AN INTEGRATED DESIGN APPROACH FOR PIPELINES AND APPURTEANCES
BASED ON HYDRODYNAMIC LOADING

by

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A thesis submitted in conformity with the requirements
for the degree of Doctor of Philosophy
Graduate Department of Civil Engineering
University of Toronto

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ABSTRACT

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Water and wastewater conveyance research is steeply based in advancements of numerical methods and models. Design engineers need more than refinements in analysis methods to evolve the standards of practice and the related design guidelines. In an effort to improve the design efficiency and operating reliability of pipeline systems, design guidelines have been developed to enfold the various technological advancements and elevate the standard of care used in the pipeline design process. In this respect, the guidelines have been successful. However, design engineers, manufacturers, and owners have developed a level of dependency on the success of the guidelines. The guidelines, which were developed as and are clearly still held to be by the various publishing associations, a minimum standard of care, have become the default standard of care. Such statements are, of course, gross generalizations, but this thesis is dedicated to move the standard of care forward through an integrated design approach that provides a roadmap to inter-relate the independent design guidelines into a composite design approach based on hydrodynamic loading. Hydrodynamic loading introduces a temporal parameter into the design process. With the temporal parameter this work demonstrates how the consideration of both the frequency and the influence of acceleration head on the magnitude of the hydraulic loading can be used to integrate and evolve the individual component designs into a more efficient, cost effective, reliable composite design result. With a temporal parameter
present in design, many opportunities present themselves to advance the current design procedures outlined in the present design guidelines. This thesis identifies some of the present shortcomings found in the modern pipeline and appurtenance design standards and introduces a recommended path forward. Specific changes to the present standards are proposed in this work and a unique analysis procedure to identify the failure potential of cement mortar lining has been developed. Introducing the integrated design approach will allow for a significant evolution to the present standard of practice in water and wastewater conveyance system designs.
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CHAPTER 1 – INTRODUCTION

"No one wants to learn by mistakes, but we cannot learn enough from successes to go beyond the state of the art." (Petroski, 1992)

1.1 Introduction

Engineering is not a pure science, but is a judicious speculation and intervention on the interplay of multiple, complex systems in an effort to produce a safe, efficient and reliably operating composite. The composite design of a ‘standard’ pipeline conveyance system has evolved over time, with more recent refinements in the numerical analysis of individual system components and loadings. The materials used and the methods of combining these materials to form a pipeline structure are continuing to improve, resulting in improvements of material elasticity and durability. The methods of joining pipeline segments have also improved resulting in more reliable conveyance systems operating at higher pressures with less joint leakage. Pumping units and specialty valve designs have been developed to handle a wide range of operating conditions while maintaining their efficiency and reliability. The number and scope of these component improvements has inevitably developed into a long and complex list of options for the design engineer to consider. As a result, various groups, American Water Works Association (AWWA), American Society of Civil Engineers (ASCE), American Society of Mechanical Engineers (ASME), International Organization for Standardization (ISO), and others, have developed design guidelines and manuals of practice for the water and wastewater engineer to help facilitate and guide the engineer through the complexities of the modern pipeline system design process.
The goal of the various guidelines has been to elevate the standard of care used in the pipeline design process and as a result to improve the design efficiency and operating reliability of pipeline systems in general. In this respect, the guidelines have been successful. However, the continued success of the guidelines has developed into a form of dependency on them by design engineers, manufacturers and owners. The guidelines, which were developed as a minimum standard of care, have become the default standard of care. As a result, a non-standard design approach which may benefit the risk profile, the cost, and/or the sustainability of a system are not fully explored or engineered because the guidelines, as used by design engineers, do not clearly define the rationale that was used to develop the standard nor do the guidelines clearly define the level of conservativeness or promote exploration of design parameter sensitivity that may introduce a deviations from the standard of practice. Such statements are, of course, gross generalizations, but this thesis is dedicated to move the state of design and care forward: it is developed in order to identify some of the shortcomings found in the modern pipeline design standards as well as to provide a recommended path forward for various standard shortcomings via a proposed “integrated design approach” to pipeline engineering.

1.2 Thesis Objectives

In the analysis of pipeline conveyance systems, significant research has been performed in the past years in the advancement of numerical methods and models to better characterize and define various loads, in both their spatial and temporal variations, and their inter-relationship to design. However, to a design engineer in the water and wastewater conveyance industry, many times these refinements in analysis methods are not relevant nor do they drive change to the standard design practice. This resistance to evolution of the standard practice could simply be the culture
for which the standard in maintained, or could be the result of a misaligned love affair of research engineers to development of numerical theory and “minimum publishable units” over the use of physical modeling and empirically developed approaches. It is the author’s opinion that numerical theory has in many respects replaced experience, recorded knowledge and applied science and as a result the standard of practice for pipeline design is diverging from the advancements in numerical theory.

This thesis does not follow a completely standard outline. This work has been developed and written in context of the author’s 18 years of design and consultancy experience in the water and wastewater pipeline and pump system planning and design industry and 7 years as a standing, contributing member of several standard and technical committees. These standards committees are the American Water Works Association (AWWA) Standard C512 and Design Manual M51 for air valves, the AWWA C906 Standard and Design Manual M55 for polyethylene (PE) pipe, and the American Society for Civil Engineers (ASCE) Pipeline Division – Pipeline Planning and Installation Committee and sitting vice chair of the ASCE technical committee for hydraulic transient analysis and control. The author is also responsible for many of conference papers as well as several peer reviewed papers that are referenced and utilized within this thesis. Specific objectives relating to the definition of pipeline risk and the creating of an integrated design framework are outlined next.

1.2.1 Improving Definition of Risk

Understanding the increasing complexity of the pipeline design practice with regard to the amount of numerical research available and the lack of complementary empirical data, a primary
objective of this thesis is to provide a roadmap for integrating various research concepts to the published design standards and guidelines used in the water and wastewater conveyance industry. The roadmap is developed around the existing pipeline design standards and includes the introduction of several assessment tools and proposed design parameters that introduce the use of hydrodynamic loading for pipeline systems. The present pipeline design standards are integrally based in static hydraulic loading. The identification and quantification of the temporal nature of hydraulic loading is noticeably absent. There are simple and robust evaluation techniques that can be introduced and applied early in the conceptual design process to help evaluate the hydrodynamic sensitivity of a pipeline system. One example of this is the use of acceleration head to couple the physical elements of a pipeline design, namely length, with operating conditions described by the rate of change of flow, not just flow. By considering the rate of change of flow or acceleration/deceleration of flow, a temporal component is introduced to the conceptual design analysis.

The acceleration head is dependent not only on the change in velocity and change in time (temporal characteristic), but is also dependent on the distance along the pipeline. This thesis shows how an acceleration head parameter,

\[
\text{Acceleration Head} \sim \left( \frac{L}{g} \right) \left( \frac{\Delta v}{\Delta t} \right) \tag{Eq: 1.2.1}
\]

where: L is the length of the pipeline system, \( \Delta v \) is the change in velocity and \( \Delta t \) is the time in which the change in velocity occurs, can be used to inter-relate the three inter-dependent variables, change in velocity, change in time and length, so that the designer can make a measure
of hydraulic transient sensitivity of the system. The primary objective of this work is to use hydrodynamic loading to influence and evolve the design guidelines and standards that are used in pipeline and appurtenance design. The introduction of an acceleration head parameter to the various design guidelines is the first example how this work can evolve the standards of the practice of designing pipe. The acceleration head parameter can be developed to produce a conceptual level decision matrix (Table 1.2.1) that can be coupled with a decision criterion to assess if the transient behavior of the system warrants consideration of a full, comprehensive transient model. This concept is further explored in Chapter 3. The use of hydrodynamic loading, which is both time and spatially dependent, as a base for design introduces this temporal parameter into the design process. Figure 1.2.1 shows the boundaries of the temporal parameter for both the frequency and the magnitude of the loading.

Table 1.2.1: Conceptual Design Decision Matrix using Acceleration Head

<table>
<thead>
<tr>
<th>Time of Change (∆t) (s)</th>
<th>1000</th>
<th>100</th>
<th>10</th>
<th>1</th>
<th>0.1</th>
<th>0.01</th>
</tr>
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<tbody>
<tr>
<td>Instantaneous Time (t&lt;2L/a)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Length (dx) (m)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(L/g)∗( ∆v/∆t), Acceleration Head</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.002</td>
<td>1</td>
<td>0.0001 0.001 0.01</td>
<td>0.1</td>
<td>1</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>0.02</td>
<td>10</td>
<td>0.001 0.01 0.1</td>
<td>1</td>
<td>10</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>0.2</td>
<td>100</td>
<td>0.01 0.1</td>
<td>1</td>
<td>10</td>
<td>100</td>
<td>1000</td>
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<tr>
<td>2</td>
<td>1,000</td>
<td>0.1</td>
<td>1</td>
<td>10</td>
<td>100</td>
<td>1000</td>
</tr>
<tr>
<td>20</td>
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<td>1</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
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<td>200</td>
<td>100,000</td>
<td>10</td>
<td>100</td>
<td>1000</td>
<td>10000</td>
<td>100000</td>
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Figure 1.2.1: Classification of Hydrodynamic Loading

Figure 1-1A, represents a low magnitude and high frequency load, Figure 1-1B, high magnitude and low frequency load, Figure 1-1C, high magnitude and high frequency load, Figure 1-1D, low magnitude and low frequency, and Figure 1-1E, high magnitude and high frequency with full vacuum.
With a temporal parameter present in design, many opportunities present themselves to evolve the current design procedures. As a prime example to illustrate and flesh out this concept, the initial evaluation of suitable pipeline materials, linings, and joint types is explored in Chapter 3.

The introduction of a temporal parameter that correlates type of hydrodynamic loading with consequence will allow a risk component to be established for design. Presently in pressure pipeline design, safety factors or design factors are used as a measure of design risk. However, these factors fall short of accurately evaluating risk and are often inappropriately used to imply safety when they are simply a measure of ultimate strength of a material relative to an applied stress (pressure). The AWWA Standards for both Polyethylene (PE, HDPE) and Polyvinylchloride (PVC) pipe ask the designer to estimate cyclic loading and suggest the pipeline strength be de-rated after a set number of cyclic loads. PE and PVC pipeline are susceptible to failure from fatigue. However, the AWWA standards for Steel, Ductile Iron, Concrete and other commonly used material do not include a temporal component nor do they recognize fatigue in the design procedures.

1.2.2 Integrated Design Approach

At its core this thesis proposes an Integrated Design Approach (IDA) that utilizes hydrodynamic analyses as a roadmap for design procedure. The concept of using hydrodynamic analyses as a link is evident in various fluid-structure interface research projects (Wiggert and Tijsseling, 2001) that seek to define the various active and reactive forces at fluid to solid interfaces, but using the hydrodynamic analyses to bridge the component design for water pipeline design is unique. The thesis concludes by using an IDA coupled with improved monitoring, and allowing an improvement in the definitions of probability and consequence (risk), will ultimately
culminate into an Integrated Design Standard (IDS) for the water conveyance industry. A conceptual outline of the roadmap is described in the Figure 1.2.2.

![Figure 1.2.2: Integrated Design Approach](image)

The thesis focuses on design issues that the author has identified within the standards of practice, issues that provide relevant context to the divergence of theory and practice. The thesis introduces multiple, tangible concepts that can be introduced to the existing standards and design guides to promote the idea of an IDA.

Specific aspects of the IDA are elucidated in dedicated chapters. For example, in Chapter 5, the IDA is used to evaluate the standard practice of sizing and locating air valves in design of a water transmission system. An evaluation of the standard practice is made and an example
system is presently to change the paradigm used for sizing and locating air valves. The paradigm shift is based on the re-definition of the prescribed design flow rate, the deconstruction of the air valve into performance component design, and the use of transient modeling to evaluate the sizing and location.

1.3 Publications Related to Thesis Research

The core of this thesis work has been published in various places during the process of the thesis development. These published works are listed here with a description on where and how they have contributed to the work.

Chapter 1 introduces the concepts within the thesis and provides the framework from which the research was drawn. Along with the 18 years of professional experience there were several key publications that inspired the concepts and framework described herein. The need of an Integrated Design Approach became evident to the author over a number of years trying to interrelate various, many times conflicting, design allowances for different components within a pipeline system design. The concept of the Integrated Design Approach was fleshed out in several works, but mainly can be attributed to:


Along with work generated as a member of various ASCE and AWWA technical committee memberships:
2. American Water Works Association (AWWA), McPherson, D.L. as a member of the C512 Standard and M51 Design Guide for Air Valves, AWWA, Denver, CO, Work in progress at the printing of this work;

3. American Water Works Association (AWWA), McPherson, D.L. as a member of the C906 Standard and M55 Design Guide for High Density Polyethylene Pipe, AWWA, Denver, CO, Work in progress at the printing of this work;

4. American Society of Civil Engineers (ASCE), McPherson, D.L. as a Member of the Task Committee on Transient Analysis, “Hydraulic Transient Analysis in the Planning and Design of Transmission Pipeline Systems, Manual of Practice”, ASCE, Reston, VA, Work in progress at the printing of this work;

During this research and drafting of this thesis, the author was fortunate to be a member of an ASCE Task Committee on Buried Flexible Pipe Load Stability and Design. This committee included a collection of highly qualified design and manufacturing personnel who at the core had the integrity and quality of the civil engineering industry in mind. One member of this committee Reynold K. Watkins, PhD, who along with Bryan W. Karney, PhD, inspired and reaffirmed my drive to continuously and persistently improve on the standards of the engineering practice. The work that was produced and was published is:

5. American Society of Civil Engineers (ASCE), McPherson, D.L. as a Member of the Task Committee on Buried Flexible Pipe Load Stability and Design, “Buried Flexible Steel
Pipe: Design and Structural Analysis - ASCE Manuals and Reports on Engineering Practice No. 119”, ASCE Reston, VA, August, 2009

The contributing participation on the ASCE and AWWA committees, listed above also provided insights for reviews that are outlined in Chapter 2. These committee memberships have also allowed me to better understand the means and methods in which standards slowly, but continuously evolve.

Chapter 2 outlines specific analysis techniques and identifies strategic areas within the current design standards and manual of practice in which the over-arching use of hydrodynamic loading can be realized and used in the Integrated Design Approach. Several published works contributed to Chapter 2, namely:


Chapter 3 provides the roadmap for the Integrated Design Approach with context on the analysis techniques presented in Chapter 2 and where and how to apply these analyses to best inter-relate
the designed components using hydrodynamic analyses. Chapters 4 and 5 are where the thesis shows where the analyses should be applied in design. Several papers were published as Chapter 4 and 5 were being developed, namely:


During the preparation of this research and thesis, the author has prepared and published several papers through volunteer work on the ASCE Task Committee for Sustainable Design of Pipelines Pipeline Sustainability. These papers include:

12. American Society of Civil Engineers (ASCE), McPherson, D.L. as a Member of the Task Committee on Sustainable Design of Pipelines, Lead Author of Materials Section of ASCE White Paper and Workshop for “Sustainable Design of Pipeline”, ASCE National Pipeline Conference, Seattle, WA, July, 2011, Pages


The author also has participated in the first International Symposium on Hydraulic Surge and Air Control in April, 2013 and has served as a reviewer for the ASCE Journal of Pipeline Systems Engineering and Practice from 2011 to present.

1.4 Framing the research

As the organization of this work is not strictly conventional, it is worthwhile to carefully frame the work within its professional and research contexts. Two particular topics are given special emphasis both in this framing and in the work as a whole, these topics being flexible pipe design and air valve design. Many forms of analysis and calculations go into the design of a flexible pipeline conveyance system. These design processes and ultimately the pipeline products have evolved through time by successive theoretical and empirical advancements. The history of these design and production advancements has been well distilled and today is represented by a series of a succinct, yet mainly empirical design practices. The evolution of these practices has been aided not only by academia, but also by design professionals requiring quality within the pipeline industry and manufacturing. However as the existing conveyance infrastructure ages and a continual push for higher pipeline capacity is made, it is opportune to reevaluate the strengths and weaknesses of these design practices.

With consideration of the past successes and strength of the existing approaches, this thesis specifically proposes that an improved approach or prospective be used in select design
processes, calling for the consideration of a composite system or an Integrated Design Approach rather than an approach that considers the pipeline system as a rather loose collection of individual components (e.g., pipeline, joints, linings, pumps, valves, etc.) that have been engineered to “play well” together.

It is the opinion of the author that the current design practices are highly focused on the individual components and as a result have generated a prescribed narrowly focused design approach that may borderline on deterministic. Also the typical interrelationships of the individual components that affect design has been integrated so well into the component design protocols that the practicing engineer is unaware of potential risk and/or improvement that may be present if the design were broken down into the basic building blocks and reconstructed with a more inclusive and/or comprehensive objective.

1.5 Framing of the Thesis

This thesis develops an outline with specific examples of how to break down this deterministic approach that has pervaded the current design practice, and allow a re-introduction of basic conceptual design philosophy into the pipeline design process.

Pipeline design practices have evolved over time to consider static or at best step-wise advancements in the industry. There have been provisions within the design practice to allow the introduction of new material. For example, the design practice for selecting the appropriate thickness and type of polyethylene (PE) pipeline material for a pressurized pipeline application is based on a method developed by the Plastic Pipeline Institute (PPI) and is known as the Hydrostatic Basis of Design (HDB). This design procedure which is outlined in AWWA M55
This thesis proposes that both static and dynamic, not just hydrodynamic, evaluations of the pipeline linings, coatings, joints, operating conditions (both normal and abnormal), transient control systems as well as handling and installation issues be considered. In pipeline design, individual components have been researched extensively, but the interrelationship and the interdependencies of the components has not. With the general considerations presented, this thesis outlines a prospective of how and where hydraulic transient pressures should be used in pipeline system design and how and where design professionals can introduce practical and theoretical improvements to the design process by using more advanced tools and techniques.

