Understanding Waterhammer in Pumping Systems and Surge Suppression Options

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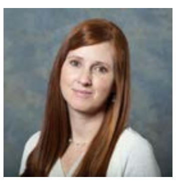

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ABSTRACT

Waterhammer in pumping systems is an area of frequent concern to designers and operators. Waterhammer has the potential to cause catastrophic failures if not properly addressed. System designers have multiple options to mitigate waterhammer in the basic design. These include pipe system design, check valve selection and motor or pneumatically operated valve selection and actuation. In some cases, surge suppression is required. Suppression options include surge vessels, relief systems, vacuum breaker valves and air release valves. This paper provides an overview of the waterhammer phenomenon, resources for engineers to assess waterhammer issues at the design stage and, for problematic systems already in operation, the issues involved with various surge suppression options.

INTRODUCTION

Waterhammer (also known as surge) occurs when fluid velocity is changed by actions such as valve position changes and planned or unplanned pump trips. Little guidance exists in codes and standards, and accidents are more frequent than we would like to admit. It is the purpose here to summarize existing knowledge and practice on waterhammer, discuss the abilities and limitations of commonly used calculation methods, provide warnings on what may happen when systems experience phenomena such as transient cavitation and liquid column separation, and give some high-level guidance on how to solve surge issues in pumping systems.

Waterhammer is fundamentally the same phenomenon across all industries which need to transfer fluids. However, depending on the nature of the fluid (benign, toxic, flammable, biologically active, etc.) and nature of the application (high pressure, proximity to people, remotely located such as in Space) different concerns and strategies are involved. It is essential that engineers take proper precautions in their design and operations to ensure safe operation of pumping systems.

APPLICABLE CODES AND STANDARDS FOR WATERHAMMER

Unfortunately, very little guidance exists from codes and standards on waterhammer. In practice, engineers are expected to use judgement and experience. Here are two excerpts from ASME piping code:

ASME B31.4: “Surge calculations shall be made, and adequate controls and protective equipment shall be provided, so that the level of pressure rise due to surges and other variations from normal operations shall not exceed the internal design pressure at any point in the piping system and equipment by more than 10%.”

ASME B31.3: “In no case shall the increased pressure exceed the test pressure used under para. 345 for the piping system.” And “Occasional variations above design conditions shall remain within one of the following limits for pressure design: Subject to the owner’s approval, it is permissible to exceed the pressure rating or the allowable stress for pressure design at the temperature of the increased condition by not more than 33% for no more than 10 hr at any one time and no more than 100 hr/yr or 20% for no more than 50 hr at any one time and no more than 500 hr/yr.”

ACCIDENTS AND SAFETY ISSUES

It is the authors’ experience that the vast majority of waterhammer accidents go unreported. In most industrialized countries there are reporting requirements to the authorities when there is a fatality or injury. In cases where there is an environmental impact or a visible and significant impact from the accident (e.g., a release of a toxic, flammable or otherwise dangerous or undesirable fluid, or possibly damage from an excessive amount of water), the facility or pipeline owners are usually required by their authorities to address the accident. Finally, there is informed speculation by some specialists that repeated waterhammer events over many years can cause fatigue damage and failure which never gets properly attributed to waterhammer due to the lack of a clear cause and effect relationship because of the deferred nature of fatigue.

For examples of the preceding, see U.S. Department of Energy (2006) for a discussion of a fatality, Nennie et al. (2009) for a discussion of an environmental impact, and Leishear (2018) for a discussion of fatigue failure from waterhammer. See Karney (2018) for a discussion of many issues in municipal water systems.

Even in cases where there is no impact to human or environmental safety, waterhammer can cause damage to piping, pumps and other equipment, pipe supports and insulation. This can lead to significant expense to system owners due to repairs and loss of production during system downtime.

WATERHAMMER OVERVIEW

Waterhammer is a broad term that encompasses fast pressure transients as a result of a rapid change in liquid velocity. The liquid velocity change can be caused by three fundamental mechanisms:

1. Liquid-full system where there is a planned or unplanned change in equipment or component operation
    a. Examples:
        i. Pump trips and starts
        ii. Valve closing or opening

2. Liquid or vapor system where there is a rapid phase change which causes a change in volume which then accelerates a liquid slug
    a. Example:
        i. Condensation of a vapor which creates and/or augments a liquid slug

3. Liquid/gas two-phase flow where differences in velocity can cause liquid slugs to impact equipment and elbows
    a. Examples:
        i. Oil and natural gas near extraction points and put into a common pipeline before separation
        ii. Starting a pump into an evacuated, air-filled line
        iii. Air trapped in storm water systems where free surfaces exist and liquid is accelerated due to movement of the air

The second and third types of waterhammer are important but not as well understood as the first. The analysis methods and tools for these types of waterhammer are typically complicated and expensive. Merilo (1992) offers a voluminous discussion of the first and second types of waterhammer mostly relevant to the power generation industry. See Klaver et al. (2018) for a discussion of the third type of waterhammer in storm water systems. This current paper focuses on the first type of waterhammer in liquid-full systems.

Note that in the English language the word "surge" is an alternate and synonymous term for waterhammer. This term will be used interchangeably in this paper.

Instantaneous Waterhammer: The Joukowsky Equation

Most engineers who have encountered waterhammer in their careers have heard of the Joukowsky equation. This equation is also known by different names such as the "Basic Water Hammer Equation", the "Instantaneous Water Hammer Equation" and the "Maximum Theoretical Water Hammer Equation". This equation relates the change in piezometric head or pressure resulting from an instant change in velocity (which is often conceptualized as an instant valve closure). Joukowsky originally published his equation in 1900 in the form of piezometric head:

$$\Delta H_J = - a\Delta V/g \tag{1}$$

It can also be presented in terms of pressure rise:

$$\Delta P_J = - \rho a\Delta V \tag{2}$$

An important parameter in Eqs. 1 and 2 is the wavespeed, a. This represents the propagation speed of pressure events in a pipe system and is typically near the speed of sound of the liquid – on the order of 1,000-4,000 ft/s (300-1,300 m/s). Recognize the negative sign on the right-hand-side of Eqs. 1 and 2. It says that an instant pressure *rise* results from an instant *drop* in velocity. Conversely, an instant rise in velocity will cause an instant drop in pressure.

The Joukowsky Equation is often understood in industry to be a worst-case, conservative prediction of maximum possible pressures. This is an unfortunate and potentially dangerous misunderstanding held by many otherwise very experienced engineers. Walters and Leishear (2018) discuss several situations where the Joukowsky Equation is not conservative as well as many of the hidden assumptions behind this equation. These situations include vapor collapse after transient cavitation, pressure wave reflections from equipment and configuration changes, and a phenomenon known as line pack where frictional pressure loss is converted into static pressure rise. Interestingly, some of these shortcomings were understood and published by Joukowsky himself but are not generally known.

