magnitude, ΔP, reflects to yield a pressure of $2 \cdot \Delta P$, which is added to the initial pressure in the piping (P_1), such that.

$$P_{total} = P_1 + 2 \cdot \Delta P \tag{15}$$

The total pressure in the piping downstream of the opening valve then equals the initial pressure in the downstream piping plus $2 \cdot \Delta P$.

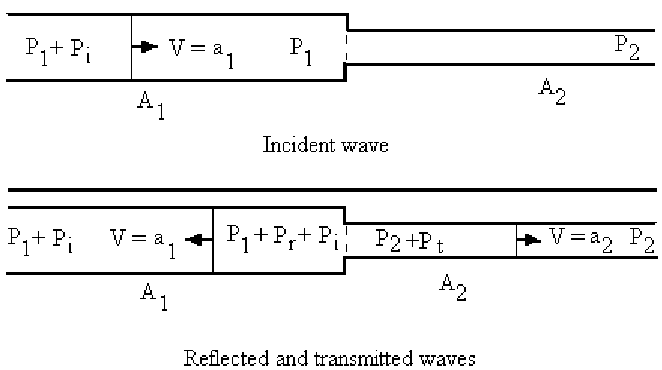

FIGURE 19: PRESSURE MAGNITUDES DUE TO REFLECTIONS AT REDUCERS AND CHANGES IN PIPING MATERIALS

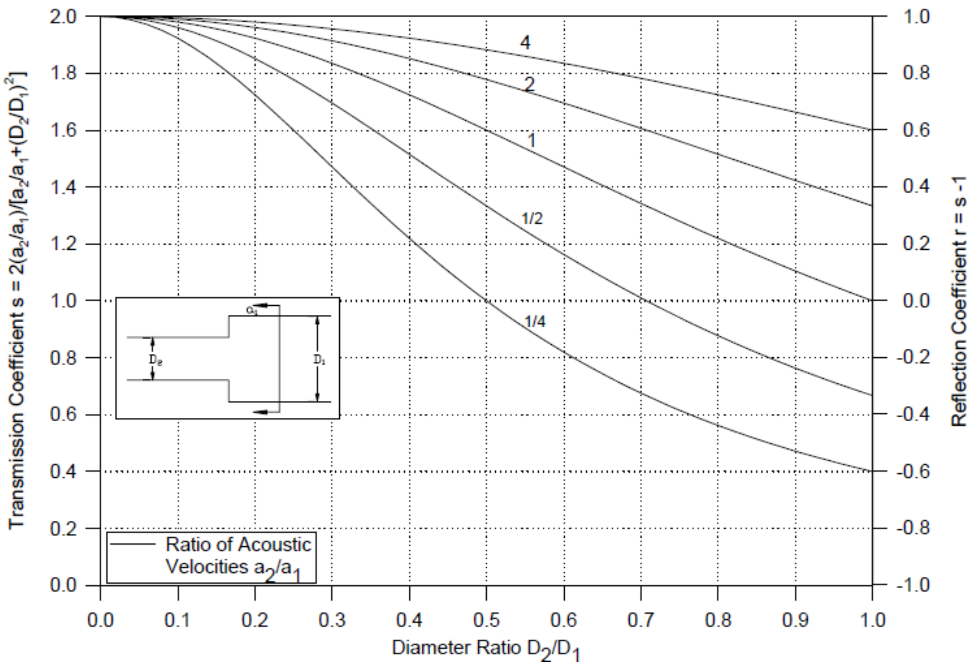

FIGURE 20: WAVE REFLECTIONS AT A REDUCER WHEN THE DIAMETER INCREASES (Martin [8])

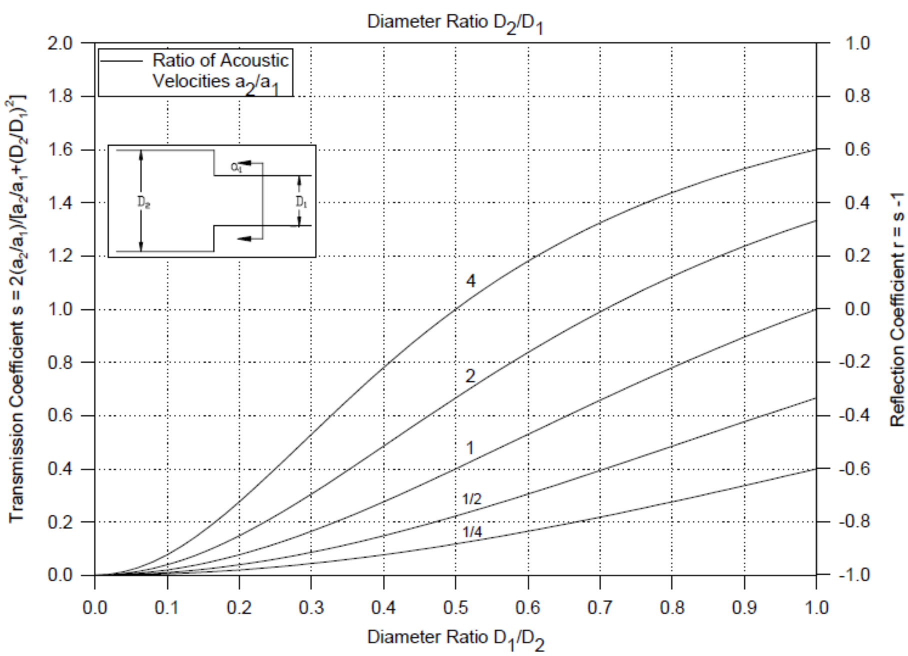

FIGURE 21: WAVE REFLECTIONS AT A REDUCER WHEN THE DIAMETER DECREASES (Martin [8])

At a pipe transition in diameter or material, the relationship between the pressure magnitudes of the incident (P_i), transmitted (P_t), and reflected pressure waves (P_r) may be expressed as

$$P_t = P_i \cdot \frac{\dfrac{2 \cdot A_1}{a_1}}{\dfrac{A_1}{a_1} + \dfrac{A_2}{a_2}}$$

(16)

$$P_r = P_i \cdot \frac{\dfrac{A_1}{a_1} - \dfrac{A_2}{a_2}}{\dfrac{A_1}{a_1} + \dfrac{A_2}{a_2}}$$

(17)

where A_1 and A_2 are the cross-sectional areas, and a_1 and a_2 are the wave speeds.