1.6 Pipeline Design

A review of applicable standards conducted for this thesis turned up no information on the influence of dynamic loading in relation to a common pipe lining known as cement mortar lining.
which is a thin layer of cement paste that is applied to the interior of metallic pipe to reduce the corrosion potential of the pipe in the presence of water. The literature found mainly focused on the allowable deformation or deflection of the pipeline if a cement mortar lining was to be considered. This information was similar in for the information found for elastomeric seals and gaskets. The collection of empirical work performed by Reynold K. Watkins at Iowa State and then Utah State University which resulted in the Iowa Formula (Watkins and Anderson, 1999) has guided the flexible pipeline design industry ongoing 50 years; however this work is fairly crude with respect to hydrodynamic loading. The design loadings, for which internal pressure is one, are simply used to predict deflection of the pipeline. This predicted deflection is then used to establish performance limits of various design components; such as, a cement mortar lined and flexible coated pipeline is allowed a 3% deflection because of the performance limit of the lining. The loading rate is not a variable in this analysis.

To address this, a basic analysis approach (developed with brief correspondence with Ivo Pothof of Deltares) is outlined in Chapter 4 of this thesis that considers the both the loading rate and magnitude on cement mortar lining. It is proposed that this improved procedure be used to evaluate the integrity of cement mortar lining design and be included in any MOP that references cement mortar lining. The key parameter in this analysis is the hydraulic connectivity of the cement lining mixture. In summary of this approach, if the cement mixtures hydraulic conductivity is relatively low then the balancing force to equilibrate the transient pressure may occur too slowly so that a full differential loading is experienced by the lining. Therefore, during a hydraulic transient event the full force resulting from the differential pressure could be realized putting the lining integrity at risk and potentially drawing the lining into the pipeline flow.
1.7 Air Valve Design

A second context developed in this work is that of air valve design. Air valves are not generally used in the petroleum or natural gas industry due to the compressible nature of the conveyed fluid. However, water is relatively incompressible and therefore the introduction of air into and out of the system has become necessary as free air is dissolved, entrained and released within the flow during the conveyance as pressure rising and falls along the pipeline profile. The introduction of air to the system when the water is at or near vacuum pressure conditions can prevent or limit the formation of vapor cavities in the pipeline. The expulsion of air through air release valves prevents a phenomenon known as air binding from occurring. Air binding happens when dissolved air is released from the fluid and collects at localized high points or low pressure points. The volume of air, which is seeking a lower energy state and therefore a lower pressure within the pipeline system, will be held at this high point. Over time, as the free air accumulates, the increasing air volume will begin to restrict the flow of water and begin to hydraulically bind the system (Falvey, 1980). This phenomenon is well documented and has been studied extensively. In general the primary duty of an air vacuum valve is to help prevent the pipeline from collapsing and the primary purpose of an air release valve is to help maintain the full hydraulic capacity of the pipeline system. This thesis discusses additional features of the air valve that are not thoroughly addressed in the current design practices. These include:

1. Proper sizing of the air release valves with respect to the compressed air volume at the valve and the hydrodynamic loading during air release,

2. Proper sizing of air vacuum valves with respect to closure after a hydraulic transient event and prevention of a secondary transient event, and
3. The composite design of air valves for both startup/shutdown, normal and transient operations.

The framing of the research draws attention to the various components that have evolved in relative isolation to other components of water and wastewater system. The next step to formulate a resolution to this issue is to identify and refine the understanding of the interdependencies that exist within the current design guidelines, but have not been coupled in a way that can benefit the design of each component as well as benefit the entire conveyance system design.
CHAPTER 2 – A REVIEW OF EXISTING DESIGN STANDARDS

Chapter 2 provides a review of a series of commonly used North American design standards and manuals of practices that are used worldwide to design and specify pipeline systems. This review highlights the divergence in the standards as the standards become more specific for various system components rather than integrated. The issue and problematic result of this specific component design is an overall over-simplification of the design standards.

2.1 Introduction

The hydrodynamic loading of a pipeline system is not thoroughly addressed in current design guidelines and as a result is not considered in design practice. Where hydrodynamic loading is addressed, it is simply framed as a statically determinant system with no temporal characteristic. These design guidelines and approaches have been developed through years of both theoretical and empirical research. The research for loading characteristics on buried pipeline has generally focused on the soil/trench conditions and its influence on the integrity of the pipeline structure and the resistance to thrust at various fittings and deflections in the system. The review of these guidelines performed for this thesis can be grouped into two categories, the first category is a thorough review of the North American and European design standards of buried pipeline design practices and standards for the water and wastewater industry and the second is the review of applicable academic and manufacturing studies that have been performed for specific pipeline materials and appurtenances.

2.2 Pipeline Design Standards and Manuals of Practice
The American Water Works Association (AWWA) and the American Society of Civil Engineers (ASCE) are two societies that serve the civil engineering profession in North America. Both societies, in which the author is an active member, maintain committees to periodically update pipeline design practices and standards. These two societies produce manuals of practice (MOP) and design guidelines that provide brief histories of the applicable product as well as define minimum design standards for various pipeline materials and appurtenances used in the pipeline industry. In the review of these MOPs, it has become evident that each guideline has evolved into a specific guideline for an individual material and/or service. For example in the AWWA design guideline for Steel (AWWA M11, 2004) and the associated standard (AWWA C200, 2005), Ductile Iron (AWWA M41, 1996), PVC (AWWA M23, 2002) and HDPE (AWWWA M55, 2006) and its associated standard (AWWA C906, 2007), the recommended calculation for imposed thrust as well as the calculation of thrust restraint are different. Also and more important to this thesis is the fact that none of the design guidelines above considers hydrodynamic loading in the design procedure. Granted, in regard to thrust, the dynamic loading is relatively small at normal design flow velocity, but the current guidelines simply do not consider it. This makes the design less robust. For instance, if a design professional considers only static thrust in design of a steel pipeline and in that design the designer assumes that only 50% of the yield strength will be used to resist the combined bearing load and load induced by the internal pressure then the assumed factor of safety in the current design guideline is 2.0. In fact because the hydrodynamic loading is not considered in the factor of safety, the factor of safety is not 2.0 but something unspecified, but certainly less. The amount less is of course dependent on the velocity and the rate of change in the loading, but the calculated safety factor in the current standard is erroneous. The author understands that the factor of safety is meant to protect the design from unknown,
uncertain or immeasurable elements inherent in design, but the hydrodynamic load is quantifiable and therefore it should be routinely considered in the loading characteristics of the system.

The recent effort of the society’s (ASCE, AWWA) committees, that are responsible to update the various design guidelines, there has been a simple debate over which current practice is better rather than a concerted effort to research and develop resolutions to outstanding and/or misrepresented design approaches. With the current design guidelines evolving over time with different design recommendations for basic design parameters, such as internal loading magnitude, the design professional is left with an inherent conflict within the guidelines that limits the ability for the profession to further evolve or modify the design practices. Today, the framework used to design a steel pipeline is not similar to the design of a polyethylene pipeline. Even though both design procedures consider similar design parameters such as thrust, allowable deflection, and loading, the DESIGN GUIDELINEs and the method of design and the results of design are different. The AWWA M55 design guideline for Polyethylene pipeline utilizes a hydrostatic design basis for the pipe barrel which was developed by the Plastic Pipe Institute (PPI), where the steel pipeline design guideline, M11, utilizes static loading consideration and elastic limits of the material. The European guidelines which are outlined in the Information and Guidance Notes (IGN) 4-32-18 “The choice of pressure ratings for polyethylene pipe systems for water supply and sewerage duties” and IGN 4-37-02 “Design Against Surge and Fatigue Conditions for Thermoplastic Pipes.” are similar in their approach, but what is evident is the lack of a consideration of the composite system. The IGN 4-37-02 opens in the foreword stating, “Past advice on dealing with transient pressures in plastic pipelines given in the Pipeline
Materials Selection Manual (PMSM) issued for the Water Industry in 1995 by WSA, and has been recognized as being extremely conservative.” This statement would make one believe that a pipeline system conforming to these standards would be over-designed or at least highly conservative. Even using this older approach, failures and/or leaks are still evident in systems using this design approach as well as the North American design approaches.

Another issue with the current design practices is that the manufacturers are steering the research and the associated guidelines. The manufacturers are conscious of the owner’s interest in refining the design guidelines and MOPs for consideration of both initial capital cost as well as life cycle cost through continued improvement of the pipeline product (e.g., improvements in pipeline stiffness and retardation of crack propagation), but these improvements to the MOPs, which are generally driven by the manufacturers of the specific material, do not consider the pipeline system as a composite. A more comprehensive improvement to the MOPs would be to integrate the design procedure for all elements of the pipeline system not just the pipe barrel. If the MOPs were to consider the specific pipeline material with respect to the various seals and gaskets that make up the joints in the system, then a more comprehensive MOP may be obtained. Interesting enough the IGN 4-37-02 alludes to the significance of this consideration in Section A.3.2 Defining Surge, “If sub-atmospheric troughs occur, these may cause wall collapse by buckling and for a system joined by sockets and spigots there may be permanent problems with sealing ring displacements.” However, this IGN does not elaborate or reference any work with regard to this issue.
In the ASTM Standard Specification for Joints for Plastic Pressure Pipes Using Flexible Elastomeric Seals (D 3139-98) Section 6.3, the statement is made, “The gasket shall be the sole element depended upon to make the joint flexible and water tight.” In this standard in Section 8, the joint is required to pass a test under three specific positive pressure conditions without leakage. The first positive pressure requirement is 50% of design pressure rating, the second is 250% of design pressure rating and the third is minimum short term rupture requirement of the pipeline. The caveat is that the standard requires that the increase in pressure from 250% times the design and the short term rupture pressure should be made over a period of 60 to 70 seconds. This allows for a relative smooth increase in pressure with little dynamic or impulse loading consideration. As a note this standard also requires a sustained vacuum pressure of -10.9 psig (-75 kPa gage), which is not a full vacuum condition for water at sea level. The material research for the elastomeric seals is vague with regard to transient (hydrodynamic loading) and none of this research is included in the current design guidelines.

2.3 **Design Standards on the Selection of Air Valves**

The design approaches for transient control that are outlined in the AWWA and ASME design guidelines as well as the European guidelines for combination air valves do not address transient or hydrodynamic loading. The research and guidelines due reference transient considerations in the sizing, location and the operation of the valves; however the reference is simply a procedure to apply a rated flow and pressure condition to the design process. No true consideration of the hydrodynamic loading is made.

The AWWA manual of design practices for air valves, M51, is the standard design guidelines that are used in the water and wastewater pipeline design industry in the USA. This manual
references several previous works that define the allowable performance limits of air valves but does not discuss rate of loading. Two documents are used primarily in this manual of design practice. These include the 1971 ASME paper “Fluid Meters” (ASME- Fluid Meters, 1971) which defines the pressure performance limits of air valves and the Val-Matic Valve & Manufacturing Corporation work entitled “Theory, Application, and Sizing of Air Valves.” (Val-Matic, 1997) that describes location and function of various air valves in a pipeline system. Neither of these referenced works provides a quantitative nor a qualitative approach to air valve sizing and design with regard to hydrodynamic loading.

Another paper published in the 1972 AWWA Journal entitled “Locating and Sizing Air-Release Valves” and written by Lescovich provides an outline of the various design considerations for locating and sizing air valves. The paper nicely describes some design limitation of both air release and air vacuum valves with respect to hydrodynamic loading, but does not provide solutions nor steps forward to resolve these issues. The paper describes the response of the air valve during a blowback phenomenon that results when the vacuum valve releases air too rapidly after the air is drawn into the pipeline system following a vacuum pressure condition. The paper also briefly discusses the difficulty of sizing the air release and air vacuum orifices for the range of flow that the valve(s) will be required to operate as well as the rate of flow for which the valves should be sized. Both these issues are resultant of hydrodynamic forces in the pipeline system. The paper does not make recommendations for sizing the valves with respect to hydrodynamic forces; but nonetheless is a significant contribution to the subject.
Potentially, Lescovich’s paper has resulted in some unintended impacts to the air valve and pipeline industry. In the first paragraph of the paper, Lescovich writes “…, under ordinary circumstances, water contains more than 2% dissolved air by volume, and often a much higher percentage.” In Chapter 5 an argument is made that the volume of dissolved air is much lower than 2% in the vast majority of normal operating pressurized pipeline systems. However, Lescovich’s statement is frequently used as a default value of air valve volume and as a result in the current design standards 2% is used as the volumetric flow rate to size air valves. The AWWA M51, the design guideline for air valves, actually uses 2% of the fluid flow rate as the default volumetric flow rate to size both air vacuum and air release valves and makes no mention of consideration of the lower compressed air volumes. In AWWA M51, the two parameters that are variables in the analysis to size air valves are the orifice size and the allowable differential pressure. No discussion of the air volume or the rate of air flow rate is presented. The value of 2% dissolved air as used in AWWA M51 is a legacy of Lescovich’s paper and is simply an approximate of the dissolved air in a body water at atmospheric pressure.

2.4 Summary of Review of the Standards of Practice

The standards of practice that have been evaluated in this review and are widely used in design practice for water and wastewater conveyance projects throughout the world; collectively they show a clear progression towards simplification and discretization. The simplification is strongly based in the use of static hydraulic loads to simplify the structural evaluation of the stresses in the system. Due to the use to the static loads, the hydrodynamic connection has been lost in many of the standards and as a result the driving design intent of conveying a fluid has been substituted by a static structural design of individual components. Continued efforts are being made throughout academia to account for the complexity of hydrodynamic loading (Fluid
Structure Interface) and soil support/loading and structure interface, but this research has not
developed into changes in the design paradigm.

The remainder of the thesis discusses how the over-simplification of the design standards and
manuals of practice can be resolved by introducing key hydrodynamic loadings concepts to
bridge the component design while utilizing an integrated design approach to identify pipeline
system allowable limits and design constraints.

**Current Design Approaches with Regard to Hydrodynamic Loading**

The next step for this thesis is to introduce how this information, this road map, can be used and
introduced into the design standards and manuals of practice to help facilitate a more inclusive,
and value oriented design. This chapter discusses transient control mitigation and how it should
be represented in the Integrated Design Approach as a cost savings mechanism rather than be
relegated to a device whose primary purpose is emergency control and system failure prevention.
This chapter also discusses pipeline design and how hydrodynamic loading can be used to
evaluate and improve the design approach for pipeline barrel, joint and lining and coating
designs.

### 2.5 Value Based Hydraulic Transient Mitigation Techniques

A hydraulic transient condition in a pressurized pipeline system is caused by the change of the
fluid’s velocity in the pipeline. Pressure fluctuations result as the velocity changes. The
magnitude of the pressure fluctuations is indirectly dependent on the rate in which the velocity
changes and directly on the amount the velocity changes in a system. These velocity changes can
result from either normal or abnormal operating conditions in a pipeline system. Normal velocity changes can be caused by opening or closure of a valve, or normal startup or shutdown of the pump station. Abnormal velocity changes typically result from a rapid closure of an in-line valve or a sudden uncontrolled shutdown of a pump station during a power outage. Each of these situations is briefly discussed below. Section 5.2 provides a description of commonly used surge control devices. Sections 5.3 and 5.4 will provide a description of the North American design practices for flexible pipeline design and air valve design.

**Normal Hydrodynamic Loading**

**Pump Station Startup:** In normal operations, pump startups follow an orderly, predetermined procedure. When a pump is activated, a positive hydraulic transient pressure (pressure rise or upsurge) is generated in the discharge pipeline as the flow velocity increases. The magnitude of the upsurge varies with the rate of change of the flow increase. High upsurges are not normally generated by normal pump startups; as long as the flow increases steadily, the pump shutoff head is not exceptionally high, and there are no air pockets in the pipeline. If variable frequency drives (VFDs) are present, the VFDs control the startup by allowing the pumps to ramp up slowly through a two-step startup procedure. The first step of the VFD operation should ramp one pump up to a discharge head equal to the static head condition in the system and then the second step of the VFD operation should slowly ramp the pump discharge head to the desired operating head and flow condition. The author notes, however, that many engineers mistake electrical soft starts and stops as a means of hydraulic transient control. In fact, an electrical soft start and stop has little influence over the movement of the fluid. The electrical soft starts and stops regulate the flux of electrical loading and therefore resolve electrical surging. This electrical surge may cause
power outages but the electrical soft start/stop does not affect the rate of flow change in the system.

**Pump Station Shutdown:** In normal operations, pump shutdowns also follow an orderly, predetermined procedure. The shutdown procedure allows the flow rate in the pipeline to slowly decrease. This can be done by slowly decreasing the speed of a pump motor that is equipped with a VFD drive or by slowly closing a pump control valve until the valve is full closed. The timing of the VFD turndown and the valve closure is dependent on the characteristics of the system. Under these conditions, the pipeline would not experience any uncontrolled pressure drops, and therefore would not experience significant downsurge or adverse hydraulic transient pressure drops. However, an uncontrolled pump shutdown, as discussed later may result in a downsurge followed by a high upsurge in the pipeline.