An example of applying the Joukowsky equation is given in Example 1 from Walters and Leishear (2018). The fluid is oil (S.G. = 0.9) flowing in a 1.64 ft (0.5 m) diameter pipe at a flowrate of 6,629 gpm (1,437 m^3/h) with a wavespeed of 4,236 ft/s (1,291 m/s) which is instantly stopped at the downstream end. The change in velocity can be shown to be -6.68 ft/s (-2.04 m/s). The resulting pressure rise from Eq. 2 is:

$$\begin{aligned}\Delta P_J &= -\rho a\Delta V = - (56.2 \text{ lbm/ft}^3)(4236 \text{ ft/s})(-6.68 \text{ ft/s})(1 \text{ ft}^2/144 \text{ in}^2)*(\text{lbf-s}^2/(32.174 \text{ lbm-ft})) \\ &= 343 \text{ psi } (2{,}367 \text{ kPa})\end{aligned}$$

Communication Time in Pipe Systems

The second most common concept in waterhammer after the Joukowsky instantaneous pressure rise concept is that of communication time. The communication time is the time it takes for a new transient event to communicate its existence to other parts of the pipe system and, importantly, to get a response from the system. It is well known that finite duration, non-instantaneous transient events whose duration is less than the communication time are, in effect, instantaneous.

The communication time is given by:

$$t_c = 2L/a \tag{3}$$

where L is the pipe length and a is the wavespeed discussed previously.

Table 1 shows Eq. 3 applied to the minimum closure time of a valve in systems of various length in order to avoid the Eq. 2 pressure rise. Note that for a 200,000 ft (37.9 mi, 62 km) pipeline the communication time is 100 seconds. Hence closing any valve in under 100 seconds will result in the Eq. 2 pressure rise in the pipe.

Table 1: Communication Time (Eq. 3) For Minimum Valve Closure Time in Water Pipeline

a (wavespeed)		**L (length)**		t_c
ft/s	**m/s**	**ft**	**m**	**sec**
4,000	1,220	200	61	0.1
4,000	1,220	2,000	610	1
4,000	1,220	20,000	6,100	10
4,000	1,220	200,000	61,000	100

Waterhammer Physics in a Frictionless Single-Pipe System

Fig. 1 shows the progression of how a waterhammer wave progresses and reflects twice during the first $4L/a$ seconds after an instant valve closure in a frictionless, constant-diameter pipe.

For Times t < 0

The pipe in Fig. 1 is flowing in a steady-state through the valve.

At Time t = 0

The valve in Fig. 1 closes instantly.

Phase A: For Times Where 0 < t < L/a

During Phase A at the upper left in Fig. 1 one can see how the waterhammer wave propagates towards the reservoir at the left. A pressure rise equal to the Eq. 2 Joukowsky pressure occurs at the wavefront. One interesting fact observed in this case is that when the valve is closed, the fluid velocity to the left of the wave is still flowing into the pipe against the closed valve. The pressure in that section of pipe remains at the initial, steady-state pressure. As the wave moves from left to right, it brings the fluid to a zero velocity and the pipe pressure rises to that from Eq. 2.

At Time t = L/a

The wave reflects off the constant pressure reservoir. The entire pipe is under high pressure at this point with zero velocity.

Phase B: For Times Where L/a < t < 2L/a

During Phase B at the upper right in Fig. 1 one can see how the waterhammer wave propagates to the right after it has reflected from the reservoir at the left for the first time. After the wave reflects from the reservoir, the fluid velocity reverses and is equal in magnitude to the initial, steady-state velocity – but just negative now. As the wave propagates to the right, it brings the fluid from a zero velocity to a negative velocity. Further, the wavefront brings the pressure down from the Eq. 2 Joukowsky pressure back to the initial steady-state pressure.

At Time t = 2L/a

The reflected wave reaches the valve for the first time. This is the first moment that the valve is informed that the reservoir is available to relieve the high pressure. Prior to this the valve and reservoir were not able to cooperate in bringing the system to equilibrium. After this point a two-way communication has been established between the valve and reservoir. This is the communication time from Eq. 3.

At this point the entire pipe is under reverse velocity and when the wave reaches the valve, the pressure inverts and now begins a cycle of pressure *reduction* from the steady-state pressure.

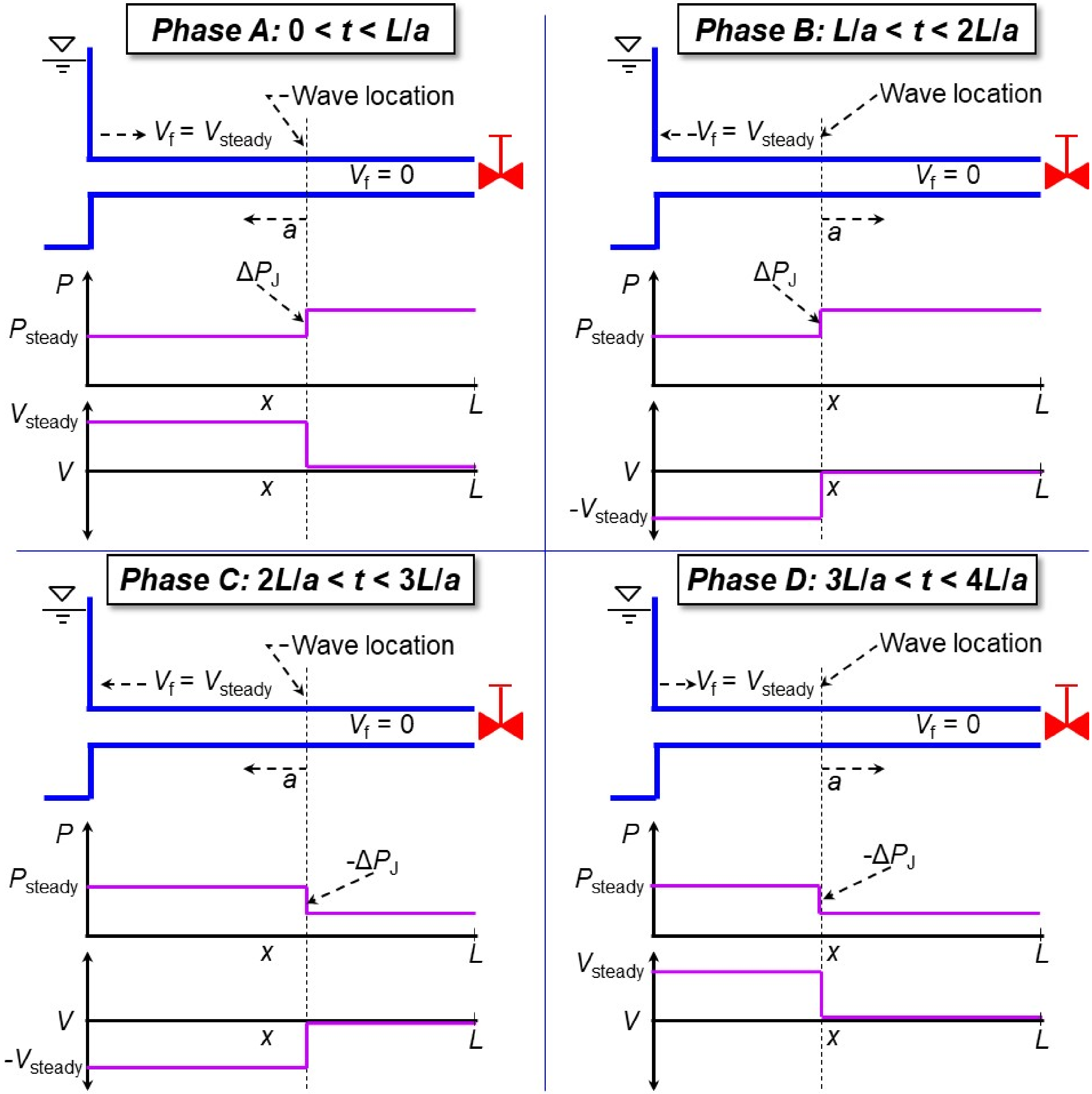

Figure 1: Sequence of steps of waterhammer wave propagation after an instant valve closure at *t* = 0.