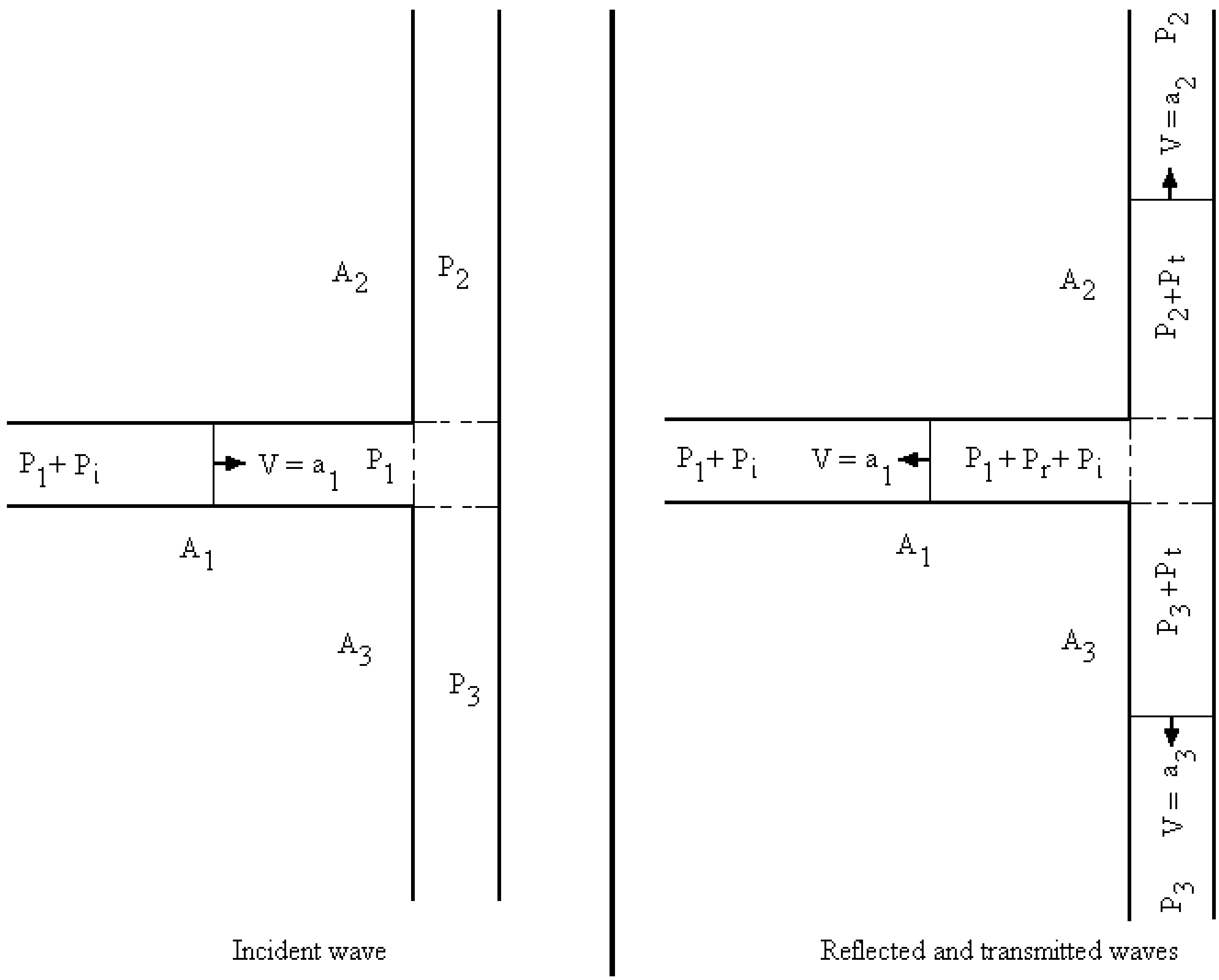

FIGURE 22: PRESSURE MAGNITUDES DUE TO REFLECTIONS AT PIPE INTERSECTIONS

The pressure magnitudes of the incident, transmitted, and reflected waves at a tee, or branch intersection, may be expressed as

$$P_t = P_i \cdot \frac{\dfrac{2 \cdot A_1}{a_1}}{\dfrac{A_1}{a_1} + \dfrac{A_2}{a_2} + \dfrac{A_3}{a_3}} \tag{18}$$

$$P_r = P_i \cdot \frac{\dfrac{A_1}{a_1} - \dfrac{A_2}{a_2} - \dfrac{A_3}{a_3}}{\dfrac{A_1}{a_1} + \dfrac{A_2}{a_2} + \dfrac{A_3}{a_3}} \tag{19}$$

where a_3 and A_3 are wave speeds and areas.

Reflected and transmitted waves occur at all transitions and tees in a system (Fig. 22). Both positive and negative pressures may be reflected or transmitted at piping and tank intersections with the piping. Computer simulations are frequently required to fully understand complicated system performance, where the method of characteristics is an accepted simulation technique. Even so, Equations 18 and 19 provide simplified calculations for many designs.

PART V, DYNAMIC PIPING STRESSES DUE TO WATER HAMMER

Dynamic load factors are presented here to describe the maximum stresses in piping due to suddenly applied fluid transients, which can be described by step pressure increases. Again, some cases are adequately represented by these simplifying assumptions, while many are not. The effects of damping and rates of loading are not considered in detail here, but these effects will reduce dynamic load factors. Several examples for the use of DLF's follow.

Pressure Loads on Safety Valve Installations

Pressure loads acting on safety valve installations are important from two main considerations. The first consideration is that safety valve installation causes membrane stresses in the pipe wall. The second consideration is that the pressure effects associated with discharge can cause loads acting on the system which create bending moments throughout the piping system. All parts of the safety valve installation must be designed to withstand the design pressures without exceeding the Code allowable stresses, and dynamic effects are considered during design. Only fluid transient loads are considered in this Appendix, and other loads are considered per the applicable standard.

Reaction Forces on Open Discharge Pipes - Forces on Discharge Elbows

The reaction force, F_1, due to steady state flow following the opening of a safety valve includes both momentum and pressure effects. The reaction force applied is shown in Figure 23, and may be computed by the following equation:

$$F_1(lbf) = \frac{W(lbm/\sec) \cdot V_1(ft/\sec)}{g_c\left(ft \cdot lbm/lbf \cdot \sec^2\right)} + \left(P_1\left(lbf\ /\ ft^2\right) - P_a\left(lbf\ /\ ft^2\right)\right) \cdot A_1\left(ft^2\right) \tag{20}$$

$$F_1(N) = W\left(kg/\sec^2\right) \cdot V_1(m/\sec) \cdot \left(N \cdot \sec^2/kg \cdot m\right) + \left(P_1\left(N/m^2\right) - P_a\left(N/m^2\right)\right) \cdot A_1\left(m^2\right) \tag{21}$$

To ensure consideration of the effects of the suddenly applied load F_1, a dynamic load factor, DLF, should be applied to the piping and components.