**Valve Operation:** Rapid closing or opening of an in-line valve in a pipeline is a potential source of surge in a pumped system. Rapid closure of a downstream valve can generate a sudden pressure rise upstream of the valve as the flow velocity rapidly decreases. Whereas, a rapid opening of a downstream valve will cause a sudden upstream pressure drop that could then propagate throughout the pipeline system resulting in an unacceptable downsurge or vacuum pressure conditions. Under normal operation, no valve should be closed or opened rapidly.

**Abnormal Hydrodynamic Loading**

**Pump Station Power Failure:** A pump suddenly stopping due to power failure often causes the most severe surge conditions in a pumped pipeline system. The sudden stopping of a pump...
causes a rapid downsurge immediately downstream of the pump. The magnitude of the downsurge is primarily controlled by the polar (rotational) moment of inertia of the pump and motor and its influence on the rate of change of the flow velocity of the fluid. Under certain conditions, the downsurge may reach full vacuum (-14.7 psig (-101.3 kPa) @ standard temperature and at sea level), causing the water to vaporize (boil) and create vapor pockets. Subsequently, depending on the pipeline length, profile, and material, column separation may occur from a full vacuum pressure condition, resulting in the formation of vapor cavities. As time elapses after the pump failure and depending on the amount of static head in the system, the forward velocity will diminish, and flow reversal will follow. The rejoining of water columns at the location of the vapor cavity may result in the cavities collapsing explosively, thereby generating extremely high, localized pressures. This phenomenon is typically referred to as “waterhammer”. It is difficult to accurately predict the location and magnitude of vapor cavities and resulting waterhammer pressures. However, a possible consequence of inadequate design for waterhammer pressures would be the collapse of the pipeline from repeated vacuum conditions and/or the failure of pipeline joint systems resulting in significant leakage or bursting of the pipeline from a waterhammer pressure spike.

2.6 Common Surge Control Methodologies

This section summarizes commonly used surge control devices and briefly describes the proper application of these surge control devices in a hydraulic system.

**Non-mechanical surge control design:** Surge control devices are generally mechanical devices that mitigate or lessen the impact of an uncontrolled or unplanned hydraulic transient. Because surge control devices are mechanical, the surge control device itself can be susceptible to failure
if not properly designed and maintained. With this being said, the best way to control adverse hydraulic transient conditions is not through the addition of mechanical surge control devices but through inclusion of design features that prevent or reduce the severity of the hydraulic transient. If the hydraulic system can be designed to react to uncontrolled and/or unplanned hydraulic transient conditions without putting the system at risk, then these design considerations should always be made. These design features may include larger or parallel piping systems to reduce flow velocity or a pipeline route and profile selection to limit the static head and/or potential vapor cavity collapse and waterhammer effect. Use of non-mechanical design features to mitigate or prevent hydraulic transients is rarely practical due to physical constraints or high costs that render the approach unfeasible.

**Mechanical transient control design:** Because design solutions to prevent or mitigate hydraulic transients by themselves are generally impractical or cost prohibitive, mechanical hydraulic transient control devices are commonly used to mitigate and control adverse hydraulic transient conditions. These hydraulic transient control devices can be categorized into two groups, primary control devices and secondary control devices. Primary mechanical controls include appurtenances like hydropneumatic tanks, flywheels, and standpipes. A primary hydraulic transient control system consists of a control device that is always present and will immediately begin to influence the system’s hydraulic behavior once positive control is lost, such as after an uncontrolled pump shutdown. These primary control devices prevent the hydraulic transient pressures from fully forming or becoming severe. In addition, a primary control device provides a positive benefit to all components of the system including pumps, valves, pipelines and other hydraulically connected appurtenances.
Secondary transient control devices are reactionary devices that respond to transient pressures that have already formed in the system. The secondary transient control devices are more suited to mitigating or protecting the system at specific point within the system. As a result, the number of devices required to protect a system is dependent on the allowable limits of the design and length of the pipeline. This thesis includes an unique evaluation of a secondary transient control device. The evaluation of air-vacuum valves is made and a procedure to use hydrodynamic loading characteristics to site and size the valves is presented.

2.7 North American Standard Design Practices for Flexible (Steel) Pipelines

To select an appropriate wall thickness for a steel pipeline, three elements are primarily considered; the internal pressure, the external pressure, and the soil (embedment) stiffness. If these three elements are appropriately accounted for, then a pipeline can be designed to be well within the applicable design standards. Although other design considerations may influence the pipeline thickness, including the types of joint and lining systems utilized, these considerations are outside the scope of this work.

2.7.1 Internal Pressure

At its most basic level, internal pressure originates from the forces the fluid applies to the pipeline wall as a result of molecular collisions. In practice, this force arises from the pressure head which is simply computed from the location of the Hydraulic Grade Line (HGL) in relation to the pipeline’s position (elevation) along the pipeline profile. In design, once identified, the positive hydraulic transient pressure is assumed as an additive component of this internal pressure and is subsequently considered a fixed (invariant with time) pressure condition. Other
components of internal pressure are the normal and maximum operating pressures, the hydrostatic test pressure and finally the static pressure. If the system is a pumped system, then the pressure generated by a pump at the shut-off head condition should also be considered in design as an internal pressure. Figure 2.7.1 shows the internal pressure considerations as defined by M11, the American Water Works Association (AWWA) Manual of Water Supply Practices for Steel Pipe (AWWA M11, 2004). In Figure 2.7.1, note the exclusion of the downsurge or negative pressure condition.

![Gravity Flow vs Pumped Flow](image)

**Figure 2.7.1: AWWA M11 Internal Pressure Design**

A pipeline can normally be assumed as a thin walled pressure vessel having a large diameter (bore) relative to the wall thickness. For example, a 48-inch (1200 mm) diameter steel pipeline may have a thickness of 0.25-inch (6.35 mm) which is a D/t ratio of 190:1. Because the pipeline is taken as a thin walled element, in design the pipeline is assumed to have a uniform wall stress.
As a result, the internal pressures are assumed to be acting uniform and radial on the pipeline cylinder. Figure 2.7.2 shows the internal loading considered in pipeline design. The standard design process assumes:

\[ S = \frac{PD}{2t} \quad \text{Eq. 2.7.1} \]

where \( S \) is the Hoop Stress, \( P \) is the internal pressure, \( D \) is the inside diameter and \( t \) is the wall thickness. Note that in the standard design, the radial stress (normal) and the longitudinal stress are not considered in the wall thickness design; rather, only the hoop stress is considered.

The above simplified hoop stress equation has become entrenched in the pipeline design community and is commonly referred to as the Barlow Formula (AWWA M11, 2004). In most design manuals, the variables are manipulated to solve directly for wall thickness (t). Because of its simplicity, inexperienced engineers and designers use this formula to quickly quantify the thickness requirements of the pipeline. What should be understood is that this is where the basic factors of safety and material properties are many times mistakenly assigned. In the AWWA
M11 Manual of Water Supply Practices for Steel Pipe (AWWA M11, 2004), the allowable hoop stress is assumed to be 50% of the material’s minimum yield strength. This would equate to a safety factor against yield of 2.0. As long as the material (steel) does not approach yield the material recovers its original shape and characteristics and therefore the pipeline is deemed safe.

The problem in this formulation is the choice of an appropriate design pressure. Many design firms assume the “worst case condition” which may be a static condition in a gravity pipeline, or a pump shut-off pressure condition in a pumped system, the hydraulic transient pressure condition, or even the specified test pressure of the pipeline. However, because of the cost of the material, many standards use a tolerance greater than 50% of yield to be used for transient and test pressures, which is typically 75% of the material’s minimum yield strength (AWWA M11, 2004). This provides a 1.5 multiplier against exceeding the material yield stress for these infrequent higher pressure conditions. For steel the AWWA standard limits these allowances to 50% and 75% of yield. The American Society of Civil Engineers (ASCE) Steel Penstock Manual (ASCE MOP 79, 2012) allows an emergency load stress of 90% of the tensile strength. It should also be noted that this design is for the steel thickness only and that the composite strength of the lining, coatings and joints components are not considered in this analysis for hoop stress from internal pressure.

It can be argued that the ASCE Penstock design for internal pressure is highly conservative for tensile stresses in a pipeline system. Later in this Chapter, an argument is made that, for normal operating pressures seen in a water system, the design for internal pressure rarely controls and the control for thickness in a steel pipeline is simply ring stiffness for handling. This naturally
provokes a critically important question: why do we have tensile failures? To set the stage, we need to consider external pressure.

2.7.2 Thrust Design

The author presently sits the ASCE Pipeline Division technical committee for Thrust Restraint Design. The committee has published several papers discussing the restraint of thrust along the pipeline as a result of the pipe and soil interface. The author is presently not an active contributor to these papers, but the committee goal is to produce a manual of practice to unify the way in which thrust is restrained within a pipeline system. In this work, the committee has elected to sustain the practice of using a static, unvarying spatial and unvarying temporally, pressure loading to quantify the thrust force on the fitting or transition. In future work on thrust restraint analysis this thesis proposes that the well-known and commonly utilized procedure known as the simplified impulse-momentum theory (Daugherty, Franzini, Finnemore, 1985) be utilized to quantify the thrust force imposed on a fitting or transition. The simplified impulse-momentum theory does not introduce variations spatial or dynamically, but does introduce the idea that there are spatial and hydrodynamic factors that should be considered in extreme cases.

\[
F_x = P1*A1 - P2*A2*\cos \Theta - \rho*Q*(V2*\cos \Theta - V1) \quad \text{EQ 2.7.2}
\]

\[
F_y = P2*A2*\sin \Theta + \rho*Q*V2*\sin \Theta \quad \text{EQ 2.7.3}
\]

\[
F_R = \sqrt{(F_x^2 + F_y^2)} \quad \text{EQ 2.7.4}
\]
In Equations 2.7.2 to 2.7.4 A1 and A2 represent the cross sectional area of the fitting or transition at two points in the system that represent the changed condition, $F_x$, $F_y$ and $F_R$ represent the horizontal, vertical and resultant thrust force, $P_1$ and $P_2$ represent the pressure at the two points defined by A1 and A2, and $\rho$ is the fluid density and $\Theta$ is the angle of the transition. As noted, the equation(s) shown are a simplification of the stream force equation for which momentum and energy flux is considered. The first assumption for simplification is that the fluid is ideal and therefore transmits no tangential stresses. This assumption results in a normal velocity profile and allows for a finite definition of the change in velocity relative to direction through the fitting or transition. The second simplifying assumption is that the flow rate is steady. A steady flow rate means that there is no time dependence or impulse and the resultant force is static. These simplifications allow a coupling with the static longitudinal (axial) and circumferential (hoop) loading along the pipeline. This equation is commonly referred to as the simplified impulse-momentum equation. By using the simplified impulse momentum equation to quantify thrust force a hydrodynamic context is introduced which is the variation in the velocity.

Impulse is a measure of force over time. However, it should be noted that the equations are a simplified form of the impulse-momentum equation and do not represent any dynamic or impulse loading with regard to thrust. It also does not assume any loading in the third axis. In special cases, impulse loading may be critical and special analysis can be made to consider the thrust force in the three spatial dimensions (x,y,z) and time. This can be done by using computational fluid dynamic (CFD) software to define the velocity profile and characterize the fluid momentum flux. This can be loaded into a finite element model (FEM) to better consider
the effect of the distributed velocity profile and effect of the loading rate (impulse). In Chapter 3 of this thesis, it is proposed that a series of analyses can be developed for a range or flow and transient conditions for common fittings and transitions to develop a catalogue that can be referenced by each manual of practice or design standard.

2.7.3 External Pressure

External pressure(s) arise from over-bearing loads such as the weight of the soil on a buried pipeline, the weight of equipment or vehicles above the pipeline, the hydrostatic pore pressure of the groundwater on the pipeline wall, and last but not least the internal pressure below atmospheric pressure (partial or full vacuum pressure) that may be present. Since atmospheric pressure forces are normally ignored, this last external pressure condition actually originates as an internal pressure; in design, it is coupled with the external pressure calculation because of the direction of its forces on the pipeline wall. As was the case for internal pressures, external pressures are traditionally treated as fixed values in design. This means the dynamic loading condition is not routinely factored into the design for internal or external pressure conditions.

In design, the external pressure is the accumulative load applied to the pipeline as a result of four loading conditions: 1) the hydrostatic load, 2) the earth load, 3) the live load, and 4) the internal vacuum load. The external pressures introduce a compressive force on the pipeline and in design this load is assumed uniform and radial. Also, the pipeline is usually assumed empty requiring the pipeline to resist all the external forces with no benefit from the internal pressures. The design approach is to sum up all the characteristics of the over-bearing load and to evaluate the summed equivalent pressure. This loading is then analyzed with respect to allowable deflection of the pipeline, the lining, the coating, and the joint. Pipeline deflection is the key or index
parameter that is usually best understood by design professional when discussing the performance limits in a pipeline design. Many pipeline design specialist assign an allowable deflection relative to the type of pipeline lining and coating and joint systems. That is, for more rigid (relatively inflexible) lining and coating systems (e.g., mortar lining, mortar coating) less deflection is tolerated. The AWWA M11 suggest using a 2% allowable deflection limit for mortar lined and mortar coated pipe, a 3% allowable deflection limit for mortar lined and flexible coated pipe and a 5% allowable deflection limit for flexible lined and flexible coated pipe. In the pipeline design process, deflection is the only parameter that considers the pipeline design as a system rather than as individual components simply because of its consideration in lining, coating and joint selection, but the loading is still taken as static. Figure 2.7.3 shows the external loading considered in pipeline design.

A significant amount of analysis and research work has been performed on flexible pipeline design over the last 50+ years (AWWA Manual M11 (AWWA, 2004), AISI manuals on Welded Steel Water Pipe (AISI, 2007), and ASCE Buried Steel Penstock Manual, ASCE, 2012). During this period, Watkins and Anderson of Utah State University has been a key player in the development and understanding of external loading on the pipeline shell (Watkins and Anderson, 1999) and the influence of soil mechanics in the design of flexible pipeline. Professor Watkins is famous, at least in American pipeline circles, for his acerbic comment “It’s the soil stupid”. And of course, for buried pipeline there is a known and practically understood benefit to having good embedment and soil material around your pipeline to help offset or redirect the over-bearing loads of external pressures. It can be shown that when comparing soil stiffness to ring stiffness that the soil strength accounts for over 95% of the buried pipeline strength.
The structural failure of a pipeline due to external loads may be from either collapse or buckling. There are various equations that are used to estimate the collapse pressure. The AWWA M11 design approach assumes Timoshenko’s formula (Gere, Timoshenko, 1999) to quantify the collapse pressure of the pipeline. In the AWWA approach for collapse, the Timoshenko’s formula is based solely on pipe mechanics, and assumes a uniform, radial load of the external pressure and a perfect and infinitely long cylindrical tube. As a result, this equation does not account for inconsistencies in the pipeline wall or material or shape and therefore may greatly misrepresent the actual collapse pressure of the pipeline. As noted by AWWA M11, there has
been many attempts to develop an empirical formula to estimate the relevance of these inconsistencies/imperfections in the pipeline, but none have been adopted by the standard design guidelines.

It should also be noted that the dynamic behavior of the over-bearing loads (external loads) albeit a vehicle’s live load or a hydraulic transient condition are not considered. The American Association of State of Highway Transportation Officials (ASHTO) and the American Society of Mechanical Engineers (ASME) have developed some impact factors for dynamic live loads (Warman and Hart, 2005), but no impact factors have been considered for hydraulic transient pressures conditions.

2.8 North American Standard Design Practices for Air Valves

The American Water Works Association has a manual of water supply practices that specifically details the pipeline design consideration of air valves in potable water systems. The manual is M51 and is entitled “Air-Release, Air/Vacuum, & Combination Air Valves.” (AWWA, 2001) There is also an AWWA standard for Air-Release, Air/Vacuum, and Combination Air Valves for Waterworks Service. This standard is AWWA standard is C512-07. These two documents are widely used by pipeline design professionals to size and locate air valves. These documents are used in this chapter to establish a comparison to the proposed design procedure (see Chapter 5) for sizing an air valve. At this point it should be noted that both the AWWA manual of practice and standard contain a reference to hydraulic transient (hydrodynamic loading), but this reference is broad and does not provide any detail to appropriately sizing the air valve in regard to hydrodynamic air release.
The following is a list of normal design considerations for air valves as outlined in the AWWA Manual of Practice M51 (AWWA, 2001). For air valve locations, the following list is provided along with the Figure 2.8.1.

**Air Valve Location:**

- High Points
- Mainline Valves (downstream)
- Increased Down Slope
- Decreased Up Slope
- Long Ascents
- Long Descents
- Horizontal Runs
- Venturi Meters (upstream)
- Deep Well and Vertical Turbine Pumps (discharge header)
- Siphons (air release valve with vacuum check)

![Figure 2.8.1: AWWA M51 Air Valve Locations](image)

The recommended locations of the air valves as outlined in AWWA M51 are general and the reasoning for each valve's location as well as its type is somewhat intuitive. In areas along the pipeline that will draw air into the system when the pipeline is dewatered a vacuum valve is recommended. The vacuum valves are recommended for long ascending pipelines (Sta. 11 in
Figure 2.8.1) and where the ascending slope changes abruptly (Stas. 5, 13, and 16 in Figure 2.8.1). Also downstream of isolation or in-line control valves is another location where air may be drawn into the system as the valves are closed and a partial vacuum pressure condition is experienced. At isolation and control valve locations, air vacuum valves are recommended (Sta. 1 in Figure 2.8.1).