Phase C: For Times Where 2L/a < t < 3L/a

During Phase C at the lower left in Fig. 1 one can see how the waterhammer wave propagates towards the reservoir at the left. The pressure now is reduced to a value equal to the negative of the Eq. 2 Joukowsky pressure as the wavefront passes. As the wave moves from left to right, it brings the fluid to a zero velocity from the previous negative velocity.

At Time t = 3L/a

The wave reflects off the constant pressure reservoir for the second time. The entire pipe is under low pressure at this point and the fluid velocity everywhere is zero.

Phase D: For Times Where 3L/a < t < 4L/a

During Phase D at the lower right in Fig. 1 one can see how the waterhammer wave propagates to the right after it has reflected from the reservoir at the left for the second time. After the wave reflects from the reservoir, the fluid velocity reverses and returns to its initial, steady-state value. As the wave propagates to the right, it brings the fluid from a zero velocity to a positive, forward velocity. Further, the wavefront brings the pressure up from the negative of the Eq. 2 Joukowsky pressure back to the initial steady-state pressure.

At Time t = 4L/a

The reflected wave reaches the valve for the second time. The system is in the same exact condition as at time = 0 and the entire four phase process repeats itself. In an idealized, frictionless system it will repeat the four phases indefinitely. In a real system with friction, the pressure wave will decay with the passing of each Phase until the pipe reaches a new equilibrium – the entire system at rest (zero velocity) and the pressure in the entire pipe at the same value as produced by the reservoir liquid head.

In a real system with a more complicated network of pipes, more complicated wave patterns develop which are difficult to predict using simple hand calculations or spreadsheets. As a result, many engineers use modeling software to understand the transient response. Some engineers, in the interest of time, just use Eq. 2 and add safety factors to account for uncertainties.

WHY PERFORM WATERHAMMER CALCULATIONS?

Engineers may quickly decide that the main reason for calculating waterhammer is to understand the maximum pressures so adequate pipe strength and wall thickness can be selected to avoid bursting the pipe. But this is only part of the story. Here are some of the many reasons engineers may want to calculate waterhammer:

- Predict maximum pressures to avoid bursting the pipe
- Predict minimum pressures to avoid large diameter pipes being crushed by vacuum conditions inside the pipe
- Ensure proper operation of equipment such as maintaining adequate NPSH to operating pumps during the transient
- Predict maximum or time varying imbalanced forces in the pipe system so pipe supports can be adequately designed
- Ensure forces on components such as closed valves do not exceed the components' rated maximum value
- Predict if the fluid reaches the vapor pressure and begins to cavitate – as discussed later this can generate even greater pressures after the cavities eventually collapse
- Predict minimum pressures to avoid pulling a vacuum on the pipe and thus potentially contaminating from outside ambient conditions a treated or a specialty fluid (e.g., treated water in a municipal distribution system)
- Size and locate surge suppression devices and systems to minimize high and low waterhammer pressures

WATERHAMMER EQUATIONS

The two partial differential equations typically used to calculate waterhammer are as follows:

Mass conservation:

$$\rho a^2 \frac{\partial V}{\partial x} + \frac{\partial P}{\partial t} = 0 \tag{5}$$

Momentum conservation:

$$\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{\partial V}{\partial t}+g\sin(\alpha)+\frac{fV|V|}{2D}=0 \qquad (6)$$

These equations can be solved by various methods. Since the introduction of the digital computer the most popular method has been the Method of Characteristics. For brevity this will not be explored further here. See Wylie and Streeter (1993) for a thorough discussion of this topic. Other methods exist such as the Wave Characteristics Method (Ramalingam, Lingereddy, and Wood, 2009). These methods can be programmed by engineers or accessed from commercially available packages.

WATERHAMMER IN PUMPS AND SYSTEM COMPONENTS

Waterhammer During Rotodynamic (Centrifugal) Pump Trips and Starts

Pump trips are comparatively easy to predict compared to pump starts for reasons that will be explained. For pumps with check valves at the discharge, there is no or little reverse flow through the pump during a transient. The closing of a check valve after a pump trip can be extremely significant and will be discussed briefly in the section on check valves.

Pumps will begin to spin down after a trip. To predict this spin down the rotating inertia (commonly called ωr^2) must be known or estimated. A torque balance on the pump using Newton's Second Law of Motion can be performed during each time step to predict the pump speed over time.

Some pumps do not or cannot have check valves at the discharge. In two cases the pumps will experience reverse flow and possibly reverse rotation after a trip:

- A rising main, where gravity will pull the discharge fluid back towards the pump
- Parallel pump operation where one pump trips and the others remain in operation

In order to predict pump behavior under reverse flow and/or rotation, pump performance data is needed for reverse flow and/or rotation conditions. This is known as four quadrant pump data. A good review of the methods for estimating such data and performing waterhammer calculations are given by Walters, Lang and Miller (2018).

For pump startups all the previous discussion of pump trips remains valid. Instead of trying to predict pump speed decay vs. time, one is now attempting to predict pump speed increase vs. time. The added complication for pump startups is the addition of the driver – typically a motor. The motor/driver adds torque to the torque balance and a thorough analysis of this requires motor/driver torque data vs. the speed of the motor. This makes pump startup predictions more complicated than trips.

Some designs include a flywheel on the pump to slow down its transient response – see Fig. 2.

Waterhammer During Positive Displacement (PD) Pump Trips and Starts

The authors are not aware of any good information on predicting PD pump waterhammer transients. It is typical to assume for surge calculations the pump operates in steady-state at a constant flow rate, and the transient is a linear change in flow rate with time. The time it takes the pump to trip or start will depend mostly on the inertia of the moving elements, and the size. It is common to get an anecdotal trip or startup time from the manufacturer. Common assumptions for startup or trips are that they occur in 1-5 seconds.

Note that this is a different issue than pulsation which is related to waterhammer. See Blanding and Walters (2016) for more on PD pump pulsation. Note that there is a broad array of PD pump designs not all types of PD pumps pulsate.

Figure 2: Pumps with flywheels for surge protection.

Waterhammer and Throttling Valves

Throttling valves, often motor or pneumatically actuated, are often a significant cause of waterhammer. Manual valves can of course cause waterhammer as well. In order to calculate waterhammer pressures, the important item to know is the valve C_v profile over time. Resulting surge pressures can vary significantly with different valve C_v profiles. The valve C_v profile over time is a function of two things:

1. Valve C_v vs. position
2. Valve actuator position vs. time

Valve Types

Different valve types have inherently different C_v profiles vs. position. Fig. 3 shows some examples.