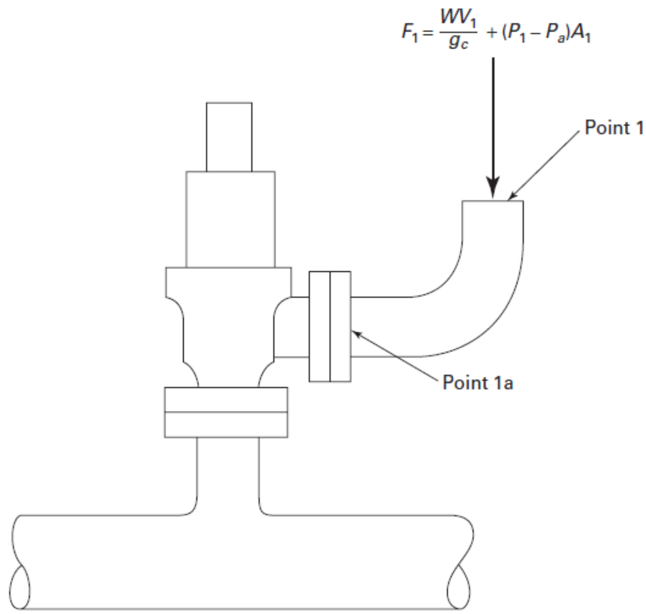

FIGURE 23: DISCHARGE ELBOW, OPEN DISCHARGE INSTALLATION (ASME [6])

Reaction Forces on Open Discharge Pipes - Forces on Vent Pipes

Figure 24 shows the external forces resulting from a safety valve discharge, which act on the vent pipe. The vent pipe anchor and restraint system must be capable of taking the moments caused by these two forces, and also be capable of sustaining the unbalanced forces in the vertical and horizontal directions.

A bevel of the vent pipe will result in a flow that is not vertical. The equations shown are based on vertical flow. To take account for the effect of a bevel at the exit, the exit force will act at an angle, φ, with the axis of the vent pipe discharge which is a function of the bevel angle, θ. The beveled top of the vent deflects the jet approximately 30 degrees off the vertical for a 60 degree bevel, and this will introduce a horizontal component force on the vent pipe system.

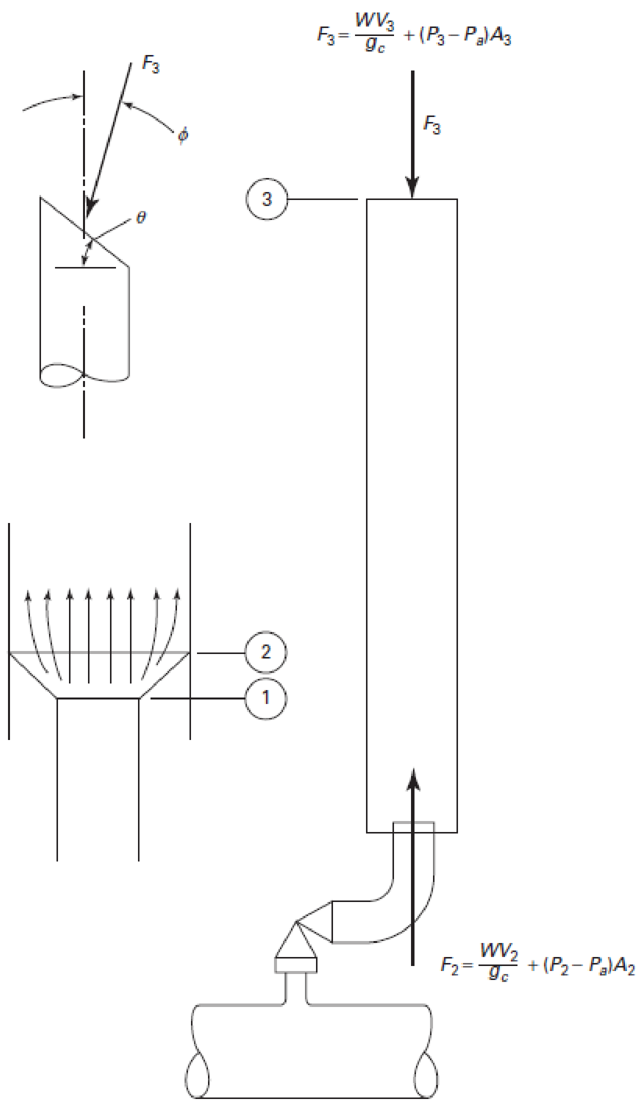

FIGURE 24: VENT PIPE, OPEN DISCHARGE INSTALLATION (ASME [6])

Piping System Loads

Dynamic loads on systems are complicated by multiple valve openings and reflected waves. The time at which these transients occur affects the system response. As different pressure waves create hoop stresses along a piping run and create bending stresses on impact with elbows and other discontinuities, stresses develop that are each damped with time. Neglecting damping, maximum stresses due to dynamic loading are described herein. The use of damping to assess system response is normally considered when performing computer analysis, and is outside the scope of this Appendix.

DLF's and Dynamic Pipe Stresses

In a piping system acted upon by time varying loads, the internal forces and moments are generally greater than those produced under static application of the load. This amplification is often expressed as the dynamic load factor, *DLF*, and is defined as the maximum ratio of the dynamic stress or reaction force at any time to the stress or reaction force which would have resulted from the static application of the load.

For simple cases which are described by one degree of freedom models, the maximum dynamic stress (S_D) is related to the static stress (S) by the equation

$$S_D = DLF \cdot S \tag{22}$$

The primary dynamic pipe stresses can be separated into hoop membrane stresses and longitudinal bending stresses, since their maximum values typically occur at much different times. Longitudinal membrane stresses should also be considered by the applicable Code.

For static loading of a piping system, the DLF = 1.