Air release valves are used in areas where air may collect and build up in the top or crown of the pipeline. Where it occurs an air release valve is installed to release or “burp” a small volume of air from the system to maintain hydraulic efficiency. Air captured in a system is generally not wanted because it introduces a reduction in the pipeline’s conveyance capacity by restricting the cross sectional area for which fluid is allowed to flow. Also air captured in a system that is under pressure will be compressed and depending on the pressure the compressed air will store a significant amount of energy. Because air density is much less than water, if a large volume of trapped, compressed air is quickly released, an explosive event may result. In this condition, the pipeline has taken on the characteristics of a pressure vessel for which it has not been designed. A pipeline designed as a pressure vessel per following even the lowest standards of the American Society of Mechanical Engineers (ASME) Boiler Pressure Code, would be cost prohibitive. Air release, properly located and sized, is essential in conveyance pipeline designs.

Other locations in a pipeline system require both the egress and ingress of air to the pipeline. These locations are typically equipped with a combination air valve that has both a large vacuum valve port and a smaller air release valve port within the same valve assembly. The location of the combination air valves are shown in Figure 2.8.1 at Stations 2, 6, 8, 14 and 15. The
A combination air valve is a relatively complex valve, which if not properly designed can have an adverse impact to the pipeline system. Because the combination air valve has both the functionality of the air release and the air vacuum valve, it is installed as a ‘catch-all’ valve to handle the normal operating as well as the maintenance and emergency and transient control operational functionality of an air valve. It is common for a designer to size the air valve for the highest required flow rate and/or pressure condition and assume that the valve will function properly for all operating requirements.

The AWWA M51 guideline also lists various assumptions and equations to help size and select orifices for both air release and vacuum valves. The following is a summary of those assumptions and equations (these are expressed in conventional units in M51 and are retained here in this original form). These equations have been presented to provide the understanding that the operating limits of air valves are bounded by the choked condition for air inlet during a partial vacuum condition and by a supersonic condition for air release under pressure. With an understanding of the imposed constraint on air valves as a result of these bounded operating conditions, the importance of the proper sizing and location of air valves becomes more evident.

**Air Release Orifice Size:**

**Assumptions:**
- Standard conditions (60°F and sea level),
- Compressible adiabatic sonic flow
- Air flow rate equal to 2% of fluid design flow \( \frac{Q_{air}}{Q_{fluid}} = 0.02 \)

\[
Q_{air} = \frac{678Yd^2C_d}{\sqrt{\Delta P P_1/TS}}
\]

**EQ. 2.8.1**

Where:
\[ Q_{\text{air}} = \text{air flow rate (scfm)} \]
\[ Y = \text{expansion factor, (AWWA M51 assumes 0.71 for air flow from Crane Technical Paper 410, 1982)} \]
\[ d = \text{orifice diameter (inch)} \]
\[ C_d = \text{coefficient of discharge, (AWWA M51 assumes 0.7)} \]
\[ \Delta P = \text{differential pressure, (for sonic flow = 0.47 P_1)} \]
\[ P_1 = \text{inlet pressure (psia)} \]
\[ T = \text{inlet temperature (Rankine)} \]
\[ S_g = \text{specific gravity, (1.0 for air)} \]

**Air Vacuum Orifice for Controlled Pipeline Filling and Draining:**

Assumptions:

- Standard conditions (60° F and sea level)
- Compressible adiabatic subsonic flow
- \( Q_{\text{air}} = Q_{\text{fillrate}} \)
- \( \Delta P = 2 \text{ psi} \)

\[ Q_{\text{air}} = 14.77d^2 \sqrt[2]{\Delta P(P + 14.7)} \quad \text{EQ. 2.8.2} \]

Where:

\[ P = \text{pipeline pressure (psig)} \]

**Emergency Pipeline Draining:**

Assumptions:

- Standard conditions (60° F and sea level)
- Assume compressible adiabatic subsonic flow
- \( Q_{\text{air}} = Q_{\text{fillrate}} \)
- \( \Delta P = 5 \text{ psi} \)

\[ Q_{\text{air}} = 14.77d^2 \sqrt[2]{\Delta P(P + 14.7)} \quad \text{Same as Eq. 2.8.2} \]
The AWWA approach is based on a static pressure differential at the valve location. This static pressure is only used to develop a volumetric flow rate for sizing the air valve’s orifices. The design process does not account for the dynamic internal pressure experienced by the valve nor does the design procedure account for the change in air volume in relation to the local pressure. Because the approach does not consider the dynamic loading nor the highly compressible nature of air, using this approach as outlined in AWWA M51 will generally result in an oversized air valve (see also Chapter 5 in this thesis). A typical air release valve sizing nomograph using the sizing procedure outlined in AWWA M51 is shown in Figure 2.8.2.

![Figure 2.8.2: Air Release Valve Sizing Nomograph from APCO-Willamette Valve and Primer Corporation](image)
Figure 2.8.2 provides a graphical representation of both the sonic and subsonic flow rates as calculated by the Equations 2.8.1 and 2.8.2 respectively. In the nomograph, if two of the three variables are known and/or assumed then the third can be obtained. For example, if a differential pressure across the valve is 33 psi (227.5 kPa) and the required air release flow rate is 120 scfm (0.057 cms), then, per the AWWA design procedure, a ½-inch (12.7 mm) air release valve would be required. The user of this nomograph should note the log-log axis and the change in slope of the capacity lines at a differential pressure above 13.0 psi (27.7 psia, 191 kPaa). At pressure differentials above 13.0 psi (89.6 kPa), the airflow rate will become sonic through the orifice and the choice of equation to calculate the flow rate is changed from EQ. 2.8.2 to EQ. 2.8.1.

The pipeline designer who is specifying the location, type and size of an air valve generally comes to the design with an available pressure (typically line pressure head in the proximity of the air valve) and an estimate of the egress or ingress flow rate. So using Figure 2.8.2 to design an air release valve located at a high point in a transmission pipeline with a 5 psi (19.7 psi, 135.8 kPa) of available differential pressure head and an estimate of the discharge flow rate of 90 scfm (1.5 cfs, 0.0425 cms) a ¾-inch (19 mm) air release port would be selected.

When considering the size of an orifice of a vacuum valve for air inlet, the over-sizing of the orifice does not cause adverse operating conditions for the system. However, from an economic perspective, a larger vacuum valve will cost more and will only provide marginal added air inlet capacity. The sizing of the vacuum valve orifice and how it should be considered along with the air release sizing are discussed in Chapter 5. Over-sizing both the vacuum and air release valves affects the performance of the air valve with regard to its air release function and its ability to
mitigate adverse hydraulic transient pressures. An over-sized air valve may result in too rapid of
an air release through either the vacuum orifice and/or air release orifice. In a pressurized
pipeline system, a water column drives this rapid air release. When the air is released at a high
velocity, the water column will fill the void in which the air is released. As a result, the water
column will seat the valve float at a high velocity and once the valve float is seated, the flow will
come to rest. Because of the significant difference in density between air and water when the
water column is rapidly brought to rest, an abrupt change in kinetic energy (velocity) to potential
energy (pressure) is experienced. This phenomenon is commonly known as air valve slamming.
Air valve slamming introduces transient pressures to the pipeline system. This type of transient
event, which is referred to here as a secondary transient event, commonly occurs after a primary
transient event where the air valve is called to respond to vacuum pressure conditions. The
AWWA design procedures provide no direction or insight to the sizing of an air valve with
consideration to air valve slamming.

This thesis proposes that the standard sizing procedure that is currently used and outlined in
AWWA M51 be modified to consider the hydrodynamic loading as well as consider the
localized differential pressure and its effect on the compressed air volume. This is developed in
more detail in Chapter 5.

2.9 Other Crucial Design Considerations

In a flexible pipeline design, especially at larger diameters, the stability of the ring stiffness
provides great advantages to the pipeline during its transport and installation. Before the pipeline
is backfilled and the stability of the embedding material can be added to support the pipeline, the
maximum allowable deflection must be carried by the thin steel shell itself. This component of
the thickness design is significant because the required ring stiffness alone may exceed the required thickness of the pipeline for internal or external pressure designs. Also, pipelines with mortar lining and coatings, benefit from the lining and coating’s added rigidity, but the allowable deflection is greatly decreased if cement mortar lining and/or coatings are used. So this can be seen as a “Catch-22” or a double bind. The equation used for the minimum thickness in steel pipeline design for handling has been empirically developed over time and is given a great deal of attention because transportation is generally seen as the riskiest part of the installation and also any damage to the pipeline (wall, lining, coating, etc.) can be easily observed. It should be noted that the AWWA M11 recommendation for this minimum handling thickness was developed by J. Parmakian in 1982 also known for his publication “Waterhammer Analysis” outlining the use of the graphical method in transient analysis.

2.9.1 Transient Pressure and Pipeline Wall Thickness Design

As discussed above, in pipeline system design, transient pressure conditions are traditionally considered as fixed pressures that are included in the analysis of the pipeline with respect to pipeline’s yield strength and collapse/buckling pressure. Currently the transient or dynamic nature of the pressure fluctuation is not accounted for in design. So is the current design approach appropriate? If not, what changes should be made to benefit future design?

There are two evident areas in which the dynamic pressure condition should be considered. The first is at the joint, especially at gasketed (push-on) joint connections. The second is in the lining. The remainder of this chapter discusses the potential use of transient pressure analysis with regard to the joint and lining designs. Special attention is paid to cement mortar lining, which is a
commonly used lining system on both steel and ductile iron pipelines in the potable water industry.

2.9.2 Joint Design

There are various mechanical systems that make up the available joints in a steel pipeline. Because steel can be welded at reasonable temperatures, many designers and owners require a welded joint. The welded joint, if made properly, is excellent from a structural integrity perspective; however, it is also expensive. Not only is the joint costly to manufacturer, to properly weld a pipeline, a significant delay in construction is required and time is money.

The joint with the least initial capital and installation cost is the simple gasketed push on joint. These joint types are typically used in smaller diameter pipelines and at pressure below 250 psig (1700 kPa). However, because of the cost of the welded joint, many owners have asked for better justification for why a push-on joint should not be used. The qualitative answer is that a push-on joint is not viable for high pressures and larger diameter pipeline systems. From the earlier discussion about the design standards and what standard design parameter effects joint design, it is clear that the pipeline deflection is the only design parameter. So, under a given external load, the deflection is limited so that the gasket does not unseat from the joint. The pipeline manufacturers (Northwest Pipe Company, American Cast Iron Pipe Company) have tested their gasketed joint systems under various loads and deflections and have been able to document the joint’s performance. Again these tests have been static test and have been performed under controlled and relative uniform deflection scenarios. In mechanical system a flanged joint with a gasket is commonly used. The potential failure mode of the gasketed joint, albeit a push-on or a flanged joint, is the gaskets extrusion through the joint due to high pressure or the disbonding of
separation of the gasket from the joint due to excessive deflection. Exhibit 1 in Figure 2.9.1 shows an extruded gasket in a flanged joint at the Ohio River raw water intake pump station in Steubenville, Ohio, that had been subjected to repeated high transient pressures in excess of the flange pressure rating. Exhibit 2 shows a failed gasketed joint in Clearview Water Supply Pipeline in Everett, Washington which resulted in significant pipeline erosion and ultimately pipeline failure.

As engineers strive to procure progressively lower cost designs, they are asking questions that were previously avoided because of the perceived risk. However, as technology improves, we as engineers are becoming more inclined to challenge safety factors and the perception of risk. Therefore, designers and engineers are asking, “What are the limitations of the gasketed joint?” In short, designers are saying, “don’t tell me when and where a push-on gasketed joint performs
well, but tell me when and where they will fail”. The author’s contention is the pipeline community does not currently have the test data, nor the experience, to answer this question.

The gasketed joint is another double bind, because the cost savings that are realized from the use of a less expensive joint system many times are used in manufacturing a thicker pipeline shell. The thicker shell is required to maintain the integrity of the pipeline system as a whole by allowing a smaller deflection under the imposed external loads. Because of a dependence and identification of deflection as the key parameter for assessing joint performance, designers seek to reduce deflection to ensure a system’s integrity. The deflection may simply be one mode of failure for a gasketed joint. Another may be the dynamic loading and its effect on the elasticity of the gasket and the ability of the gasket to deform dynamically with the imposed transient load. With this in mind, a stiff gasket, that may be suitable for higher internal pressure, may fail due to its inflexibility when a downsurge or negative pressure transient wave is propagated through the joint. The density/stiffness of the material may prevent the gasket from deforming rapidly enough to prevent the gasket from being carried into the pipeline. And, of course, the counter argument can be made. If the gasket is made pliable enough to handle the downsurge or negative differential pressure change, then the gasket may extrude through the joint. This is a topic remaining to be analyzed and a question left to be answered by the pipeline design professional community.

2.9.3 Lining System Design

Linings have been used for many years either to separate the steel shell from the fluid conveyed or to weaken the corrosive potential of the fluid. The primary purpose of a lining system in a steel pipeline is to reduce the potential for corrosion.
Although there are various kinds of linings used, there are two main types. The first type utilizes Polyurethane. The Polyurethane material is heated and then sprayed on to the pipeline wall like a painting process. The bond of the Polyurethane lining to the pipeline wall is through adhesion. Because of how the Polyurethane lining is installed (generally sprayed on) there is a potential to develop “painter holidays” that expose the steel to the product fluid and thus introducing a corrosion cell. These gaps may also become a point in which the pressurized fluid can seep behind the lining and potentially disbond the lining from the pipeline wall during a change in pressure or transient pressure event.

The second lining type, which is much more common in the raw and potable water industry, is the cement mortar lining. This lining system consists of a dense homogenous cement and sand mixture that is centrifugally cast onto the pipeline wall by the manufacturer. It can also be hand applied in the field during installation. In the centrifugal application process the lining is not adhesively bonded to the pipe wall, so once the cement has dried and set the lining is simply held in place by the radial stresses in the lining. Additional stresses occur in the lining during the transport of the pipeline due to the allowed ring deflection of the pipeline. These stresses due to ring deflection generates small cracks in the lining system. Also, because the cement mortar lining is a porous material, it is proposed that the hydraulically conductive property of the lining allows a differential pressure to set up across the lining thickness during a hydraulic transient event. Currently, transient pressure is not considered in the design of the lining system. Actually, there is little consideration, other than the lining’s influence of ring stiffness, of the lining in the pipeline system design process. Overall, the cement mortar lining is a thin layer of fine grain
cement that is centrifugally cast in the past following the pipe barrel manufacturing. There are typically no adhesives applied to the cement mortar to bond it to the pipe interior wall; therefore the cement mortar is relatively free to move within the pipeline.

In large diameter (greater than 48 inch, 1200 mm) flexible pipeline designs it is common to increase the mortar thickness with the pipeline diameter. The AWWA C205-07 (AWWA, 2007) Standard entitled “Cement-Mortar Protective Lining and Coating for Steel Water Pipe-4 In. (100 mm) and Larger-Shop Applied requires a minimum lining thickness of 0.25” (6 mm) for pipeline 4-10 inches (100-250 mm) up to 0.5” (13 mm) for pipelines over 36” (900 mm). The author is presently designing a 108” (2750 mm) steel pipeline that is specified to have a minimum of 0.75” (19 mm) of cement mortar lining applied. The extra lining is proposed as a sacrificial layer to prolong the leaching of the cement in the presence of the acidic raw water being conveyed. The thickening of the lining is an appropriate measure with the current understanding and lack of developed research for lining failure in the presence of hydraulic transient conditions. In Chapter 4 a hydrodynamic analysis of cement mortar lining is made that shows a potential for a failure mechanism in the presence of a hydrodynamic load.

2.10 Chapter 2 Summary

Chapter 2 identifies multiple areas in the current design guidelines for various appurtenances and components that are common in water and wastewater pipeline designs. With consideration of a more comprehensive roadmap, as outlined by the Integrated Design Approach in Chapter 3 along with the introduction of a temporal parameter defined through the use of hydrodynamic analyses, Chapter 4 and 5 will propose multiple improvements that can be readily introduced to the present design guidelines to improve and refine the risk profile used in design.
Chapter 3 begins to outline an approach to use hydrodynamic loading coupled with an integrated design approach to resolve conflicts that have developed within the devolving North American design standards. The standards have become too specific and the inter-relationships and dependencies on other system components goes unrealized resulting in gross over design of some components and/or under design of other components. The concept of an Integrated Design Approach that requires the designer to identify the performance limitations of each of the pipeline system’s components, and introduces an overarching design parameter, hydrodynamic loading, that can be used to evaluate these performance limitations. The hydrodynamic loading is evaluated not only on the interrelation of the various components of a system, but also on the sensitivity of the system to handle a range of operating conditions. This Hydrodynamic Loading Map can be used to properly select viable materials and operating conditions for a system.

3.1 A Prospective for Pipeline Design

A broad definition of a hydrodynamic pipeline design approach would include all parameters that are changing, varying, or altering in one way or another in reference to a varying hydraulic loading. As an engineer it is easy to be lost in the physical definition of hydrodynamic loading and begin to use it to define extremes in the system rather than applying it to more practical refinements in the standard design of a system within its normal, or common operating conditions.

Pipeline design taken in consideration of this thesis which recommends the identification of a system’s performance limits and allowances is actually quite linear. The various
interdependencies and inter-relationships that are introduced are direct and can be simply applied to the pipeline design guidelines. The goal is to identify these simple steps in an Integrated Design Approach that helps quantify physical inter-relationships of various components in a pipeline system design so that they can be recognized as the pipeline design processes evolves and improves. Taking the Integrated Design Approach further, the dynamic nature of social and political influences may also be enfolded and finally, the ever dynamic definition of risk and value may be considered.