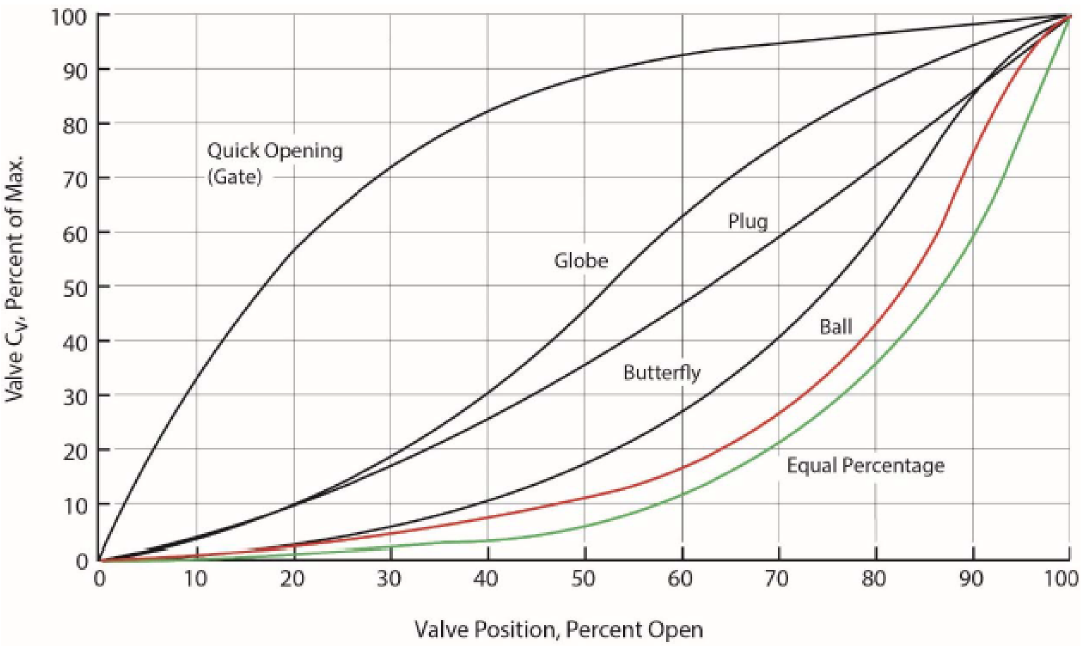

Figure 3: Example valve C_v profiles vs. position for various types of valves. (Valmatic, 2018a, used with permission)

Actuator Types

There are different types of actuators which change valve position with time. See Fig. 4. Some actuators have control features whereby the closing profile can be programmed.

The impact of a valve closure on pump system surge pressures depends on the length of pipe involved. For a shorter pipe, the valve pressure drop is a larger percentage of the overall pressure drop. For longer pipes, it is smaller. This affects how the piping and valve interact from a surge point of view.

Valve C_v Profile at Closing

It is often the case that the valve C_v profile as it approaches closing is the dominating factor on peak surge pressures. In Fig. 3, look at the lower left of the chart and see how different valves have different slopes as they approach closure ($C_v = 0$). A gentle slope usually leads to lower surge pressures.

Swaffield and Boldy (1993) discuss how surge can be reduced by changing the actuation on a valve and closing it 80% of the way in the first 20% of the time, then closing it the remaining 20% in the final 80% of the time. This principle was used successfully by Witte, Jackson and Walters (2018) to solve numerous surge problems in a large marine fuel oil facility (discussed as a Case Study in a later section).

If actuation cannot be changed, then an alternative is to design two valves in parallel – one large valve and one small valve. The large valve is set to close quickly while the small valve is set to close slowly. This allows the large valve to slow down the bulk of the fluid quickly while the small valve slows down the remaining fluid velocity much more slowly. The effect is similar to Swaffield and Boldy's "80/20" recommendation.

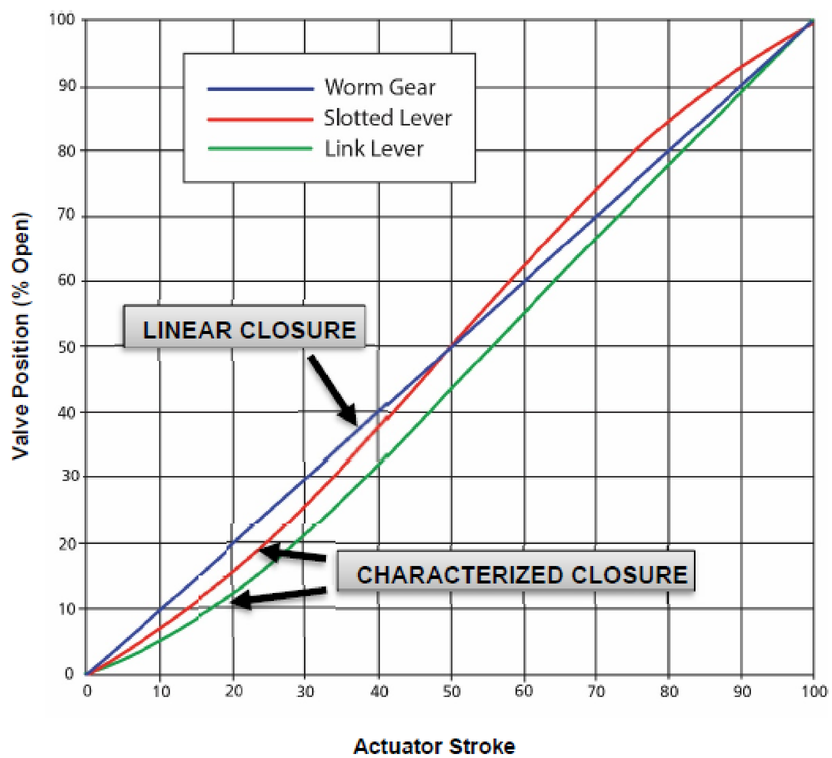

Figure 4: Actuator position vs. travel for various types of actuators. If the actuator stroke is linear with time, then one can put time on the x-axis here. (Valmatic, 2018b, used with permission)

Waterhammer and Check Valves

Check valves are a notorious cause of large waterhammer pressures when they slam closed. Swing check valves are considered to be the worst offenders (see Lozano, Bosch and Walters, 2018, discussed later as a Case Study). There are two basic approaches to predicting check valve behavior during waterhammer. One is to estimate the check valve closing velocity from charts on similar types of valves.

Thorley (2004) and Ballun (2007) have developed such charts (see Fig. 5). Many check valve suppliers have become more aware of the surge pressure caused by their valves and have developed better valve performance data for waterhammer analysts to use (see Ballun). For example, Fig. 6 shows two check valves – a swing check and nozzle check valve.

The second method is to perform a fundamental torque or force balance on the valve and calculate its motion based on fundamental principles. See Wylie and Streeter (1993) for more on this.

Another resource is Adriasola and Rodríguez (2014) who developed a CSI (Check-valve Slam Index) parameter for parallel pump systems that can be estimated based on valve and pump properties.

Waterhammer in Other Types of Components

There are many other types of piping system components. Due to space limitations the reader is referred to the excellent text by Wylie and Streeter (1993).