A simplified example is considered here for a 6 inch, Schedule 80, pipe and elbow. For this example, pipe supports at one end are neglected and the other end is fixed, as shown in Fig. 25. The resultant stresses due to the applied pressure, ΔP are shown in Figure 8.2B. Note that the maximum, undamped elastic stress in Fig. 26 is twice the static bending stress at equilibrium (DLF < 2), if one assumes the loading is applied as a step change with a duration much longer than the period of the piping system. Also note that the maximum, limiting or bounding, undamped hoop stress for an elastic material is less than four times the static stress (DLF < 4) (Ref. 3), and that the hoop stresses decrease to nearly the static hoop stress at the time the bending stress reaches its maximum value. A DLF 2 < 4 is only used for fatigue and linear elastic fracture mechanics analyses.

For plastic deformation during dynamic hoop stresses: 1 < DLF < 2, since damping during plastic deformation is significant and plastic precursor waves do not occur.

The unbalanced forces in the pipe system need to be addressed. Axial loads, and resultant axial stresses, in piping and equipment may be significant and should be considered in the applicable pipe code equations.

Experimental data may be used by the designer to establish lower DLF's.

In cases where applied loads are repeated at a frequency close to a system mechanical natural frequency in such a way that an associated mode is excited, dynamic load factors on the order of 10 to 15 may be observed. This type of applied load may result from the use of some types of positive displacement pumps, which provide pulsating flows. The designer should address this design concern when fluid pulsations are induced in a piping system

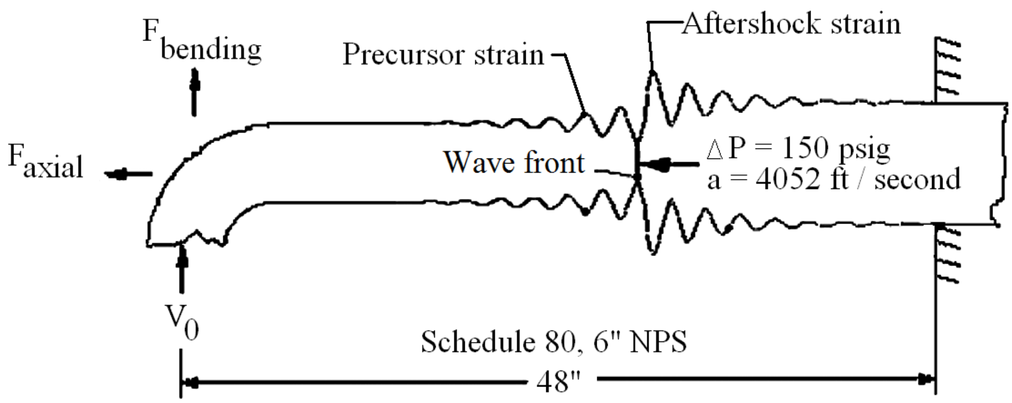

FIGURE 25: PRESSURE WAVE IMPINGEMENT ON AN ELBOW (Leishear [1])

Dynamic Hoop Stresses

Dynamic elastic hoop stresses are further complicated by the presence of precursor and aftershock vibrations as a wave travels the bore of a pipe at a nearly sonic velocity. Figure 25 depicts a typical hoop strain response. The maximum dynamic stress is no more than four times the static stress that would be created by a static pressure of equal magnitude to the pressure following a pressure wave.

DLF's for hoop stresses vary, depending on the damage mechanism of interest. During plastic deformation, the maximum DLF varies between 1 < DLF < 2. During fatigue, elastic deformation is prevalent, and the maximum DLF varies between 2 < DLF < 4.

DLF's for Hoop Stresses

For fatigue considerations, the maximum hoop stress can be expressed using a DLF < 4, where the static stress (S_{hoop}) and maximum, undamped, dynamic stress ($S_{D(hoop)}$) due to a sudden pressure increase on the inner pipe wall equal

$$S_{hoop} = \frac{\Delta P \cdot \left(OD^2 + ID^2\right)}{\left(OD^2 - ID^2\right)}$$

(23)

$$S_{D(hoop)} = DLF \cdot S < 4 \cdot \frac{\Delta P \cdot \left(OD^2 + ID^2\right)}{\left(OD^2 - ID^2\right)}$$

(24)

For the example shown in Figs. 25 and 26, the undamped dynamic stress equals four times the static stress.

For fatigue considerations, the total stress (S_{total} due to a sudden pressure increase, or pressure wave, of magnitude ΔP then equals the stress due to the initial operating stress plus the dynamic effects of the pressure wave, such that

$$S_{total} < \frac{\left(P_0 + 4 \cdot \Delta P\right) \cdot \left(OD^2 + ID^2\right)}{\left(OD^2 - ID^2\right)}$$

(25)