The concepts introduced in the Integrated Design Approach are mentioned here to bring a more inclusive, multi-component approach to the design process. The true intent of this research is not to re-define proven practices but to re-open specific concepts, and reuse these basic principles to enrich a relatively deterministic design philosophy.

3.2 Integrated Design Approach

The hydrodynamic loading consideration is a piece of the design process that has been insufficiently considered in the evolution of manufacturing and design practices for the water and wastewater pipeline industry. What is needed is a way to integrate the current design standards and practices that have been developed in isolation into a more comprehensive and Integrated Design Approach. This requires a conceptual map for framework to inter-relate the relevant factors of each component. This thesis proposes that a more comprehensive consideration of hydrodynamic loading is the key means to develop such a map of the inter-relationships.
The first step in this development is to better define the understanding of hydrodynamic loads within the pipeline industry. In practice, the rupture of a pipeline system is almost inevitably blamed on a hydrodynamic load or “Surge” condition. This term, “surge” has become a catchall for many types of failures and as a result has been demonized as something to avoid or at least minimize the influence of in design, or as a scapegoat for blame. There are indeed hydraulic transient or “surge” failures, but the duration of the transient may be years and may be better characterized by the number of loading cycles instead of the instantaneous magnitude of the loading. The current design standards do not address normal and/or abnormal hydrodynamic loading. The magnitude of the loading is considered as a static force. The duration and frequency of hydrodynamic loading is not explicitly considered, and the probability and/or risk of failure are only considered from a statically determinate perspective via the question, “What is the safety factor of the maximum static load?”

This Integrated Design Approach proposes that if the hydrodynamic loading of the pipeline system is considered in conceptual design then the sensitivity of the system to the loading can be considered so that the potential for failure can be abated and potentially the efficiency of the design may be increased. For the Integrated Design Approach, each component should be interrelated to other components via the hydrodynamic loading characteristics. With this integrated approach a composite model of what is susceptible to failure in the system may be developed.

### 3.3 Quantifying the Hydrodynamic Behavior of a System

In many cases, during a pipeline design scoping study or a system pre-design, the information and time/budget required to perform a comprehensive transient analysis, assess the severity of the transient pressures, and recommend a surge control strategy is not available. When this
occurs, the design engineer will generally default to using various design guidelines or pipeline design manuals where the well-known Joukowsky equation is utilized. (Wylie and Streeter, 1993)

\[
\Delta H = a \frac{\Delta v}{g} \tag{EQ 3.3.1}
\]

In Equation 3.3.1, \(\Delta H\) is the transient pressure head, \(a\) is the celerity or pressure wave speed within the pipeline, \(\Delta v\) is the instantaneous change in velocity, and \(g\) is gravitational acceleration. The Joukowsky equation is often used to quantify the transient pressure potential.

The fault with the Joukowsky equation is not the equation itself, but in its unintended application and use. The Joukowsky equation calculates a change in pressure head resulting from an instantaneous change in velocity of an ideal and incompressible fluid. In pipeline systems, rarely is a transient pressure generated from an instantaneous change in velocity. The results generated by the Joukowsky equation usually leads to disbelief from the design engineer because the estimate of the resultant transient pressure head is typically high. For a system with an instantaneous change in velocity of 2 m/s (6.6 ft/s) a transient pressure head of ~200 m (656 ft) would be anticipated using only the Joukowsky equation.

### 3.3.1 Understanding the Sensitivity of a System to Transient Pressure

The current method used in all the AWWA pipeline design standards to quantify the magnitude of a transient event is the Joukowsky equation. As discussed, this method is simplistic and, as used, grossly over-estimates the transient pressure for all but a very limited number of transient events. Also, the current method does not provide a means to introduce the time dependence of
the transient event nor rate the sensitivity of the system to a transient condition. This thesis proposes that a transient analysis that defines the sensitivity of a pipeline system to a hydraulic transient condition rather than quantifying the transient condition would be of higher value within the standards. Once a designer understands the sensitivity of a system to a transient condition, the designer can make a value based decision on the type and effort required to facilitate a safe design. This decision may result in the development of a full hydraulic transient model or may show that the current Joukowsky equation is most applicable.

3.3.2 Defining the Sensitivity of a System for Transient Pressures

In long pipeline systems, the idea of an instantaneous change in velocity is defined as a change in velocity that occurs any time less than the time for the pressure wave to travel down the pipeline and back (t < 2L/a) (reference Wylie Streeter), where L is the length of the pipeline system. However, in lieu of using the Joukowsky equation when a hydraulic transient occurs within this instantaneous time, this thesis recommends that a full, comprehensive transient model and analysis be performed. A rapid (instantaneous) change in velocity will generate a large number of pressure waves that will require a sophisticated model to properly track and quantify the pressure fluctuations within a system. Also with an instantaneous change in velocity, the formation of a full vacuum and the potential for vapor cavity formation is more likely and a more sophisticated modeling tool is required for design to better estimate the sensitivity of the system to a hydraulic transient as well as define the magnitude of the resultant transient pressure conditions.

However, for the likely case that the time of the transient event is greater than the instantaneous time (t > 2L/a) of the pipeline system, a better tool is needed to rate the sensitivity of a pipeline
system to the transient event. Understanding this sensitivity will allow the designer to employ the
correct resources to perform a proper transient analysis. In an integrated design approach for a
pipeline system, a designer is considering the various design limits of each component. An early
assessment of the transient behavior of a system may allow for early refinements in the type of
material and appurtenances used with the system.

A transient event is defined best by three inter-dependent constituents: 1) the time of change (Δt)
of velocity, 2) the magnitude of the change in velocity (Δv), and 3) the physical and spatial
characteristics (Δx) of the pipeline system. Therefore, an efficient first step in the integrated
design approach is to understand the sensitivity of the system to a transient condition. This
understanding will allow the analyst to better apply the required resources.

In hydraulic modeling, there are three equations that when solved simultaneously represent the
hydraulic characteristics of a system. These three equations are simply conservation of mass
(Qin=Qout), the conservation of momentum (F=ma), and conservation of Energy(Ein=Eout).
One of the main differences in transient hydraulic models when compared to steady state
hydraulic models is the use of the momentum equation. The momentum equation introduces an
interrelationship of hydraulic head and velocity of the fluid to space and time within the system.
This relationship provides a way to record disturbances in the fluid that can be quantified and
tracked as they form and propagate. The momentum equation can be expressed in a partial
differential form:

\[
g \frac{\partial h}{\partial x} + \frac{\partial v}{\partial t} + v \frac{\partial v}{\partial x} + \frac{fv^2}{2D} - g \sin(\Phi) = 0 \quad \text{EQ 3.3.2}
\]
Rearranging and assuming the fluid is incompressible, has a constant density throughout the length of the pipeline, has no viscosity, and as a result has no energy loss to friction, and the pipeline system has a flat slope ($\Phi=0$), the momentum equation can be rewritten as:

$$\frac{g \partial h}{\partial x} + v \frac{\partial v}{\partial x} + \frac{\partial v}{\partial t} = 0 \quad \text{EQ 3.3.3}$$

If Equation 3.3.3 is rearranged to solve for $dh$, two parameters can be isolated on the right hand side of the equation. The first is a partial derivative form of the velocity head $vdv/g$. The velocity head is the most important parameter in quantifying the magnitude of the transient pressure; but it alone does not provide an inter-relationship of the change in velocity, change in time and length. The second is a partial derivative form of acceleration head ($\partial x/g \partial v/\partial t$) which does provide that inter-relationship.

$$\partial h = v \frac{\partial v}{g} + \left( \frac{\partial x}{g} \right) \left( \frac{\partial v}{\partial t} \right) \quad \text{EQ 3.3.4}$$

The acceleration head is dependent not only on the change in velocity and change in time, but is also dependent on the distance along the pipeline ($x$). Assuming: $\partial v = \Delta v$, change in velocity, $\partial x = L$, pipeline length, and $\partial t = t$, the time of change, these values can be applied to the acceleration head parameter in Equation 3.3.4 to yield:

$$\text{Acceleration Head} \approx \left( \frac{L}{g} \right) \left( \frac{\Delta v}{t} \right) \quad \text{EQ 3.3.5}$$
Equation 3.3.5 is not intended to represent a real or a quantifiable solution to the acceleration head but is simply a way to inter-relate the three inter-dependent variables so that the designer can make a measure of hydraulic transient sensitivity of the system. Written this way, the designer can introduce values to the dependent variables and develop a matrix that can be coupled with a decision criterion to assess if the transient behavior of the system warrants consideration of a full, comprehensive transient model.

Introducing a change in velocity of \( \Delta v = 1 \text{ m/s (3.3 ft/s)} \) with a range of pipeline lengths (L) and time of change (t) the following table can be generated that shows the sensitivity of a pipeline system to hydraulic transient conditions using the relationship found using Equation 3.3.5. In Table 3.3.1, the smaller acceleration head indicates a lower sensitivity of the system to hydraulic transient conditions.

**Table 3.3.1: Conceptual Design Decision Matrix using Acceleration Head**

<table>
<thead>
<tr>
<th>Time of Change (( \Delta t )) (s)</th>
<th>Length (dx) (m)</th>
<th>( (L/g) \times (\Delta v/\Delta t) ), Acceleration Head</th>
</tr>
</thead>
<tbody>
<tr>
<td>Instantaneous Time (t&lt;2L/a)</td>
<td>1000 100 10 1 0.1 0.01</td>
<td></td>
</tr>
<tr>
<td>0.002</td>
<td>1</td>
<td>0.0001 0.001 0.01 0.1 1 10</td>
</tr>
<tr>
<td>0.02</td>
<td>10</td>
<td>0.001 0.01 0.1 1 10</td>
</tr>
<tr>
<td>0.2</td>
<td>100</td>
<td>0.01 0.1 1 10</td>
</tr>
<tr>
<td>2</td>
<td>1,000</td>
<td>0.1 1 10</td>
</tr>
<tr>
<td>20</td>
<td>10,000</td>
<td>1 10</td>
</tr>
<tr>
<td>200</td>
<td>100,000</td>
<td>10</td>
</tr>
</tbody>
</table>

Chapter 3 – An Approach to a Hydrodynamic Design Prospective
3.4 Characterization of Hydrodynamic Loading Type

In current design practices, hydrostatic loading is the norm. Even the surge or transient pressure are considered as a maximum or minimum with no duration and therefore by definition are static. These transient pressures are simply higher or lower in magnitude and therefore are used as test pressure to check the structural integrity of the system. In fact in AWWA M11 which is the manual of practice for steel pipeline design, this “static” transient pressure is used to define only two parameters. The first is the thickness of the steel shell. In this calculation the transient pressure is given a higher allowable limit with respect to the yield stress than the normal operating pressure and therefore can be rationalized as a design check rather than an actual design parameter.

The second consideration is within the definition of thrust. The design procedure asks the designer to identify the highest pressure in the system whether it be test pressure or pump shut off head or maximum expected transient pressure. This pressure is then used as a variable in a statically determinant equation that defines the appropriate level of thrust restraint for the system. Currently the author is a member of the ASCE committee for Pipeline Location and Installation and sits on a subcommittee to evaluate the current industry methods used to define pipeline system thrust and thrust restraint. As part of this research, the author sat on a committee that assessed current standard practices for thrust restraint analysis which was presented in multiple papers at the 2009 and 2010 ASCE Pipeline Conference.

Presently, and surprisingly, no pipeline design guideline documents hydrodynamic loading in thrust. The consideration of hydrodynamic loading and its influence on thrust is presented in
multiple texts (Daugherty, R., Franzini, J, Finnemore, E., 1989) and has become known as the ‘impulse-momentum’ theory. This theory, when applied in general design, uses a loose definition of dynamic thrust in the sense that the applied equation assumes a time of loading is equal to 0 and the velocity profile is assumed non-distributed (plug flow), normal to the cross section. The first assumption effectively removes the concept of duration and thus makes the loading static and the second assumption does not identify loading concentrations that may form as a result of a distributed, non-normal velocity profile. This is an example of where the techniques required to evaluate the loading appropriately are available, but the manuals of practice do not utilize this technique nor do they even recognize the existence of a hydrodynamic load.

In fact, the evaluation of hydrodynamic loads can be performed using simple, well defined tools. The distribution of the velocity profile can be fairly well represented by a two dimensional model of the flow field through the use of a computation fluid dynamic (CFD) model which identifies vectorization of the flow velocity profile. Once the flow velocity distribution is developed, a finite element or even a method of characteristic algorithm could be used along with the establish velocity distribution to model various loading rates to develop a resultant hydrodynamic thrust. The cost of this type of modeling may be prohibitive. However, this type of modeling can be performed on a select group of typical fittings and fluid transitions and through a typical range of transient flow conditions that will result in a catalogue of hydrodynamic loading that can be made available in the design guidelines for the designers.

This thesis proposes that the manuals of practice introduce the static, non-distributed normal form of the impulse-momentum theory as a minimum for the calculation of thrust. The thesis
also proposes that the manuals of design practice include catalogs of hydrodynamic loadings so that the designer, end user of the manual of practice, can decide on the level of conservativeness that is appropriate in the design. Currently the AWWA M11 as well as other manuals of practice are void of these considerations. Chapter 2, Section 2.7.2 discusses the use of the impulse momentum theory as a refinement to the static loading that is currently utilized throughout the standards and manuals of practice.

Work has been performed with regard to hydrodynamic loading on various materials used for water and wastewater conveyance (JANA Labs, 2012). These studies have been primarily concerned with the limits of the associated material and the allowances with regard to magnitude of the applied hydrodynamic loads. The manufacturers of PVC and PE pipeline material are generally concerned with frequency and duration of loading due to material fatigue issues. In the review of papers and journals for this work, the author found no work that considered the pipeline barrels, fittings, and joints as a composite system with regard to hydrodynamic loading.

Even though the number and the quality of the studies that have been performed are adequate to draw basic conclusions about the sensitivity of hydrodynamic loads on individual components, the piece of work that is still missing within the pipeline design industry is the road map that inter-relates the research of hydrodynamic loading with design and with operational design considerations. The history (Loading Magnitude (psi or kPa) vs. Time (sec)) of various hydrodynamic loads are shown in Figure 3.4.1A-E. A basic understanding of the magnitude and relative frequency of the hydraulic loading conditions will allow engineers to make better decisions early in the design process on the viability of various materials that the pipeline barrel
and appurtenance can be manufactured from, as well as what type of joints and linings should be considered for the design application. Figure 3.4.1 shows five (A through E) basic loading conditions that represent typical operating hydrodynamic loads within a conveyance system. At the end of this section, these five basic loading conditions have been used to develop a roadmap or decision model (Table 3.4.2) to select viable design alternatives for materials, joints and linings.

Figure 3.4.1A shows a low magnitude transient load with a high frequency of the loading rate. While Figure 3.4.1B shows a high magnitude transient load with a low frequency loading rate. The Figures 3.4.1C and 3.4.1D represent the extremes of high-high magnitude to loading rates and low-low respectively. The loading history shown in Figure 3.4.1E is unique in that it shows the extremes of the high-high loading rate, but it also includes a full vacuum condition. This type of loading may introduce to phase flow by vaporizing the conveyed fluid that is brought to a vacuum condition. These loading histories can be used to develop this road map. These loads can be crudely classified by frequency and magnitude. If this type of loading information were available during the design process, a conscientious pipeline designer should be able to piece together, using the various design guidelines and manufacturing literature, several basic assumptions to help select what pipeline material should be considered. A low frequency and low magnitude loading would allow design options that a high frequency and high magnitude loading may not.

The pipeline system appurtenances, including linings, fittings, and joint systems are not as well prescribed in the existing design guidelines. In addition, the inter-relationships of these elements
Figure 3.4.1: Classification of Hydrodynamic Loading
and their strengths and weaknesses are not well defined, nor is the type of surge control that may be most suitable for the system protection. Many of the components of a pipeline system have elastic and/or poroelastic characteristics that may leave the system susceptible to failure. The key is defining the sensitive loading characteristics of all the design components and making design decisions that are complementary rather than are isolated. The following is a short list of key facts and data that the designer must bear in mind as part of the decision matrix that begins to steer the designer into a more comprehensive design approach.