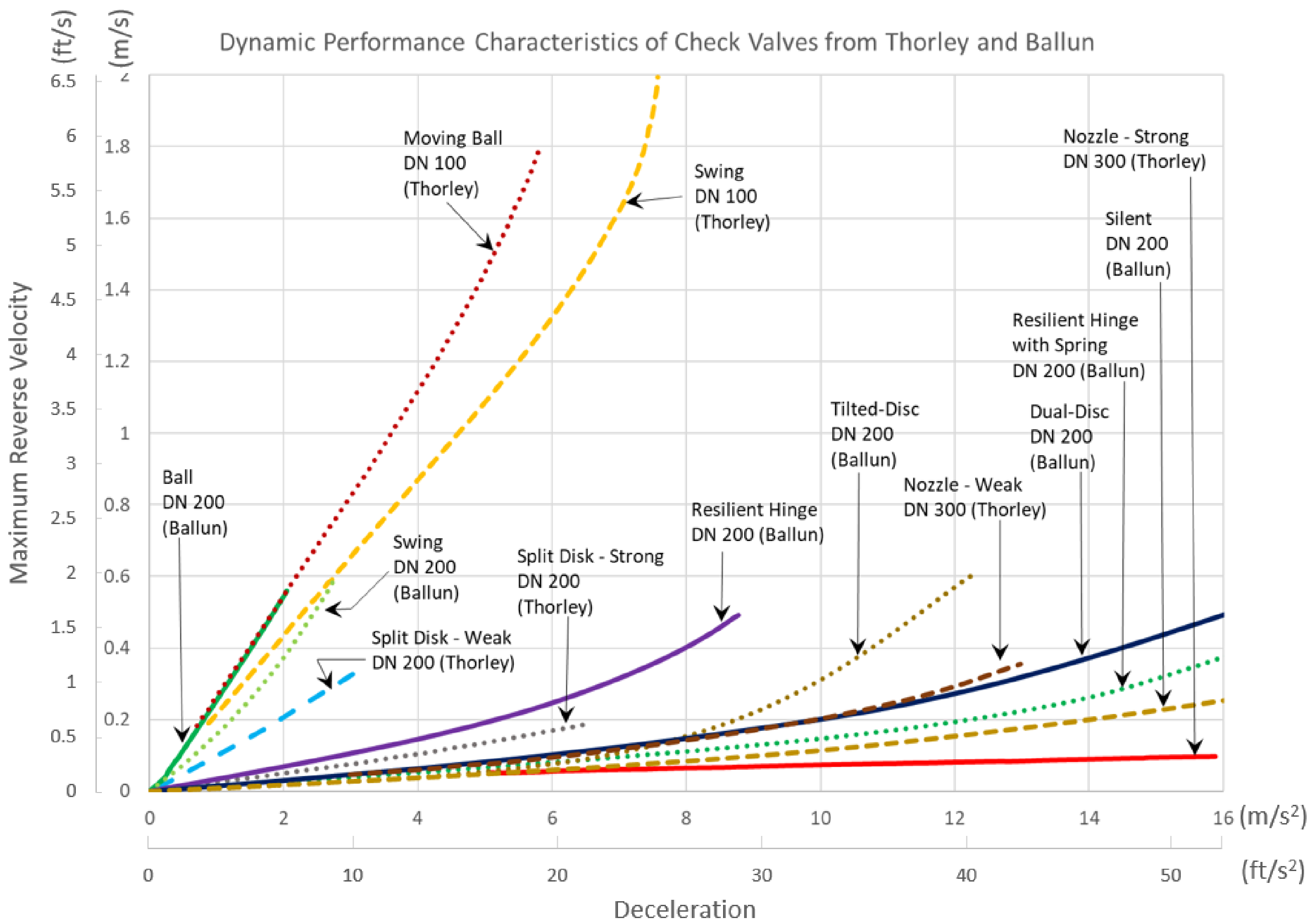

Figure 5: Check valve reverse velocity data from Thorley (2004) and Ballun (2007). Used with permission from Lozano, Segarra and Walters (2018).

Swing Check Nozzle Check

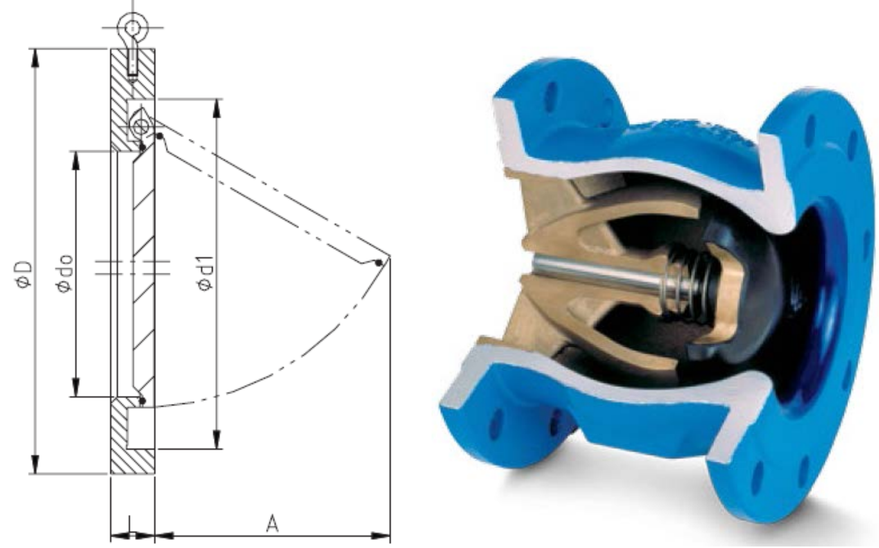

Figure 6: Swing check valve (at left) and nozzle check valve (at right). Used with permission from Lozano, Segarra and Walters (2018).

Some General Wisdom on Valve Closure Rates

Engineers typically calculate waterhammer for valve closures based on expected valve actuation rates. However, the first author is aware of several cases where a valve malfunctioned and closed virtually instantly – obtaining the full Eq. 2 Joukowsky pressure rise in systems not designed to handle that much pressure. This resulted in serious accidents.

One such accident was published as a case study by Alameria, A., and Locher, F. A. (2004). Engineers should keep this in mind as a possibility, especially in safety sensitive systems.

TRANSIENT CAVITATION AND LIQUID COLUMN SEPARATION

When a waterhammer transient reduces the fluid pressure temporarily to the fluid's vapor pressure, vapor is generated. This is called transient cavitation and sometimes liquid column separation. This is a very complicated area of waterhammer so only a brief summary will be given here.

Whenever vapor or gas is in a liquid piping system, it will impact the wavespeed of the fluid. The impact will be to reduce the wavespeed. As noted by Walters and Leishear (2018), transient cavitation can cause waterhammer pressures that exceed the Eq. (2) predictions. The two most popular methods for predicting transient cavitation are the Discrete Vapor Cavity Model (DVCM) and the Discrete Gas Cavity Model (DGCM). See Wylie and Streeter (1993) for a basic discussion of each model. See Bergant, Simpson and Tijsseling (2006) for a detailed discussion of these and other models.

It is well known that even the best transient models have various weaknesses and are susceptible to numerical noise which can generate large, non-physical pressure spikes. This makes interpretation of results a difficult process. See Stewart et al. (2018) for design guidelines for transient cavitation interpretation affecting pressure and imbalanced force predictions.

When attempting to predict imbalanced forces caused by waterhammer (discussed in a later section), the presence of transient cavitation makes this especially challenging. This is a result of the fact that imbalanced force calculation depends on properly predicting the timing of pressure waves between two locations. But when transient cavitation occurs, the wavespeed is altered (lower) and the timing thereby disrupted. There are no good methods of predicting this and Stewart et al. (2018, Part 2) provide a guideline in how to estimate the forces for design purposes.

SURGE SUPPRESSION OPTIONS

When at the design stage, engineers have more latitude in their surge suppression options. Below is a list of options available to the design engineer.