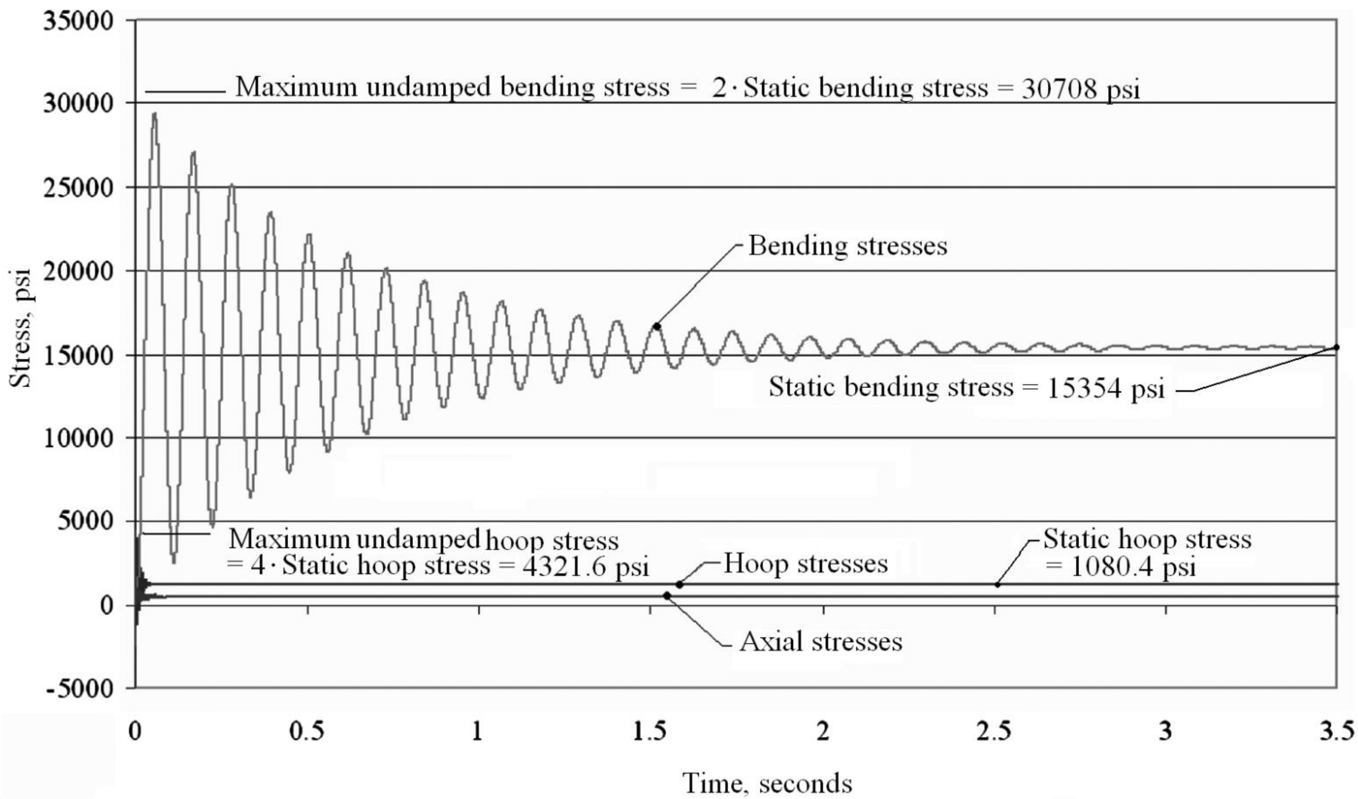

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FIGURE 26: STRESSES AT AN ELBOW SUBJECT TO A STEP PRESSURE INCREASE (Leishear [1])

Reflected Pressure Waves and Hoop Stresses

Hoop stresses near the end of a closed end pipe are even higher as reflected waves double the pressure near the pipe end. The stresses due to the incident and reflected waves add together at points along the pipe. Consequently, the highest stresses due to wave reflections occur near the closed end of a pipe, near a reducer, or near a change in pipe wall material. As a reflected pressure wave returns into the piping, the dynamic stresses due to the incident wave are also reduced due to damping.

For plastic stresses due to wave reflections near the end of a closed end pipe, the maximum DLF ≈ 1.

Dynamic Bending Forces and Reactions

Permitting simplified calculations, dynamic bending stresses and reactions due to fluid transients (such as slug flow) are caused by forces equal to

$$F(lbf) = \frac{W(lbm/s) \cdot \Delta V(ft/\sec)}{g_c\left(ft \cdot lbm/lbf \cdot \sec^2\right)} + \left(\Delta P\left(lbf\ /\ ft^2\right)\right) \cdot A\left(ft^2\right) \tag{26}$$

$$F(N) = W\left(kg/s^2\right) \cdot \Delta V(m/\sec) \cdot \left(N \cdot \sec^2/kg \cdot m\right) + \left(\Delta P\left(N/m^2\right)\right) \cdot A\left(m^2\right) \tag{27}$$

where W equals the mass flow rate.

In liquid filled systems where a valve is suddenly closed, changes in momentum are not a significant contributor, and the forces at elbows are due primarily to pressure waves, where the resultant forces are described by

$$F(lbf) = \Delta P\left(lbf\ /\ ft^2\right) \cdot A\left(ft^2\right) = \frac{\rho\left(lbm/\ ft^3\right) \cdot a(ft/\sec) \cdot \Delta V(ft/\sec) \cdot A\left(ft^2\right)}{g_c\left(ft \cdot lbm/lbf \cdot \sec^2\right)} \tag{28}$$

$$F(N) = \Delta P\left(N/m^2\right) \cdot A\left(m^2\right) = \rho\left(kg/m^3\right) \cdot \left(N \cdot \sec^2/kg \cdot m\right) \cdot a(m/\sec) \cdot \Delta V(m/\sec) \cdot A\left(m^2\right) \tag{29}$$

The effects of dynamic forces at elbows may be significantly affected due to the effects of similar forces at opposing elbows, i.e, U-bends or Z-bends. The distance between bends will affect the forces and resultant piping reactions due to the time of impact at each bend and the time dependent mismatch of opposing forces at the elbows.

As elbows are placed closer together at U-bends, forces will cancel perpendicular to the direction of the pipe runs, since the forces at the two elbows will be opposite in direction, but two distinct additive forces will occur in the direction of the pipe runs at the two elbows.

As elbows are placed closer together at Z-bends, forces will cancel perpendicular to the direction of the pipe runs, since the forces at the two elbows be opposite in direction, but moments will occur due to opposing forces at the two elbows in the direction of the pipe runs.

Bending Forces and DLF's

Bending stresses may also be affected by the duration of the pressure surge or the type of loading, where elastic stresses have a DLF< 2 and plastic bending stresses DLF ≈ 1, due to a sudden pressure increase caused by a steep fronted pressure wave water hammer.

For example, the load may be applied as a step pressure increase, a linear ramp increase, or some other type of loading. An approximation of ramp response loading is shown in Fig. 27, where τ equals the period of the piping vibration, and t_1 equals the rise time. Note that the step pressure increase equals the ramp response with a zero rise time. The effects of the load duration are shown in Fig. 27, where t_0 equals the duration of the load, or in this case, the duration of the pressure surge.

Piping configuration also significantly affects pipe stresses. If U-bends are installed, the opposing forces at the two elbows are in phase and will cancel the bending stresses for short distances between elbows. Note however that axial stresses will be doubled. As the distance between the elbows increases, the forces are increasingly unbalanced as they become out of phase with each other. However, pipe supports between elbows can reduce or even eliminate the counterbalancing forces at elbows.

For Z-bends, the forces are similarly related, but a moment also occurs between the two elbows that increases as the distance between the elbows increases.