**Elastic (Flexible) Pipeline Material**

- Higher transient magnitudes acceptable, but limited by number and frequency of loading
- High magnitude / low frequency loading usually acceptable

**Inelastic (Rigid) Pipeline Material**

- Relatively insensitive to high frequency loading, but magnitude of transient loading limited
- Low magnitude / high frequency loading usually acceptable

**Gasketed Joints with Elastic Characteristics**

- Sensitive to both magnitude and frequency of loading
- Low magnitude/low frequency loading usually acceptable

**Cement Mortar Lining**

- Insensitive to frequency of loading, but may be a limit to magnitude of loading
- Low magnitude / high frequency loading usually acceptable

An elastic or flexible system has the ability to deflect and recover with hydrodynamic loading. This ability to deflect is important in the design so that performance limits are not exceeded where the requirement for the elastic material to recover itself to its original shape or form. Examples of elastic or flexible pipeline are Steel, PVC and PE pipelines. In a buried flexible
pipeline design, close consideration of soil strength and soil stiffness are required for structural support of the pipeline (ASCE MOP 119, 2009). An inelastic (rigid) buried pipeline design is much less dependent on the soil support and therefore may be more suitable for installation in soils with low strength and stiffness (Watkins and Anderson, 1999). Again, the inter-relationships of loading on deflection and deflection on material and appropriate bedding conditions are not well evolved in the current design manuals. Table 3.4.2 summarizes the decision attributes:

<table>
<thead>
<tr>
<th>Frequency of Hydrodynamic Loading</th>
<th>Magnitude of Hydrodynamic Loading</th>
<th>Pipeline Material</th>
<th>Lining Material</th>
<th>Joint Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>High</td>
<td>High</td>
<td>PM: Elastic</td>
<td>LM: Bonded</td>
<td>J: Weld</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Steel, Iron</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>PM: Inelastic</td>
<td>LM: Cement</td>
<td>J: Weld,</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Concrete</td>
<td></td>
<td>Special Gasket</td>
</tr>
<tr>
<td>Low</td>
<td>Low</td>
<td>PM: Elastic</td>
<td>LM: Bonded</td>
<td>J: All</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Steel, Iron, PVC/PE</td>
<td></td>
<td>Acceptable</td>
</tr>
<tr>
<td></td>
<td></td>
<td>PM: All Acceptable</td>
<td>LM: All Acceptable</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>J: Weld, Fused</td>
<td></td>
<td>All Acceptable</td>
</tr>
</tbody>
</table>

Note: PM: Pipeline Material, LM: Lining Material, J: Joint Type

Table 3.4.2: Hydrodynamic Loading Decision Model

A pipeline system design using current design guidelines and design practices generally steers the designer away from high magnitude and/or high frequency loading characteristics. From an integrated design perspective, is this direction necessary? A concrete pipeline would be suitable for applications with a high frequency loading characteristic, but that same material would be sensitive to high magnitude with high positive pressure spikes. Whereas, steel would be suitable
for both high frequency and magnitude; however in a steel pipeline design, it is ideal to have pressure fluctuations that remain positive and as a result do not add to the existing external, overbearing load on the pipeline. For a flexible pipeline, a positive pressure will allow the pipeline wall to remain in tension. A large diameter flexible pipeline has a thin shell and may be susceptible to collapse. A concrete pipeline is good in low and/or vacuum pressure conditions because concrete has good compressive strength and is durable in this environment but is less durable when put in tensional loading. Concrete is common in low pressure pumped sewerage conveyance because in these systems the static head is generally low and as a result, the dynamic head (hydrodynamic load) becomes a high percentage of the total head condition in the system. Sewer force mains are also required to come on and off line frequently. So a concrete sewer force main would typically experience high frequency as well as a high magnitude hydrodynamic fluctuation in head every time the pump unit is put on or taken off line. Using only the Hydrodynamic Decision Model shown in Table 3.4.2 would steer the design engineer away from both the concrete pipeline material as well as the bonded lining system. Therefore, this author wants to reiterate the need for an Integrated Design Approach in which the Hydrodynamic Design Model is considered as a step in the process.

3.5 Chapter 3 Provisional Summary

In a pipeline system design, the design is invariably presumed linear, and in many ways it is. Certainly, a failure of one component in the system results in a failure of the entire system. However, the design parameters and the criteria used to formulate the design are anything but linear and because of that the isolated and monotonic approach that currently is used in the design guidelines requires a rethinking. It is proposed that a hydrodynamic inter-relationship of
all the system components be required in the design standards. The inter-relation of the strengths and weaknesses of the system components is required for an Integrated Design Approach.

An Integrated Design Approach (IDA) requires:

- A better definition (empirical and theoretical) of strengths and weaknesses of pipeline segments and pipeline system components with respect to hydrodynamic loading,
- Development of a design approach, a MAP, that requires associative design of all components, and
- Early identification of the sensitivity of the system to hydrodynamic loads and the characteristics of the hydrodynamic loadings within a system.
• With these parameters defined a value based design can be performed for all the components of within the system, and transient control and mitigation designs can be considered to allow or broaden the list of viable design components. This may allow for a more flexible and/or more economic design options.

This IDA helps identify a more economically and socially responsible design by exposing the systems components strengths and weaknesses and their inter-relationships through a parameter (hydrodynamic loading) that has been studied, in detail, in isolation but has not been considered as an overarching decision variable.

3.6 Chapter 3 Conclusion

When utilized, the roadmap developed by identifying sensitivity of the system to transient conditions, the transient loading histories along with the integrated design approach will allow a designer to quickly assess the most appropriate materials and appurtenances for the pipeline system design and if transient control and mitigation may prove cost effective. Ultimately, this integrated design approach will allow for a more efficient design with a greater potential to consider the allowances and constraints of the integrated components resulting in a cost effective, valuable oriented design.

Chapter 4 and 5 of this thesis explore these inter-relationships via control of hydro-dynamic loads and design considerations. Chapter 4 uses hydrodynamic analysis as a basis for assessing the viability and characteristics of pipeline linings. Chapter 5 uses the current air valve design guideline published by the America Water Works Association to propose ways to better include
hydrodynamic loading into air valve design. These discussions will ultimately be proposed to become integral parts of an Integrated Design Approach for pipeline systems.
CHAPTER 4 – MODIFIED DESIGN APPROACHES FOR LININGS WITH CONSIDERATION OF HYDRODYNAMIC LOADING

The hydrodynamic loading consideration is a piece of the design process that has been insufficiently considered in the evolution of manufacturing and design practices for the water and wastewater pipeline industry. This thesis has proposed several ways to use hydrodynamic loading to integrate the current design standards and practices and evaluate the inter-relationships of various components that make up a pipeline system using an Integrated Design Approach. As outlined in Chapters 2 and 3, this integrated design approach acts as a roadmap for an engineer in the design and starts with the evaluation of the hydrodynamic sensitivity of each of the components in the system.

4.1 Introduction

This thesis proposes that a more comprehensive consideration of hydrodynamic loading within the AWWA Standards and ASCE and AWWA manuals of practice, with consideration of both hydraulic loading magnitude and temporal parameters, is the key for development of an integrated design approach. Chapter 4 uses hydrodynamic loading to evaluate the performance limits of cement mortar lining in the presence of a hydraulic transient load. Through the use of the hydrodynamic loading a designer will be able to more readily understand the inter-relationship of the design parameters.

4.2 Hydrodynamic Analysis of Cement Mortar Lining

Introduced in the paper “Sub-atmospheric transient pressure conditions – where and what it may influence in design” (McPherson, Karney, 2008) and subsequently identified as a potential failure mode in a case study outlined in a paper (McReynolds, Peng, Romer, 2010) presented at
the 2010 ASCE Pipelines conference. The case study was an analysis of potential failure modes that caused the delamination of approximately 35% of the cement mortar lining in 5.3 miles (8.5 km) within the 144 inch (3650 mm) diameter Etiwanda pipeline located within the Metropolitan Water District (Los Angeles, California) system Figure 4.2.1.

![Figure 4.2.1: Typical Mortar Lining Failure on Etiwanda Pipeline (McReynolds, Peng, Romer, 2010)](image)

An analysis is developed herein to assess the failure potential of cement mortar lining due to the dynamic loading from a hydraulic transient event. The current design procedure for cement mortar lining only considers the static loading on the pipeline. Even though the extreme static loading procedure may be adequate for the thin steel shell design, the author proposes that the static analysis is not adequate when considering a porous lining material (cement) that has low tensile strength. To properly analyze the dynamic loading on the cement mortar lining, two
elements need to be closely considered. The first is a description of the loading itself and how the
differential pressure is setup across the lining. The second is the analysis of the stress incurred
by the loading and how it compares to the lining’s material strength.

4.2.1 Lining Assumptions

This analysis, as proposed, requires several general assumptions to allow the problem to be
simplified and analytically formulated. It is assumed that the cement mortar lining has no
adhesion to the pipeline wall, and therefore, the mortar and steel are two separate bodies not
acting on one another. When the pipeline is operating normally, the internal pressure will stress
the pipeline wall and slightly expand the steel. The cement mortar lining, which was
centrifugally cast on to the pipeline wall, will also be positively stressed from this expansion.
However, this analysis assumes that the additional stress resulting from the expansion of the steel
is not present. Therefore when the dynamic load is applied, the relaxation of the steel and the
cement lining is not considered. Also the shear stress at the steel/lining interface will not be
realized. When combined with the shear stress in the lining, this unconsidered shear stress at the
pipeline wall may have significant strengthening effect on the lining. Even though the adhesive
property of the centrifugal casting is not high, it is proposed that this parameter along with the
normal operating stress imposed should be evaluated for sensitivity.

A failure is considered as a stress greater than the ultimate tensile strength of the mortar itself. So
the dynamic load (normal to the pipe wall and mortar) would need to overcome the
circumferential stresses and ultimate shear strength of the mortar. Only the hoop stress is
considered; the longitudinal stress and the radial stress are not considered. The cement mortar
lining is initially fully saturated and therefore the initial pore pressure throughout the lining is
assumed to be the same. The cement mortar lining is homogenous with fixed conductivity through its entire cross section. The dynamic load is uniform and radial.

4.2.2 Hydraulic Transient Loading Analysis of Lining

If the cement mortar lining is homogenous and 100% saturated and free of air voids, then a common assumption in soil mechanics is the celerity (sonic wave speed) through the lining would be equal to the fluid sonic wave speed. However, this also assumes that the fluid is incompressible and the boundary layers are rigid. These two later assumptions would result in an instantaneous pressure change across the lining and therefore, the dynamic nature of the loading would not be realized. The author proposes that these later assumptions are invalid with regards to the lining.

Because of the porous nature of the cement mortar lining, many hydraulic analyses, similar to the analysis considered here, can be found in soil mechanics. In soil mechanics there is a classic one dimensional theory describing the dynamic behavior of hydraulic loading to the corresponding change in volume for a completely saturated compressible soil. This theory, introduced by Karl Terzaghi in the early 1920's, is now known as the Terzaghi One Dimensional Theory of Consolidation (Whitlow, 1990). The author proposes that, through a similar approach to Terzaghi’s, a succinct and viable numerical method can be developed to assess the dynamic loading and differential pressure across the cement mortar lining during a hydraulic transient event.

A basic approach using Darcy Law is shown for a differential pressure across the porous lining. This analysis shows that depending on the density, porosity and thickness of the lining material and the hydrodynamic loading rate additional investigation is warranted through
experimentation. For this simplified analysis, there are two assumptions required. The first provides the description of the lining material and the pore space and pore velocity due to the transient loading. The Darcy equation is used to describe this component of the loading rate. The second describes the transient pressure through the lining material. The two assumptions are then coupled to describe the failure potential of the lining due to the hydrodynamic loading.

The pore volume per unit length \( v \) is dependent on the diameter of the pipe \( D \), porosity \( n \) and thickness \( e \) of the lining.

\[
v = \frac{\pi D e n}{L} \quad \text{(L}^3/\text{L)} \quad \text{Eq 4.2.1}
\]

Pore velocity \( v \) is described by the one dimensional Darcy equation with hydraulic conductivity \( K \) of the lining material and the thickness of the lining material \( y \).

\[
v = K \frac{\delta H}{\delta y} \quad \text{(L/s)} \quad \text{Eq 4.2.2}
\]

With the pore volume per unit length and the pore velocity characterized, the effect of the transient pressure can be assessed by setting the resistance to flow that is inherent to the hydraulic conductivity parameter and to the change in velocity for the given pressure differential. To relate these two components at a position along the thickness of the lining \( dy \) the Joukowsky equation is utilized.

\[
\Delta H = \frac{c}{g} \Delta v - v \frac{dy}{K} \quad \text{(L)} \quad \text{Eq 4.2.3}
\]

In Equation 4.2.3 \( v \frac{dy}{K} \) is the resistance and \( c \) is the sonic wave speed through the lining material and \( c \Delta v / g \) is the driving force per unit area of the transient pressure head \( \Delta H \). An inverse relationship of conductivity to transient pressure is shown. A more conductively thinner lining will transfer the transient pressure very quickly while a less conductive, thicker lining would be susceptible to failure.
A model was formulated to analyze the lining characteristics with consideration of the hydrodynamic loading. The boundaries of the model are shown in Figure 4.2.2 below where the exterior or wall side of the lining is assumed to be a reflective boundary and the interior or water side of the lining is assumed a constant pressure head boundary.

\[ H(0, t) = H_{\text{transient}}, \quad t > 0 \]  
Eq 4.2.4

\[ H(y, 0) = H_0, \quad 0 < y < e \]  
Eq 4.2.5

\[ \frac{\partial H}{\partial y}(e, t) = 0 \]  
Eq 4.2.6

Where \( H_0 \) is the initial liquid and pore pressure and prior to the transient pressure and \( H_{\text{transient}} \) is the transient pressure after the transient pressure has passed. The flow and continuity equations can be combined by taking the derivative across the lining thickness \( (y) \) and substituting for \( \frac{dv}{dy} \) for the \( y \) term. Assuming the characteristics of a standard one dimensional diffusion process as derived by DuChateau and Zachmann (DuChateau and Zachmann, 1989) a standard partial
A differential equation is developed for flow (pore velocity) and continuity (balance of pore volume).

\[
\frac{\partial H}{\partial y} = -\frac{v(y)}{K} \quad \text{Eq 4.2.7}
\]

\[
\frac{\partial H}{\partial t} = \frac{\partial^2 H}{\partial y^2} \frac{Kc^2}{g} \quad \text{Eq 4.2.8}
\]

The solution to the differential equations takes the form:

\[
H(y, t) = \frac{1}{\sqrt{\frac{4\pi Kc^2 t}{g}}} e^{-\frac{y^2 g}{Kc^2 t}} \quad \text{Eq 4.2.9}
\]

From this, the timing to balance the pore pressures across the lining thickness is directly proportional with:

\[
t = \frac{y^2 g}{Kc^2} \quad \text{Eq 4.2.10}
\]

Using the solution to the differential and assuming a pipe lining is approximately 19 mm (0.75 in) and conductivity of the lining material is 1x10\(^{-6}\) cm/s and the celerity or sonic wave speed is 1000 m/s, the timing needed to balance the differential pressure is approximately 0.36 seconds. This time suggests that a potential for a significant force across the section of the lining is present and warrants further consideration. Using Equation 4.2.10, Figure 4.2.3 shows the time to equilibrate the transient pressure across the lining. The time increases with increased thickness; therefore, the potential to crack or disband the lining from the pipe wall increases the longer it takes to equilibrate the transient pressure across the lining and as the thickness of the lining increases. The author would like to recognize the correspondence with Ivo Pothof with Deltares / Delft Hydraulics in the development of this model.
When a dynamic load is introduced to the internal wall of the cement lining, the pressure and stress on the top side of the cement lining will for a time maintain the initial pressure (Po) pressure condition while the internal face of the lining will be subjected to a lower transient pressure (P₁). Because of the hydraulic conductivity, assumed similar to the Terzaghi’s One Dimensional Theory of Consolidation of the porous material (Whitlow, 1990), a pressure differential will develop due to the flow resistance across the media. Using a simply one dimensional analysis as described above the potential for failure of the cement lining in the presence of a transient pressure is present.
4.3 Conclusions to Cement Mortar Lining Analysis

Although structural considerations are crucial to the success of any pipeline system, there has been a strong tendency to treat transient loadings superficially and simplistically during pipeline design. The unwarranted simplifications can arise when the range of loadings are being considered, when their duration and transient nature is neglected, or when the behavior of the lining is being considered. One mode of failure that is particularly worrisome and almost wholly neglected is associated with the possible influence of transient loadings on cement mortar linings.

This analysis of cement motor linings when subjected to transient events comes down to how much differential uniform radial load introduced by a negative transient pressure wave can the cement mortar lining resist without failing. The only resistant force holding the lining together is the circumferential compressive load. Due to the history of cement mortar lining and steel pipeline systems as a whole, the author believe that the stresses incurred from the sub-atmospheric dynamic loading of the cement mortar lining will not introduce a failure concern. However, this research may allow for refinements in the thickness and characteristics of the lining material allowed and which may result cost savings and increased reliability of the piping system. The author proposes that the pipeline manufacturers be solicited to perform physical testing to evaluate the potential for lining failure.

When considering the dynamic behavior of an elastic/flexible or rigid/inflexible material, a dynamic loading is required to properly assess the deformation and ultimate strength/strain that the material has. In hydraulic transient analysis, there has been a significant history of empirical analysis and testing of mechanical equipment so that the analyst can better define the dynamic
characteristics of the pipeline system. However, the author proposes that the inclusion of the dynamic behavior of the joints and lining systems and a closer assessment of sub-atmospheric pressure conditions may provide the pipeline design community a more refined and possibly a justifiably less expensive product through an equally robust design process.

Because cement mortar lining is used in both Steel and Ductile Iron pipelines to reduce the corrosion potential of water, if this failure mode proves to be significant, then the need for more proactive or primary hydraulic transient control will be required in conveyance systems not only to protect against catastrophic failures but also to provide a comprehensive corrosion control on the pipeline by allowing the lining systems to maintain its integrity. This may also influence the design professional on the type of hydraulic transient control devices that may be used.
Chapter 4 addressed the use of hydrodynamic loading in analysis of the design performance limit of cement mortar lining. Chapter 5 introduces the integrated design approach for the analysis for sizing and locating air valves within a conveyance system with an undulating profile.

5.1 Introduction

Chapter 5 provides an evaluation procedure using hydrodynamic loading to identify the purpose, size and locate an air valve. This procedure will refine the present standard of practice as identified in AWWA M51.