- Use larger pipe diameters
  - Since waterhammer pressures depend on velocity changes (Eq. 2) then using larger diameter pipes will reduce waterhammer peak pressures
- Use pipes with lower wavespeeds
  - Plastic pipes such as PVC and HDPE have significantly lower wavespeeds than metal pipes
  - Since waterhammer pressures depend on wavespeed (Eq. 2) then using alternate pipe materials such as plastic pipe will reduce waterhammer peak pressures
- Change how pumps and components behave over time
  - Add a flywheel to the pump to slow down its speed decay
  - Slow down valve closures or change the valve closure profile over time
- Add surge suppression equipment or systems
  - Gas accumulators
  - Surge tanks
  - Surge relief systems
  - Air inlet/vacuum breaker valves

When considering surge suppression, it is important to understand the source of the transient. For solving installed system surge problems, high frequency transient monitoring alongside computer surge modeling are two methods for discerning the root cause or potential root causes of a transient. It is also important to understand the likelihood of a surge event that is initially caused by a drop in pressure due to events such as pump trips or in some cases the downstream side of a valve closure.

Understanding the source of the transient will enable one to pick the most appropriate means for alleviating the surge pressures. For example, in the case of a valve closure, as described earlier in this paper, there is an immediate upsurge in pressure on the inlet side of the valve. On the downstream side of the valve, or in the case of a pump trip – the discharge of the pump, there is an immediate drop in pressure. Drops in pressure are not as intuitive and are often missed when deriving the root cause of transients since they are later followed by upsurges in pressure as the wave returns. In some cases where cavitation is at risk, the subsequent upsurges in pressure when cavities collapse are quite significant and become the primary focus.

Mitigating Upsurge – High Pressure Protection

In general, mitigating high pressure waves involves incorporating means in the design for accepting and dampening the kinetic energy of the pressure wave. The opposite is true for low pressure waves, where means for providing energy to the system is necessary.

Accepting or dampening energy might involve a surge vessel especially one with a gas charge. See below for descriptions of the most common types of surge vessels. It might also involve a relief valve; more specifically a “surge” relief valve that is designed to open quickly with a slow closure. When containment is not a concern (the fluid is water or otherwise not dangerous), a relief system may just spill the fluid to the ground. Containment is a concern when the fluid is potentially dangerous (toxic, flammable, biologically active, odorous, etc.) and a relief containment system will be needed to receive the fluid. Sizing relief valves can be problematic, especially when a system has multiple operating scenarios. A relief valve suitable for one set of pressures/flows may cause surge events in other operating scenarios and at the very least can be susceptible to “chattering”.

Another method for transferring energy would be to consider a more elastic pipe such as PVC pipe, previously discussed, which reduces the wave speed sometimes significantly. When considering the elasticity of a pipe, the type of pipe restraints should also be factored in as they add to the rigidity of the piping system promoting faster wave speeds.

Mitigating Downsurge – Low Pressure Protection

There are two primary means for giving energy to a system for downsurge protection: using air vacuum valves and/or surge vessels. Air vacuum valves are more common on water systems, but rarely found in non-water systems. Hence surge vessels are often the only viable option in non-water systems.

Air vacuum valves can be single stage or multiple stages when the rate of air out of the system needs to be controlled. See Fig. 7. The reason for controlling the air out of an air vacuum valve is to prevent fluid columns from abruptly merging and causing secondary surge events through a phenomenon called “air slam”. In some systems, such as petrochemical applications, introducing air into the system is not allowed for safety reasons and risk of combustion. In other cases, adding air vacuum valves make the system more susceptible to corrosion. Fluid "slugs" may also be possible in areas of trapped air where the velocity is increased due to the change in flow area. Finally, depending on the system, adding air into a system that is not subsequently released can alter the performance of the pumps. For these reasons, air vacuum valves, while necessary, are often incorporated as safety measures similar to relief valves.

The second method for providing energy to a system is through a surge vessel. Surge vessels designed to give energy to a system are no different than surge vessels that are designed to receive energy. Surge vessels are strategically sized to account for multiple surge scenarios using computer surge modeling. When possible, the design of surge vessels is also proven through transient monitoring. Below are the most common types of surge vessels:

- Open surge tanks (also referred to as standpipes) are used in processes where air exposure is not a concern. The height of the standpipe is designed such that the fluid level inside fluctuates up and down with the changes in pressure of the system. Computer surge modeling is used to discern the height of the standpipe. When more volume of fluid is needed, an actual tank can be used rather than a simple standpipe.
  - For high pressure systems open surge vessels are usually not viable, as the height of the standpipe/tank will be prohibitively tall.
  - Using a check valve in combination with a surge tank (a so-called "one-way" surge tank) can reduce the height needed for the surge tank and provide low pressure protection (however, with no ability protect against high pressures because of the check valve).
- Closed surge tanks (also referred to as compressor tanks, hydropneumatic tanks, or gas accumulators) are empty pressure vessels that are typically charged with air or nitrogen (see Fig. 8). Like standpipes, the liquid fluctuates in and out of the surge vessel. However, in closed surge tanks, the precharge pressure allows the energy to be dampened upon entering the vessel through compression of the gas. Closed surge tanks tend to be more costly as the design requires controls and a compressor (to maintain the precharge). Recharging the tanks may also require more time since there is no physical boundary between the fluid and charge.
  - Bladder type surge tanks are similar to closed surge tanks in that they are precharged, typically with air. The precharge however is held in the bladder. Bladder tanks tend to be more economical than closed tanks. There are also different bladder materials depending on fluid compatibility.
  - Dipping tube vessels are used in low pressure applications where exposure to air is not a concern. Dipping tube vessels provide more flexibility than standpipes when the height of a standpipe is not feasible. They can also be designed where "air entry" is only a last resort. Like the name, they incorporate a "tube" or pipe down the middle. The vessel volume and length of pipe is sized using computer surge software. The vessel is capable of being "precharged" to a pressure no higher than atmospheric pressure.
  - Composite dipping tube vessels are somewhat new. Composite vessels, such as those made of thermoplastic carbon fiber, are not ASME pressure vessels and are often used in conjunction with air valves when air slam is a concern. Composite vessels are less costly and lighter weight than ASME pressure vessels. They behave the same as dipping tube vessels but are limited in size.

Lastly it is important to consider the placement of the surge vessel and the line size leading to the surge vessel. In general, the vessel should be located as close to the source of the transient as possible to minimize the resistance. Line size should be considered to avoid high velocities in and out of the vessel as well as to reduce the resistance. This is especially true for bladder vessels as high velocities can lead to ruptured bladders.

IMBALANCED PIPE FORCES CAUSED BY WATERHAMMER

As discussed earlier, waterhammer waves result in temporary imbalanced forces which can cause pipes to move if not properly supported. Moving pipes cause additional secondary stresses and moments on pipe systems that can damage the piping, damage the supports, damage insulation and potentially damage flanges which can result in leaks. If the fluid is toxic or flammable, leaks are not acceptable. Thus, pipe supports must be designed to handle waterhammer events. An end-to-end methodology combining surge analysis and pipe stress analysis is discussed in Wilcox and Walters (2016).

Figure 7: Vacuum breaker valves (in green, at left) at pump discharges.

Figure 8: Closed surge vessel at a reused water pump station.

CASE STUDIES

Case Study #1: Municipal Water Pumping Station Pump Trip and Check Valve Slam

A municipal water pumping station in Barcelona, Spain, was undergoing a complete redesign. The discharge pipe was roughly 0.8 miles (1.3 km) long and rose roughly 515 ft (157 m) to the discharge tank. The engineering contractor recommended the preexisting pump discharge check valves be changed from swing check to nozzle check valves (see Fig. 6). As part of the project, high frequency pressure measurements were taken at the pump discharge and compared to a computer model of the system during a pump trip. Fig. 9 shows the pump station. Only one pump operated at a time, with the other held as a backup. Two tests were done, with one pump running and the opposite pump isolated. Fig. 10 shows measured and predicted results. See Lozano, Bosch and Walters (2018) for a more in-depth discussion.