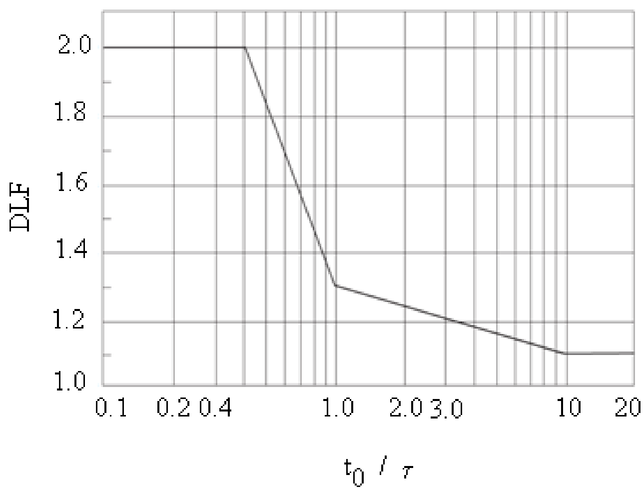

FIGURE 27: EFFECTS OF SHORT DURATION PRESSURE SURGES ON SYSTEM RESPONSE, STRESSES OR REACTIONS (ASME [6])

Reactions

Reaction forces due to fluid transients are dynamic in nature. A time-history dynamic solution, incorporating a multi-degree of freedom lumped mass model solved for the transient hydraulic forces is considered to be more accurate than the simplified form of analysis presented in this Appendix.
If the piping support is located at the elbow where the pressure wave impacts, then Fig. 28 can be used to approximate the maximum value of the reaction by neglecting the reduction in reaction force due to the attached piping. Otherwise, reaction forces may be significantly reduced due to the interrelationship between the pipe vibrations and the pipe support vibrations.

The effects of frequency on piping / pipe support interaction may be considered. Computer based analyses may consider the variation in vibration frequencies that accompany assumptions for pipe supports. For example, the frequency for a pipe with fixed ends varies by a factor of more than 2 when compared to that same pipe supported with simply supported ends, which have lower stiffness and yield a lower frequency.

Loads on Other Piping Components

In general, the DLF to be applied to valves is DLF < 2, since precursor strains are not expected to form in the valve body. However, in large valves there may be zones within the valves where wave reflections occur and the DLF < 4 at those locations. Fluid flow simulations may be warranted.
For flange bolts, the DLF of the load applied to flange surfaces and exerted on the flange bolts is expressed as DLF < 2. However, if dynamic loads cause the bolts to stretch sufficiently to permit fluid between the gasket and the flange, additional possibly catastrophic loads may be exerted on the flange bolts.

DLF's for other equipment, such as instrumentation and pressure gauges, will be similar to valve DLF's. Note also, that pressure gauges and other instrumentation frequently have inadequate response times to measure transients accurately. In other words, measured flow rates and pressures may be orders of magnitude higher than indicated if the transients are of short duration.

DAMAGE ASSESSMENT AND CORRECTIVE ACTIONS

Damage mechanisms may occur in piping systems following valve and pump operations or condensate accumulation in steam systems. Observations and corrective actions with respect to water hammer follow.

Observations include:
- Ductile pipe failures due to valve openings and closures in liquid filled systems typically include (but are not limited to) fatigue cracks causing pipe shear and plastic deformation.
- Valve leaks frequently precede fatigue damages during operations.

- Fracture of brittle components.
- Piping support damages.

Corrective actions for valve operations in liquid filled systems include:

- Automatic slow closing of valves. The definition of slow closure varies, since the closure time depends on the system resistance, where a recommended valve closure time may vary from a few seconds to a few minutes depending on pipe length. A valve closure time of $2 \cdot L / a$ does not guarantee that pressure surge magnitudes are below the maximum permitted pressures. A valve closure time of $20 \cdot L / a$ is recommended here, but faster closing times may be acceptable as determined by design.
- Manual slow closing of valves.
- Two speed valve closures (valve stroking) may also be used.
- Water hammer arrestors are a smaller version of accumulator tanks, and are used to reduce wave reflection effects near the end of piping. They contain air pockets to reduce transient pressures. Arrestors are frequently used in building piping systems, when transients are created elsewhere in the supply system.
- Note that relief valves and safety valves offer minimal control to fluid transients, since pressure waves travel past the valve into the pipe system before the valve has time to open. However, relief valves do limit the effects of reflected waves in the piping after the first wave passes the valve.
- Stronger piping and / or additional piping restraints. The preferred alternative is to design the system operation to minimize the fluid transient.

Corrective actions for transients caused by centrifugal pump operations or liquid filling of air filled pipes include:

- Variable speed, drive control for pump start-up and shut-down.
- Automatic or manual closing of pump discharge valves before starting or stopping pumps. Pump overheating and vibration should be evaluated before controlling flow using this method.
- Combined automatic valve and pump controls for start-up and shut-down of parallel pumps.
- Evaluate the effects of power losses to pump control equipment, since transients may occur due to inadvertent shut-down on loss of power to pumps.
- Spring operated check valves, weight operated swing check valves, or air cushioned swing check valves installed at the pump discharge.

Corrective actions for vapor collapse in liquid filled systems include:

- Surge tanks are typically open to atmosphere and introduce liquid into the vapor space to mitigate vapor collapse.
- Accumulators are closed, pressurized tanks that may contain a bladder between the liquid in the pipe and the pressurizing gas in the tank. Similar to surge tank operation, accumulators also force liquid into the vapor space to mitigate vapor collapse.
- Vents introduce air into a vapor space to minimize the effects of vapor collapse. Air acts a cushion to the returning water column in the pipe, while vapor does not. Air effects on system performance should be considered.

Common corrective actions for condensate induced water hammer include:

- Condensate should not be added to, or allowed to accumulate in, horizontal piping or low pipe sections of steam systems.
- Shut down the steam system immediately if a condensate induced water hammer accident occurs. Operate valves slowly when responding to an accident to mitigate an additional transient.
- During restart, steam should be slowly admitted into the piping system to mitigate excessive thermal stresses and condensate induced water hammer. A common technique is to admit steam through the regulator bypass valve, since the main valve typically cannot provide the low flow rates required to mitigate transients. Blow-down valves / drain valves are opened to drain water from the regulator at various points along the pipeline. When steam exits the blow-down valve, the valve is closed, and the next valve in succession drains until steam exits the piping. This process is performed for all blow-down valves in the system until condensate is removed. Then the main pressure regulator valve is opened, and the pressure regulator bypass valve is closed.
- Steam traps should be regularly monitored to minimize condensate accumulation.
- Safety valves and rupture discs do not open fast enough to prevent all damages from fluid transients. Typically, the first pressure wave passes the relief valve before it opens. Successive pressure waves may be reduced by these devices.
- A common method to mitigate water cannon is to introduce air into the pipe to displace the steam prior to sudden valve closure.
- Fatigue damage mechanisms should be evaluated in accordance with the applicable Code. Damping values required to evaluate fatigue shall be defined by the designer.