5.2 Air Valve Design

Air valves in pipeline systems serve two primary functions. The first and most common is the release of unwanted, accumulated air that comes out of solution within the pipeline. This freed air will result in bubble formation, which will coalesce at localized high points along the pipeline profile. This air accumulation will occur when the bubble’s buoyancy is greater than the energy to convey the bubble with the fluid. The air valve used to release this free air is known as an air release valve. The second function of an air valve is to draw air into the system when the pipeline’s internal pressure falls below atmospheric pressures. By drawing air into the pipeline system during as the internal vacuum condition develops, the magnitude of the vacuum pressure can be reduced and as a result help prevent the pipeline from experiencing excessive deflection and/or collapse as well as help prevent the formation of a full vacuum condition in which vapor cavities may form from the fluid vaporizing. This air valve is commonly referred to as an air vacuum valve. An air vacuum valve is also used to release large volumes of air from the
pipeline system when the pipeline is initial filled and after a hydraulic transient event occurs that drew in a significant volume of air. Unless designed otherwise a positive pressure and/or an elevated water level within the valve, body is required to seat the vacuum valve float. During pipeline filling, the vacuum valve will be in an open position. When both an air release and an air vacuum valve are present, the air valve is known as a combination air valve. There are many variations in air valve design including subtle design differences in the valve float arrangement, changes in shape and characteristics of the valve body design, as well as changes in the valves actuation and response. Each of these design characteristics allows the designer a wide range of choices to meet specific operations and design requirements.

A design detail specific to the air valve itself is the orifice sizing. The orifice sizing will directly influence how much and how quickly air can be either released from or drawn into the pipeline system. Another critical design consideration for air valves is their placement in the system. Most air valve manufacturers have sizing tables to size the orifice and qualitative recommendations of where to install air valves in a pipeline system. This chapter will critique the current procedures used for sizing and locating air valves and propose a new process for sizing the air release orifice with consideration of both the compressed air volume, which is directly impacted by its location and hydrodynamic loading.

5.2.1 Analysis of Standard Air Valve Design Approach and Proposed Changes

Air release valves used in pressurized systems are necessary because of the amount of air that is dissolved in water under normal atmospheric pressures and temperature ranges. The standard approach to air valve sizing that has been used in the industry for an extended period of time defines the mass of air in water as 2% by volume. As a result, it has become standard practice to
define the airflow rate (Qair in Equations 2.8.1 and 2.8.2) as 2% of the fluid flow rate. For example, in a system that has a fluid flow rate of 100 cfs, the Qair will be 2.0 cfs or 120 cfm. It should be noted that an air valve designed for this flow rate is designed to release all the air dissolved in the fluid at any one air valve station.

The volume of dissolved air in a saturated fluid has been studied and empirically quantified at various temperatures and pressures. It should be noted that the solubility of air, assumed as an ideal gas, in water is most significantly influenced by changes in pressure and only slightly by temperature. Because the solubility of air in water changes very subtly with temperature, this thesis assumes a standard temperature of 68 F (20 C) for both the fluid and the air. In pipeline systems with undulating profiles, the pressure may fluctuate significantly from high to low and low to high as the water is conveyed through the pipeline. Because the volume of dissolved air in fluid is more influenced by pressure, this thesis assumes a range in pressure of 0.0 to 250 psig, which is a typical operating pressure range for a water conveyance system.

Using the mole fraction solubility of O$_2$, N$_2$ and Ar in water as presented in the 2009 CRC Handbook of Chemistry and Physics, the percent volume of air was calculated over a pressure range of 0.0 to 250 psig. Table 5.2.1 provides the data and assumptions used in this analysis and Figure 5.2.1 shows the result of this analysis at three temperatures 59 F (15 C), 68 F (20 C), and 77 F (25 C). Figure 5.2.1 shows not only the significance of pressure to solubility but also the insensitivity of temperature to solubility.
Table 5.2.1: Data and Assumptions Used in Solubility Analysis

<table>
<thead>
<tr>
<th></th>
<th>H₂O</th>
<th>O₂</th>
<th>N₂</th>
<th>Argon</th>
</tr>
</thead>
<tbody>
<tr>
<td>% of Atmosphere</td>
<td>21%</td>
<td>78%</td>
<td>1%</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temp (F)</th>
<th>Temp (C)</th>
<th>O₂</th>
<th>N₂</th>
<th>Argon</th>
</tr>
</thead>
<tbody>
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<td>1.27386E-05</td>
<td>2.7475E-05</td>
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<tr>
<td>77</td>
<td>25</td>
<td>2.29245E-05</td>
<td>1.1625E-05</td>
<td>2.51836E-05</td>
</tr>
<tr>
<td>86</td>
<td>30</td>
<td>2.12205E-05</td>
<td>1.10797E-05</td>
<td>2.32791E-05</td>
</tr>
<tr>
<td>95</td>
<td>35</td>
<td>1.98218E-05</td>
<td>1.04725E-05</td>
<td>2.16902E-05</td>
</tr>
</tbody>
</table>

Dry Air Ideal Gas Constant 287.05 J/(kg-k)
Water Vapor Ideal Gas Constant 461.495 J/(kg-k)

Figure 5.2.1: Percent Dissolved Air in Water (Fully Saturated Condition)

As reference above, the current AWWA M51 manual of practice for air valves utilizes 2% of the fluid flow rate to size the airflow rate through the air release valve. As can be seen in Figure
5.2.1, the percent dissolved air by volume varies significantly with pressure. However, a 2% dissolved air content in water only occurs at or near an atmospheric pressure or 0.0 psig. What the author wants to highlight and will subsequently focus the remainder of this section on, is the dissolved air volume. In the current design procedure, the general use of 2% to establish an airflow rate will produce oversized air release valves in most pressurized water distribution and conveyance systems in North America where the current procedure is used.

In a typical pressurized water system, air is introduced at a free water surface. Once the water is pressurized, the introduction of air or any gas is very difficult and will require an active air compression and injection. This analysis assumes that all air volume is introduced at atmospheric pressure conditions and as a result, the total volume of air is approximately 2.12% by volume at 68 F, as shown in Figure 5.2.1. As pressure is increased albeit by a pumping unit or gravity flow to a lower elevation in the profile, the mass of air will remain constant in the system; however, the density of air will increase and as a result, the percent air volume in the fluid is reduced. Currently the standard approach of air valve sizing does not account for the reduction in air volume with pressure. The air volume compressed in the air valve body at any position along the pipeline profile will be dependent on the local pressure, saturated air density, and air mass. Assuming the mass of dry air is held constant at 0.025 grams for each liter at 68 F, the percent volume of dissolved air will change inversely with saturated air density. Figure 5.2.2 shows the relationship between percent dissolved air and saturated air density.
The volume of air that is accumulated at any air valve in the profile will be dependent on the localized internal pressure at the location of the air valve. Assuming the mass of air remains constant, an air release valve with an internal pressure of 250 psig (1724 kPa) will be required to expel 0.12% of dissolved air while an air valve with an internal pressure of 0.0 psig (0.0 kPa) the volume is 2.12%. For air valves sighted with an internal pressure of 40 psig (275.8 kPa), 60 psig (413.7 kPa) or 80 psig (551.6 kPa), which are normal operating pressures for water distribution systems, the percent dissolved air is 0.61%, 0.45%, or 0.35% respectively. The recommended air release valve airflow rate of 2% as described in AWWA M51 design procedure would require 3 to 6 times the airflow rate for these three operating pressures. It should be noted that the same mass of air will be released at each one of these air valves and therefore the conservative
assumption that any one air valve will be capable of expelling the full volume of air is maintained. Following in Section 5.2.3 is an analysis using the compressed air volume to size a series of air release valve orifices with a comparison to the AWWA M51 procedure. This comparative analysis assumed a range of fluid flow rate from 0 to 100 cfs (2.83 cms). The results of this sizing are charted in Figure 5.2.3

![Air Release Valve Sizing](image)

**Figure 5.2.3: Air Release Valve Sizing Using Compressed Air Volume**

This thesis proposes that the compressed air volume in saturated dissolved air be used to size air release valves rather than 2% of the design flow rate.
5.2.2 Air Valve Sizing for a Pumped System

The addition of potential energy to a system through a pump will rapidly compress the dissolved air from one volume to another. As a result of the analysis shown in Figure 5.2.2, an air release valve placed on the discharge of the pump assembly can be sized in consideration of the pump pressure head. Therefore, a pump adding 125 psig (861.8 kPa) of pressure to a system will require an air release valve sized for 0.24% of the fluid flow rate. It should be noted that a 0.24% dissolved air release sizing criterion does not consider the air that may be present in the pump column (i.e. vertical turbine pump), but is simply the rational sizing required through consideration of the compressed air volume. When sizing air release valves on wellhead and vertical turbine pump discharge manifolds, it is common to use specially designed air release valves that have been equipped with oil-filled dashpots to retard or slow down the air release. These dashpots are used to prevent air valves from slamming which will result in high transient pressure conditions. A reduction in the air release orifice will help mitigate this air valve slamming and may resolve the need for this dampening mechanism.

5.2.3 Air Valve Sizing and Location for a Pipeline System

In a pipeline system, as the elevation changes, pressure changes. The pressure may increase and decrease significantly depending on the location along the pipeline profile. Figure 5.2.4 shows a hypothetical profile with air valves located through the qualitative procedure outlined in the AWWA M51 manual of practice. Table 5.2.2 provides a summary of the air release valve sizes for both the AWWA M51 design procedure and the proposed compressed air design procedure.
This analysis shows that a more rational design using the compressed air volume to determine the airflow rate ($Q_{air}$) will significantly reduce the size of the air release orifice. This thesis proposes that an air release valve design procedure include the compressed air analysis shown here in and that each air valve design consider the appropriate $Q_{air}$ for its purpose and location.
If a ‘rule of thumb’ is required for the pipeline design industry, the following is proposed in lieu of the current prescribed 2%. For distribution systems and systems with a relatively stable operating pressure range, it is proposed that an airflow rate equivalent to 0.5% of the fluid flow rate is used. For systems with relatively low pressure (0-30 psig) (0-206.8 kPa) and a wide range of operating pressure, an airflow rate equivalent to 1.0% of the fluid flow rate is proposed. This criterion would significantly decrease the orifice size presently outlined in the AWWA M51 manual of practice and by doing so will have benefit to resolving secondary transient events. This topic will be discussed later in this chapter. Figure 5.2.5 compares the proposed air release orifice sizing criterion with the AWWA M51 criterion. Figure 5.2.5 assumes a sonic flow rate through the air release orifice as described in the M51 manual and therefore assumes a minimum pressure differential of 13.0 psi (27.7 psia, 191 kPaa). In addition, Figure 5.2.5 considers the higher pressure that may be present in a distribution system where the proposed 0.5% criterion would be used to size the air release orifice. The orifice sizing resulting from a 40 psi (275.8 kPa) differential pressure was analyzed. This additional differential pressure further reduced the air release valve orifice. A similar analysis was performed for the 1% design criterion. For this case, the 1% design flow rate was used, but only a 5 psi (34.5 kPa) differential pressure was considered. EQ 2.8.2 representing the sub-sonic flow rate was used to calculate the airflow rate. This result is plotted in Figure 5.2.5 as well.

In areas where larger volumes of air are likely to accumulate, like the pump column upstream of the check valve and above the free water surface of the wet well, the proposed 0.5% criterion with its smaller air release orifice will require a longer air release time than a valve designed for 2.0%. Table 5.2.3 is a comparative table showing the calculated time for a system with a fluid
design flow rate of 100 cfs (2.83 cms) with a steady 100 psig (689.5 kPa) discharge pressure at the air valve. For this comparison, the 100 cfs (2.83 cms) and the applied air release sizing criterion will control the orifice sizing for the air release valve. This comparison assumes an initial excess air volume equal to 10 feet (3.05 m) of 8-inch (203 mm) diameter pump column or 3.49 ft³ (0.099 m³).

![Figure 5.2.5: Comparison of Air Release Valve Sizing Flow Rate (Qair) Criterion](image)

<table>
<thead>
<tr>
<th>Orifice Size (inch)</th>
<th>Air Flow Rate (cfm)</th>
<th>Time (sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>M51</td>
<td>5/16</td>
<td>120.00</td>
</tr>
<tr>
<td>Proposed</td>
<td>5/32</td>
<td>30.00</td>
</tr>
</tbody>
</table>

Table 5.2.3: Timing for Air Release for Excess Air
Utilizing the proposed method the additional time required to release the excess air volume is 7.0 seconds or four times longer than the AWWA M51 criterion, but the airflow rate and as a result the water column flow velocity is four times less and the potential for an air valve slam is significantly reduced.

5.2.4 Air Valve Location

The AWWA M51 recommendation for locating air valves for all three operating conditions (Normal operations, Line Fill, and Line Drain) is fairly comprehensive. There has been some recent studies with regard to locating air release valves in consideration of air bubble movement and transport along the pipeline profile, but the general air valve locations as shown in the AWWA M51 document are not directly impacted by this work. Air valve location is not addressed comprehensively in this thesis, but the author supports the most recent research performed by Escarameia in a work entitled “Air Problems in Pipelines, A Design Manual” (Escarameia, 2005), and in that respect the author finds no fatal flaw or need for revision of the qualitative recommendations proposed in the AWWA M51 manual of practice.

5.3 Air Valve Design Analysis Procedure

The following section lays out a design analysis procedure that includes a comparison to the current design practices as well as outlines the steps recommended for the proposed procedure.

5.3.1 Base Case, AWWA M51 Design Procedure

The AWWA M51 manual of practice for air valves was used to locate and size an air valve for the following hypothetical system. The analyzed system is comprised of 10,000 feet (3048 m) of 48-inch (1219.2 mm) diameter steel pipeline with an undulating profile as shown in Figure 5.2.4.
This system has been set up to convey 100 cfs (2.83 cms) of water with a water and air temperature of 68° F (20° C). The water is lifted approximately 302 feet (92 m) from a free water surface level of 0.0 feet (0.0 m) to an elevated reservoir with a free water surface level of 266 feet (81 m). Therefore, there is 36 feet (10.97 m) of pressure head loss in the system. A hydraulic transient event was imposed on this system by reducing the flow by 20% (20 cfs, 0.566 cms) in 3.75 seconds. Figure 5.3.1 shows the steady state hydraulic grade line (HGL) and the hydraulic transient head envelopes resulting from this imposed hydraulic transient event. As shown in the results, the minimum pressure in the system does not reach a full vacuum (-34 feet, -14.7 psig, -101.3 kPa) at the intermediate high point which is located approximately 6000 feet (1828.8 m) from the source. However, the pressure does drop to -29.3 feet (-8.93 m) below the pipeline crown producing a negative internal pressure of -12.7 psig (-87.56 kPa). As a result of this sub-atmospheric pressure condition, this system would most likely be designed with an air valve at this location to resolve this negative pressure issue developed during a hydraulic transient event. It should also be noted that an air valve would also be located at this position for initial filling and potential draining of the pipeline system as well as for normal operating air release.

Using the AWWA M51 design approach the air valve(s) would have a ½ inch (12.7 mm) orifice for air release to pass 2% of the design flow (Qdesign = 100 cfs, 2.83 cms) at 33.5 psig (230.97 kPa) which is 120 cfm (0.0566 cms). This air release orifice sizing can also be obtained by using the nomagraph in Figure 5.4.1. The large air valve orifice would be designed at 3 inch (76.2 mm) for the controlled line filling and draining at 1 ft/s (0.3048 m/s) with an airflow rate of 754 cfm (0.356 cms) at 2 psig (13.79 kPa) differential pressure. However, the large orifice for a gravity flow condition would require a 14-inch (355.6 mm) orifice producing an airflow rate of 21,901.5
cfm (10.34 cms) that is equivalent to a free discharge in a pipeline with a 5% slope and an allowable negative pressure differential of -5.0 psig (-34.5 kPa). Therefore, the design, with consideration of a line break with gravity flow condition, would generate a combination air valve design a ½-inch (12.7 mm) air release and 14-inch (355.6 mm) vacuum orifice.

To test this design with regards to the transient loading, the system was analyzed with a representative transient model. The two primary equations used in predicting transient flow conditions for hydraulic transient events are the continuity and momentum equations. These equations form a pair of quasi-linear hyperbolic partial differential equations in terms of two
dependent variables, velocity and HGL elevation, and two independent variables, distance along the pipeline and time. For numeric modeling, the equations are transformed into four ordinary differential equations by the characteristics method. A model that utilizes the characteristics method was used to analyze this system. The TransAM model was developed by Duncan McInnis, Bryan Karney, and David Axworthy and is commercially available software used in the analysis of hydraulic transient phenomenon. The TransAM model utilizes the method of characteristics to solve the transient equations and the model uses Benjamin Wylie and Victor Streeter’s formulation of air volume expansion and contraction (TransAM Reference Manual, 2004) for analysis of air valves.

In TransAM, the air valve assembly was modeled as a 3-stage air valve where the large orifice would draw and vent air during a transient event until a positive differential pressure was developed to close the orifice. With the large orifice (vacuum orifice) closed the air release orifice would be isolated and continue to release air until the full air volume that was drawn into the system during the transient event is expelled. The modeling of the air valve assembly using this 3-stage characterization more closely represents the function of a standard combination air valve. The TransAM model contains a boundary condition for a 3-stage air valve to facilitate this type of analysis.