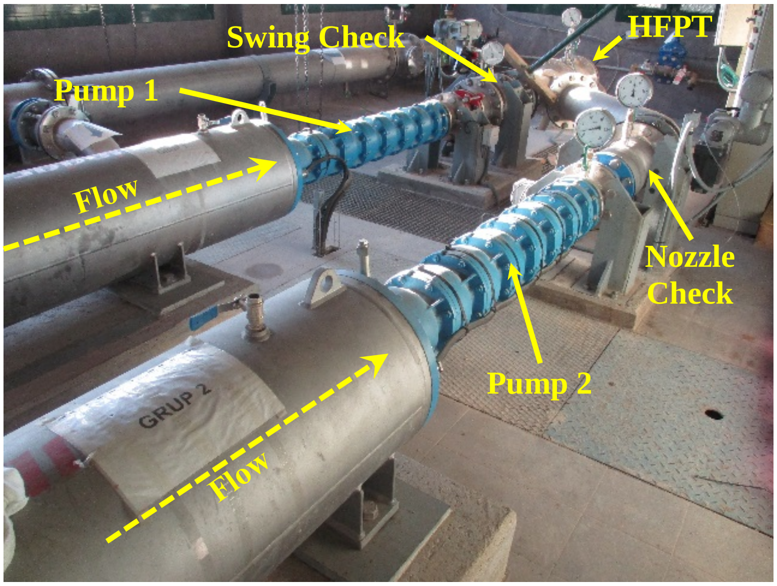

Figure 9. Pictures of the pump sets installed at Tibidabo Pump Station in Barcelona, Spain. Courtesy Lozano, Bosch and Walters (2018).

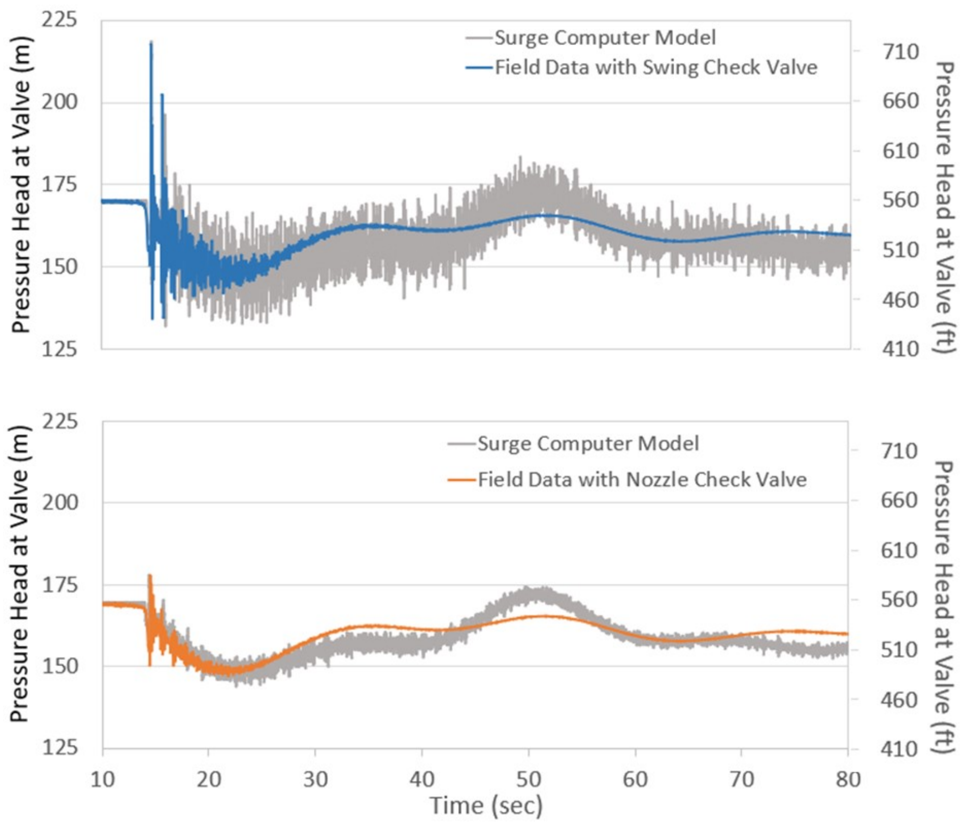

Figure 10. Measurements and surge computer model results for Tibidabo Pump Station. Swing Check (top) and Nozzle Check (bottom). Courtesy Lozano, Bosch and Walters (2018).

Case Study #2: Marine Fuel Oil Terminal Relief System

Fig. 11 shows a marine fuel oil terminal in the U.S. Gulf Coast region. There were many tanks and lines evaluated and this case study focuses on the one shown in Fig. 11. Fig. 12 shows a simplified version of the surge computer model for Fig. 11. The operation of concern was when the gate valve was closed (see Figs. 11 and 12) which caused an overpressure of the line, even with a relief system in place. Fig. 13 shows the original 75 second closure profile of the gate valve (dotted yellow curve). Fig. 13 also shows the proposed 170 second closure profile (dashed blue curve) with an alternative actuator used on the gate valve. Unlike the original actuator, the alternative actuator had the ability to control the valve position in a non-linear manner to achieve the Fig. 13 profile. Basically, it closed quickly for the first 70 seconds, then more slowly for the remainder.

Fig. 14 shows predictions of maximum pressures along the 3.8 km (2.4 mile) for various cases, with Case 3 being the selected case. The ability of the facility owner to use an alternative, non-linear actuator allowed the system to operate safely at pressures below the maximum allowable.

See Witte, Jackson and Walters (2018) for a more in-depth discussion.

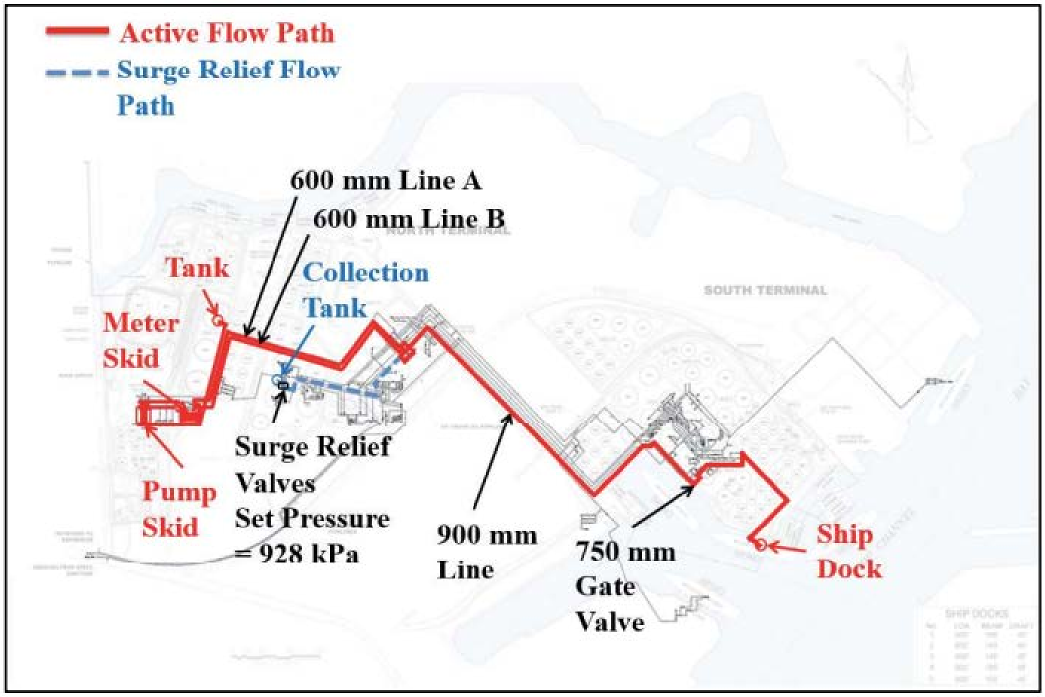

Figure 11. Marine fuel oil terminal 3.8 km (2.4 mile) long pipeline with surge relief valves. Courtesy Witte, Jackson and Walters (2018).