Fatigue cracks and fitness for service may be evaluated using API 579-1/ASME FFS-1 [9].

Plastic deformations and their disposition are the responsibility of the owner. Even so, plastic deformations and fatigue in piping are expected to have occurred in systems designed to existing pipe codes where transients occurred.

An important observation with respect to wave velocities is that fluid transients can cause damage in pipeline systems more than a mile away from the location where the transient initiated. Failures may occur near the end of a pipe, near a corroded piping section, or may occur anywhere in the system due to the wide variation in fatigue properties for a given pipe lot. Accordingly, a fluid transient may not be immediately recognized as a cause of failure following a transient.

CONCLUSIONS

An overview of fluid transients, or water hammer, was presented with respect to piping stresses, where the reader is referred to the references for more detailed information. In particular, water hammer fluid transient theory is well defined by Wiley [4]; corrective actions are described by Energy and Power Research Institute (Merilo [5]); and both experimental and theoretical discussions of dynamic stresses are available from this author (Leishear [1 and 2]).

NOMENCLATURE (U.S. Standard Units)

a, a_1, a_2, a_3	= pressure wave velocities	(L/T)
A_1, A_2, A_3	= areas	(L^2)
c	= constant	(-)
DLF	= dynamic load factor	(-)
E	= modulus of elasticity	(F/L^2)
F, F_1, F_2, F_3	= reaction force	(F)
ft	= feet	(L)
g	= local gravitational acceleration	(L/T^2)
g_c	= gravitational constant	(-)
Δh	= change in head	(-)
h_v	= vapor pressure	(F/L^2)
h_b	= barometric pressure	(F/L^2)
h_0	= initial head or static head	(-)
I	= moment of inertia	(L^4)
ID	= inside pipe diameter	(L)
k	= bulk modulus of a fluid	(F/L^2)
L	= pipe length	(L)
L_s	= slug length	(L)
L_v	= vapor space (void) length	(L)
OD	= outside pipe diameter	(L)
ΔP	= change in pressure	(FL^2)
P_a	= atmospheric pressure	(F/L^2)
P_i	= incident pressure wave magnitude	(F/L^2)
P_r	= reflected pressure wave magnitude	(F/L^2)
P_t	= transmitted pressure wave magnitude	(F/L^2)
P_{total}	= maximum total pressure	(F/L^2)
P_1, P_2, P_3	= steady state operating pressures	(F/L^2)
Q_1, Q_2	= volumetric flow rates	(L^3/T)
sec	= seconds	(T)
S	= stress	(F/L^2)
S_D	= dynamic stress	(F/L^2)
S_{total}	= maximum total stress	(F/L^2)
t	= time	(T)
t_0	= pressure surge duration	(T)
t_1	= rise time	(T)
$\bar{\bar{T}}$	= pipe wall thickness	(L)
V	= velocity	(L/T)
V_o	= initial velocity	(L/T)
V_1,V_2, V_3	= velocity in different pipe sections	(L/T)
ΔV	= change in velocity	(L/T)
V_s	= slug velocity	(L/T)
V_0	= initial velocity	(L/T)
W	= mass flow rate	(M)
γ	= weight density	(F/L^3)
ρ	= mass density	(M/L^3)
τ	= period of vibration = 1/frequency	(T)
ν	= Poisson's ratio	(-)

APPENDIX A: WATER MAIN FAILURES DUE TO WATER HAMMER – A BILLION-DOLLAR-A-YEAR-PROBLEM (Leishear [10])

Ground breaking research hammers out new methods to stop hundreds of thousands of piping failures per year that have been misdiagnosed and misunderstood for more than a century.

This research centers on two original theories. The first theory was the "dynamic stress theory" that was published in a 15 year series of publications, which culminated in an ASME press book titled, "Fluid Mechanics, Water Hammer, Dynamic Stresses, and Piing Design", by R. A. Leishear (American Society of Mechanical Engineers). This book proved that the pipe stresses that cause pipes to break had been improperly calculated by engineers for centuries. The basic equation that they were using needed to be multiplied by four to obtain the actual stress to cause failures in pipes. This multiplier is known as a dynamic load factor (DLF), where the resultant dynamic pipe stresses may be referred to as "Leishear Stresses". Engineers may simply calculate the static pipe stress that would occur by pressurizing pipes, and multiply that static stress by four to obtain the maximum dynamic stress. This theory has been recently applied to show that water main failures are caused by water hammer.

Briefly consider an example of the dynamic stress theory that was investigated at the Savannah River Site, which operates a nuclear waste storage plant in South Carolina, along with other facilities. At that plant, a cooling system prevents the boiling of radioactive liquid waste that is stored in nearly 50 storage tanks with tank volumes near a million gallons each. Rows of vertical serpentine cooling coils were installed in most of these tanks, where the coils were two inches in diameter and several hundred feet long. More than 200 of those coils were removed from service when they leaked, where corrosion was assumed to be the cause of coil failures in those tanks since 1970. However, radiation resistant cameras were used for the first time in the 1990's to observe the coils, where cracks and not corrosion were the causes of pipe leaks (Fig. 28).

Also in the 1990's, underground pipes in that cooling system began to crack and leak, where there was no corrosion at the pipe cracks. The cause of the cracks was unknown, where ASME Piping Standards used static stress calculations, which of course showed that the pipes could not crack since the equations were, and still are, incorrect. Extensive research was performed to invent the dynamic stress theory, where advanced calculus (fourth order differential equations) was used to derive new equations, and these equations were shown to be correct through experimental validation.

In short, water hammer occurs when valves and pumps are operated, where pressure waves are transmitted along the bores of pipes at near sonic velocities. These pressure waves pressurize the piping and DLF's can be used to determine the dynamic pipe stresses that cause fatigue cracks. Fatigue cracks occur when pipes are subjected to water hammers again and again. Having identified the source of piping cracks, corrective actions were implemented, and pipe failures stopped fifteen years ago, where 40 years of failures were abruptly brought to a halt. The causes of pipe cracks were identified and stopped by using this one-man theory!

The second original theory from this research is the finding that nearly 250,000 pipe failures per year in the U.S. and Canada are caused by water hammer, where this opinion is proven by new research that is in publication (Water Main Breaks, Water Hammer, Corrosion, and Fatigue Failures, R. A. Leishear). That research approximated piping failure stresses due to fire hydrant operations, which were performed annually (Fig. 29). Although there are many other sources of water hammer in water mains (e.g. any pump and valve operations), this investigation showed that water hammer cracks piping in many systems.