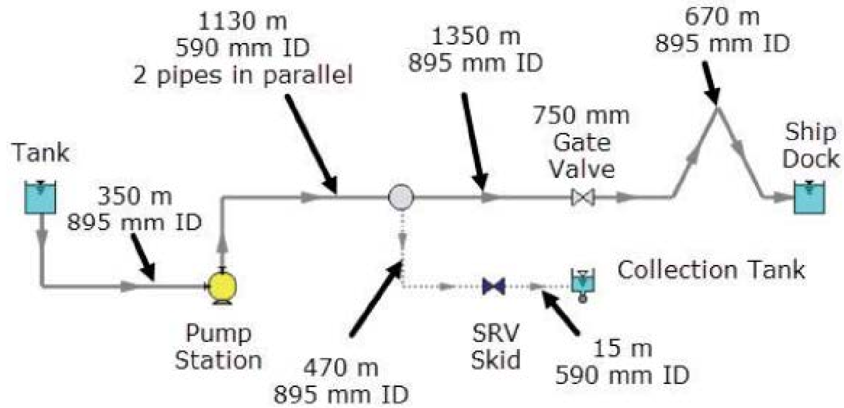

Figure 12. Marine fuel oil terminal line model with surge transient caused by closure of the gate valve. Surge relief system shown. Courtesy Witte, Jackson and Walters (2018).

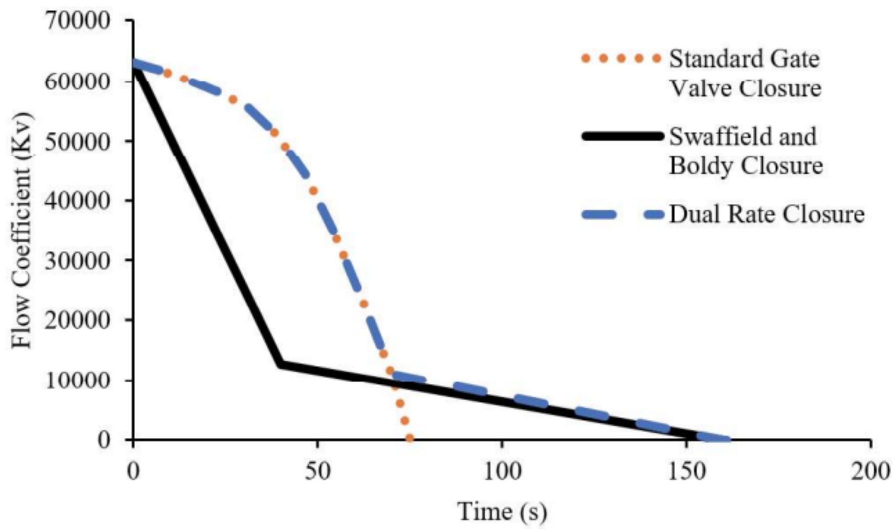

Figure 13. Marine fuel oil terminal gate valve closure profile options. Courtesy Witte, Jackson and Walters (2018).

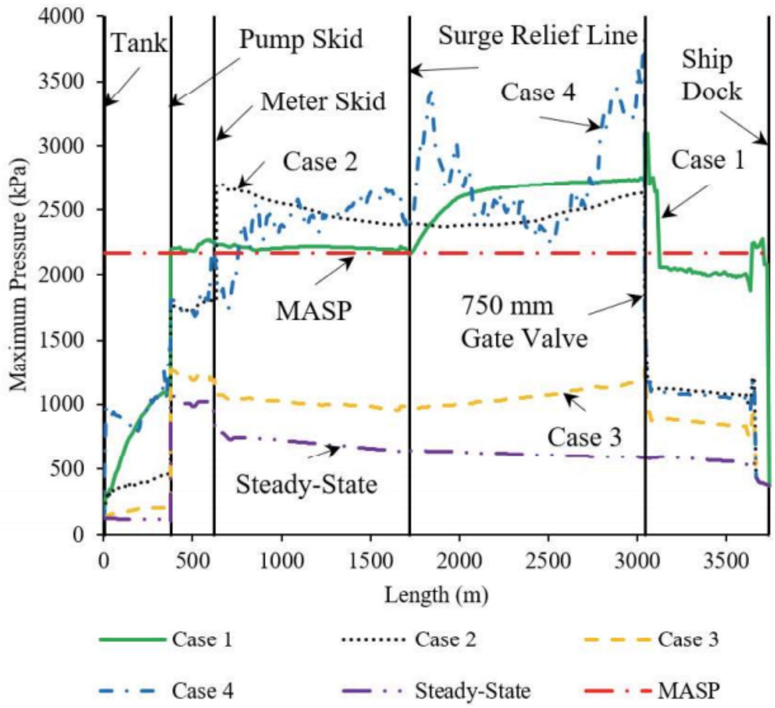

Figure 14. Marine fuel oil terminal maximum pressures for various surge relief valve and valve closure profile options. MASP (red) is Maximum Allowable Surge Pressure. Case 1 (green) was the original 75 second linear closure with a relief system (see Fig. 13 dotted yellow), but it still overpressurized the line even with the relief system. Cases 2-4 all had 170 second valve closures as in Fig. 13 (dashed blue). Case 2 (grey dotted) had no relief system and overpresurrized the line. Case 4 (blue) had a relief system but the SRV was allowed to unrealistically move positions instantly in the simulation, and it overpressurized the line. Case 3 (yellow) had a relief system with realistic movement limits on the SRV per the manufacturer and it resulted in pressures below MASP and was thus the preferred option. Courtesy Witte, Jackson and Walters (2018).

CONCLUSIONS

Waterhammer is an important issue to consider in pumping system design. It impacts maximum pressures, pipe structural support design and safety of pure fluids inside the pipes. Various options to control waterhammer exist and should be considered when unacceptable pressures result during system operations.

NOMENCLATURE

a = wavespeed
C_v = valve loss coefficient
D = diameter
f = friction factor (Darcy-Weisbach)
g = body force acceleration (e.g., gravitational)
ΔH_J = Eq. 1 Joukowsky head rise
K_v = valve loss coefficient, metric units
L = length
P = Pressure
ΔP_J = Eq. 2 Joukowsky pressure rise
r = radius of gyration of a pump
t = time
V = Velocity
x = Distance along a pipe
α = pipe angle (with respect to body force acceleration vector)
ρ = liquid density
ω = rotational velocity

Subscripts
J = Joukowsky

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