Additionally, a 2011 study was performed by the University of Utah to determine the cause of piping failures. Seventy percent of water main failures were from cracks, and 28 percent of water main failures were attributed to corrosion. According to the work presented here, water hammer causes cracks in the absence of corrosion, and cracks due water hammer also initiate most, if not all, corrosion failures, where water hammer cracks the pipes and provides a moisture source to accelerate underground corrosion, and corrosion increases the size of leaks. Leaks percolate to the ground from both cracks and corrosion accelerated cracks, where soil can then be carved out to cause large leaks to the ground. Additionally, all underground pipes are cracked by water hammer regardless of material, e.g., PVC, and HDPE plastics, steel, ductile iron, cast iron, and concrete. In other words, water hammer causes 98% of the water main failures in America, and thousands, perhaps hundreds of thousands, of other piping failures occur in other industries and countries around the world!

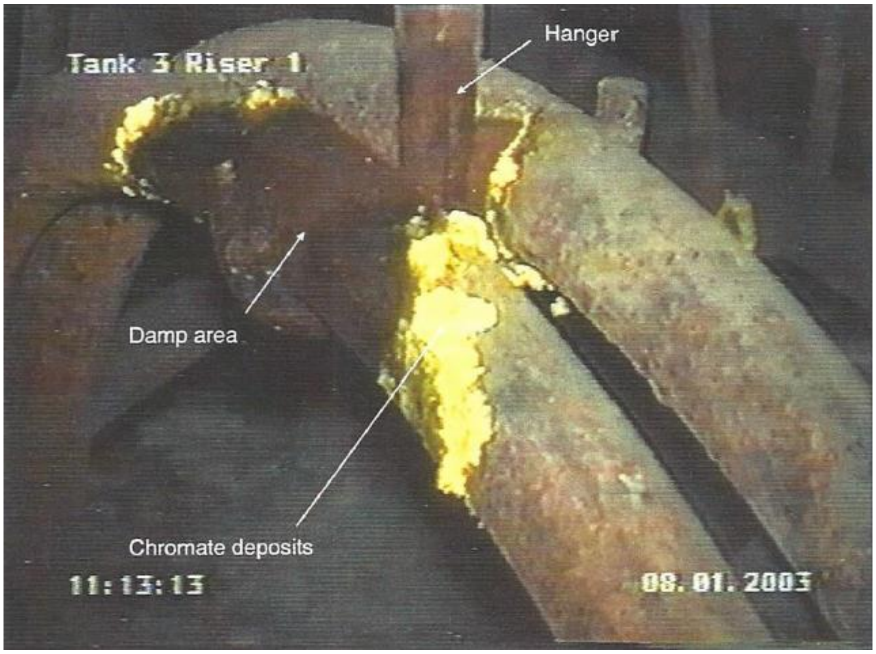

FIGURE 28: TYPICAL SAVANNAH RIVER SITE COOLING COIL FAILURE
Note that hundreds of piping failures in multiple tanks stopped when water hammer was stopped.

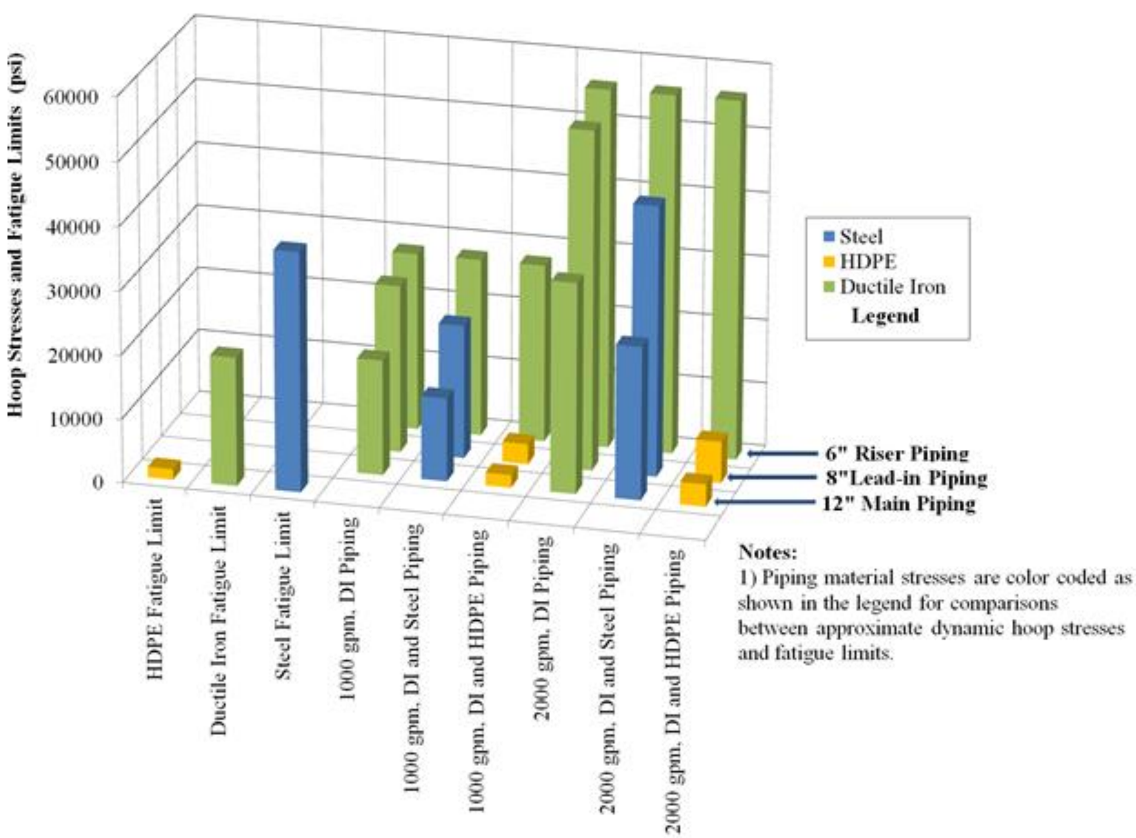

FIGURE 29: AN ANALYSIS OF WATER MAIN FAILURES DUE TO FIRE HYDRANT OPERATIONS
Note that fatigue limits are exceeded for many examples of fire hydrant operations, where water mains are cracked by cyclic fatigue.

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