The results of the base analysis assuming the AWWA M51 design procedure is shown in Figures 5.3.2 and 5.3.3. Figure 5.3.2 provides a plot of result similar to Figure 5.3.1 in that the maximum and minimum transient head envelopes are shown along with the pipeline profile and
steady state HGL. Figure 5.3.3 plots the model result representing the air volume history and pressure history at the air valve during the transient event.

![Figure 5.3.2: AWWA M51- Steady State HGL and Hydraulic Transient Envelope](image)

In Figure 5.3.2, the sub-atmospheric pressure condition at the air valve location has been resolved. However, through further inspection of the plot, the model results show that both the rising and falling limbs of the pipeline on either side of the air valve location will experience a full vacuum pressure condition. A full internal vacuum pressure condition within the pipeline introduces the potential for vapor cavity formation. The collapse of this vapor cavity will introduce a positive shock pressure. This shock pressure is commonly known as ‘waterhammer’. Therefore, in this system, as modeled, a combination air valve sized with the AWWA M51
procedure will exacerbate the transient pressure problem rather than resolve the issue. In comparison of Figure 5.3.2 and Figure 5.3.3, the reader can see that both the downsurge or low transient pressure and the upsurge or high transient pressure are greater than the system analyzed without the air valve.

![Figure 5.3.3: AWWA M51-Accumulative Air Volume and Pressure Head History at Air Valve Station](image)

As can be seen in Figure 5.3.3, the rate of inflow and outflow through the large orifice of the air valve has a similar volumetric flow rate. The volumetric flow rate through the air valve can be qualitatively considered by inspecting the slope of the volume history. What is also shown is that there is a significant and sudden increase in pressure after all the air is released from the system. Through further inspection of the model result data, it can be shown that this upsurge resulting
from the air valve rapidly closing is the cause of the full vacuum pressure condition shown on the falling and rising limbs of the pipeline. The secondary transient event caused by the air valve closure is the root of the more severe high and low hydraulic transient pressures. The model results show that the minimum pressure occurs in the rising limb feeding the terminal reservoir at $T = 26.0$ seconds or 16 seconds after the initial transient event from the decrease in source flow. Also in Figure 5.3.3, it can be seen that the ½-inch (12.7 mm) air release valve is not providing any benefit during the hydraulic transient event because all the air is being vented through the large vacuum orifice. If the smaller orifice were being utilized the slope of the accumulated air volume history would change abruptly. The TransAM model provides the user with a notification when the large orifice is seated and the small orifice is activated. This notification was not shown and therefore, as modeled, the air release valve remained inactive.

The remainder of this analysis section provides various air valve sizing scenarios set up to resolve the secondary transient event and better protect the pipeline system than the “do nothing” or “provide no air valve” scenario.

5.3.2 Modified Design Scenario 1, AWWA M51 Controlled Draining Procedure

The Base Case considered the sizing of the vacuum valve for gravity flow and/or line break rather than for controlled draining. Scenario 1 considers the reduction in the vacuum valve size to the size required for a controlled drain down. This sizing, based on the AWWA M51 design procedure is 3 inch (76.2 mm). Figures 5.3.4 and 5.3.5 show the profile and head envelope and air volume and pressure history for this scenario, respectively.
Reducing the vacuum valve orifice alone does not provide appreciably better results. A full vacuum pressure on the rising limb to the terminal reservoir is still shown in the model results. The minimum transient pressure is still along this rising limb and occurs at approximately the same time or $T = 26.1$ seconds or 16.1 seconds after the transient event is initiated. In addition, it should be noted that in this scenario the pipeline system is no longer protected from a line break or gravity flow condition.
Figure 5.3.5: Scenario 1-Accumulative Air Volume and Pressure Head History at Air Valve Station

The shape of the inflow and outflow curve shown in Figure 5.3.5 is similar to Figure 5.3.3; however, both the air vacuum large orifice and the air release orifice are active in this scenario. The results are not significantly different because the majority of the air volume drawn into the system is expelled through both the large and small orifices at a relatively unrestricted, sub-sonic flow rate before the vacuum valve closes and the air release valve acts alone. The remaining air volume passes through the 1/2-inch (12.7 mm) air release valve very rapidly again causing a secondary transient event.
5.3.3 Modified Design Scenario 2, Air Flow Rate for Air Release of 0.5% of Fluid Flow Rate

The next scenario considers the reduction of the air release valve size. The size criterion used in this scenario was the ‘rule of thumb’ criterion established for systems with stable pressure conditions; therefore, the air release valve was sized for an airflow rate of 0.5% of the fluid flow rate. This air release valve orifice size will produce a sonic flow rate during the air release. A sonic flow rate will be achieved through the air release valve when the air is being expelled at pressures in excess of the critical pressure, which is 13.0 psig (89.63 kPa) or 27.7 psia (190.98 kPaa) (Fluid Meters, Their Theory and Application, ASME 1971). For this scenario, the vacuum orifice or large orifice is kept at 3-inch (76.2 mm) but the air release or small orifice is reduced to ¼ inch (6.35 mm). Figures 5.3.6 and 5.3.7 show the model results for this scenario including the profile and head envelope and air volume and pressure history. As can be seen in Figure 5.3.6, the full vacuum pressure previously shown at or near the air valve and along the rising limb to the terminal reservoir has been eliminated. This is due to the more restricted air release generated by the smaller air release orifice. The ¼-inch (6.35 mm) air release orifice, which is sized for approximately 0.5% of the design flow rate rather than 2% of the design flow rate provides a dampening effect on the release of air from the system thus resolving the occurrence of the secondary transient event that had previously resulted in the Base Case and Scenario 1 from than air valve closure.

However, by using a ¼-inch (6.35 mm) air release design, the air that is drawn into the system by the 3-inch (76.2 mm) vacuum valve is not fully expelled during the first open and closing cycle of the air valve. Figure 5.3.7 clearly shows that the volume of air drawn into the system is never fully expelled. This remaining air volume does not cause problems during the transient event, but
the system will require additional time to expel all the air before full capacity is again realized. In that respect,

Figure 5.3.6: Scenario 2-Steady State HGL and Hydraulic Transient Envelope
the sizing the air release valve for 0.5% of fluid flow rate would introduce a need for an operating plan to re-startup the system slowly after the transient event. In this scenario, it should be noted that the lowest pressure is a result of the initial pressure transient condition and is no longer due to the air valve closure and secondary transient event. In that respect, this air valve design provides good protection to the system for a hydraulic transient event.

### 5.3.4 Modified Design Scenario 3, Compressed Air Volume for Air Release Valve Sizing

The compressed dissolved air volume analysis outlined in Section 5.3 showed that for a system with an air valve with a normal operating pressure of 35 psig (241.3 kPa) the compressed air
volume would be 0.67%. Using this design procedure, an air release valve designed for 0.67% of the fluid flow rate would be a rational criterion. Therefore, at a 100 cfs (2.83 cms) design flow rate, an airflow rate (Qair) would be equivalent to 40.2 cfm (0.0189 cms). At 40.2 cfm (0.0189 cms) and a differential pressure of 35 psi (242.3 kPa), the air release orifice size is 9/32 inch (7.14 mm). Therefore, in this scenario, the vacuum orifice is maintained at 3-inch (76.2 mm) and the air release is sized at 9/32-inch (7.14 mm). This air vacuum valve sizing will allow a subsonic inflow rate while the air release flow rate will be sonic. Figures 5.3.8 and 5.3.9 again show the profile and head envelope and air volume and pressure history for the reduced air valve respectively. The pressure envelope in Figure 5.3.8 shows that the low pressure everywhere along the profile is maintained at atmospheric pressure (0 psig, 0 kPa) or higher and the lowest pressure occurs at the valve location approximately 6000 feet (1828.8 m) from the source. Again, in this scenario, it should be noted that the lowest pressure is a result of the initial pressure transient condition and is no longer due to the air valve closure and secondary transient event. In that respect, this air valve design provides good protection to the system for a hydraulic transient.
Similar to the ¼-inch (6.35 mm) air release design in Scenario 2, an air release valve design with an orifice of 9/32 inch (7.14 mm) will not fully expel the air in the first opening and closing cycle that is drawn into the system by the 3-inch (76.2 mm) vacuum valve. Figure 5.3.9 shows that the volume of air drawn into the system is never fully expelled. This remaining air volume does not cause problems during the transient event, but the system will require additional time to expel all the air before full capacity is regained. In that respect, sizing the air release for actual compressed air volume would introduce an additional design and operational issue for re-startup.
of the system, but the adverse conditions introduce by the transient event have been resolved.

Figure 5.3.9: Scenario 3-Accumulative Air Volume and Pressure Head History at Air Valve Station

Figure 5.3.10 shows the modeled accumulative air volume of Scenario 3, as shown in Figure 5.3.9 but with a longer duration. This figure shows that in 60 seconds (50 seconds after the transient event is initiated), the air volume drawn into the system will be fully expel, but it
should be noted that this is after multiple openings and closings of the air valve assembly.

Figure 5.3.10: Scenario 3-Accumulative Air Volume over 60 seconds

The last scenario will provide a composite design that considers the normal operations, the transient control, and the subsequent air release required to allow the system to restart (come back on line) quickly after a transient event is experienced.

5.3.5 Modified Design Scenario 4, Composite Design Procedure

In the final scenario, the air release orifice was sized to maintain the restricted, sonic airflow rate through the air release valve but was also sized to release all the air that is drawn into the system during a hydraulic transient event in the first cycle of the air valve opening and closing. It is important to sustain the dampening affect introduced by the restricted air release so the adverse
impact of the secondary transient event caused by the air valve closure can be abated. The air release valve orifice was sized to 3/8-inch (9.52 mm), which introduced a restricted flow rate through the valve greater than Scenario 1 where the orifice was ½-inch (12.7 mm), but not as great as Scenarios 2 and 3. Figures 5.3.11 and 5.3.12 again show the profile and head envelope and air volume and pressure history for the air valve respectively. The pressure envelope in Figure 5.3.11 shows that the pressure everywhere along the profile is maintained at atmospheric pressure (0 psig, 0 kPa) or higher and the lowest pressure occurs at the valve location approximately 6000 feet (1828.8 m) from the source.

Figure 5.3.11: Scenario 3-Steady State HGL and Hydraulic Transient Envelope
Again, it should be noted that the low pressure is a result of the initial pressure transient condition and is no longer due to the subsequent air valve closure and secondary transient event. As shown in Figure 5.3.12, all the air drawn into the system is expelled. This air valve design provides good protection to the system for a hydraulic transient event and provides no additional time to expel air. The air release valve design for this scenario is equivalent to 1.2% of the fluid design flow rate. The AWWA M51 recommendation is 2% of the fluid flow rate.

Figure 5.3.12: Scenario 3-Accumulative Air Volume and Pressure Head History at Air Valve Station

5.4 Air Valve Design Analysis Summary

From the analyses presented above, it is proposed that the combination valve’s air vacuum valve orifice be sized utilizing the AWWA M51 procedure for controlled pipeline drain down and/or line filling, which for this example, resulted in a 3-inch (76.2 mm) orifice sizing. This vacuum
valve sizing procedure is simply a controlled drain down rate, which maintains a pipeline fluid
flow velocity of 1 ft/s (0.3048 m/s). In the AWWA M51 procedure for controlled drain down,
the air vacuum valve is sized to flow at a sub-sonic rate with a differential pressure of 2 psi
(13.79 kPa). The benefit of this sizing is that a vacuum valve sized using the controlled drain
down procedure will be more likely to close during a hydraulic transient event because the
vacuum valve is smaller and lighter than the valve sized using the line break or gravity flow
sizing criterion. As a result, the air vacuum valve float will be smaller and therefore more easily
displaced when positive pressure is experienced in the pipeline at the valve. Once the large
vacuum orifice is closed, the air release valve will be allowed to operate as designed and reduce
the potential for secondary hydraulic transient events.

This thesis proposes that the current design practice for sizing air release valves be modified. It is
proposed that the most defensible and rational approach to sizing air release valves would be to
size the air release orifice with consideration of the compressed air volume. It should also be
noted that the 2% of design flow rate criterion that is currently recommended by AWWA M51 is
a relic of a paper written by Lescovich in the July 1972 AWWA Journal. The 2% represents the
volume of dissolved air in water in a fully saturated condition. This measure of air volume is
relatively inaccurate when one considers the range of pressure in a pipeline system. The author
would like to note that the volume of free air in a pipeline system under higher pressure is small
and the release of air in high pressure pipelines is not significant. The compressed free air is
generally conveyed to a point within the pipeline system where operating pressures are much
lower. At this lower pressure and expanded air volume is where air will most be released from
the system. The author would also like to take note that many distribution systems are equipped
with air release valves throughout. Because these air release valves generally do not see low operating pressures, many times these valves are persistently inactive which may result in the air valve float and seal developing a calcium deposit over time. Calcification, if allowed to build up on the valve ports, will likely prevent the valve from operating properly and potentially exacerbate the problem (a potential point of leakage or cross-contamination) rather than an aid in the operation of the pipeline. It is recommended that AWWA M51 make note of this common problem and recommend maintenance strategies.

With regard to the gravity flow design of the vacuum valve, this thesis proposes that a larger vacuum valve be designed as a second air valve. This second air valve assembly, separate from the combination valve design, should be set to open only under a severe vacuum condition. The severe vacuum condition can be defined with a safety factor in relation to the theoretical collapse pressure of the pipeline system. The definition of collapse pressure is well documented throughout the pipeline industry. It is important that this larger vacuum valve be designed so that the pipeline’s allowable negative pressure limits are not exceeded. For this larger vacuum valve, the orifice can be spring loaded to prevent opening under minor partial vacuum pressure where only the combination valve is required.

5.5 Conclusions of Air Valve Design with Consideration of Hydrodynamic Loading

The analysis of dissolved air and the subsequent series of modeled scenarios provide an important critique of the standard design practices in North America with regard to air valve sizing. The analysis shows that air valve sizing utilizing the AWWA M51 practices may be inadequate and may introduce or exacerbate hydraulic transient conditions. By using a
compressed air analysis along with a dynamic hydraulic model a better understanding of the
dynamic loading characteristics present over the life of an air valve may be considered in design.

Special consideration will be required for air release valves sighted on pump discharge headers
and at locations where excess air is introduced to the system. Currently, the AWWA M51 design
procedure recommends to upsize both the air release and air vacuum orifices if the orifices are
shown to have a capacity that is marginal in relation to the sizing criterion. This thesis proposes
that if the air release valve capacity is marginal then the air release valve be down sized instead
of upsized.

Along with the changes to the design practices discussed above, this thesis also proposes changes
to the monitoring and maintenance requirements for air valves. Currently supervisory control and
data acquisition (SCADA) systems are used throughout the water conveyance industry to help
identify and improve upon performance goals. However, the author through fifteen years of
design experience and exhaustive research has yet to come across a comprehensive SCADA
system that monitors air valve performance. With a comprehensive monitoring system in which
air valve open/close positions are recorded as well as differential pressures, a better
understanding of both the air valve sizing and location can be made. For example utilizing the
current design practices outlined in the AWWA M51 manual of practice a 30 mile (48.3 kM)
long transmission pipeline may have more than 100 air valve assembles. Many of these air valve
assemblies will be used once (during line filling) in the life span of the pipeline and will never be
required again. However, some of these facilities may be used every time a pump starts or stops
or the flow rate is changed in the system. Without thorough monitoring, the location of these two
extreme cases may never be known over the life of the pipeline and as a result, the maintenance of one extreme will be given similar consideration as the other. This thesis proposes that modified design practices and guidelines (e.g. AWWA M51) include a recommendation for SCADA systems that monitor air valves stations. Then with this recorded data, a more systemic maintenance program can be developed as well as a more comprehensive design procedure that considers the true operational requirements of each valve and valve location. This procedure can be used as a feedback loop to assess where additional maintenance may be required or where there may be a potential for an alternate design. For example, at air valve stations that are only required during startup and shutdown the air valves may be replace with secured pit cocks. This will reduce the up from capital cost as well as provide a more solid boundary against cross contamination, and reduce the maintenance requirement as well. The valves that are found to be frequently used can be assessed for performance in relation to sizing and location as well as be assigned to a more rigorous maintenance program. Conceptually, it is easy to assume that this type of monitoring and feedback to re-design may pay for itself within the first few years of operation and if considering the reduction in potential cross contamination the pay back may be immediate. In any regards, as can be seen in the above analyses, air valve sizing is not wholly conducive to prescribed tables or nomographs. These proposed changes to the standard design practices also require a unique study of the air valve location and sizing requirements. Utilizing transient models that allow a way to consider the hydrodynamic loading characteristics is critical to a proper design. Also, the assessment of the designed system through monitoring and feedback with consideration for re-design should also be noted on the critical path as well. Ultimately, this thesis proposes that the AWWA M51 “Air-Release, Air/Vacuum, & Combination Air Valves” be re-written with this chapter as a foundation.
This thesis also proposes that a standard data request developed by design professionals be introduced to the manual of practice. The data that is not readily available through published manufacturer’s data but is required for a basic design is the pressure requirement needed to seat the float of the large vacuum orifice. This data can be calculated if the mass and effective surface area of the float is known, but that information is not widely published and as a result, many of the installed vacuum valves also provide full air release during hydraulic transient events. As shown in the modeling above, this rapid expulsion through the air vacuum valve may cause a significant secondary transient event from air valve slam that may result in more severe hydraulic transient pressures within the pipeline system.

5.5.1 Air Valve Design

Several important observations should be highlighted from the analysis proposed in this TM. An air valve is a reactionary device and has several components that need to be carefully considered both individually and collectively. The first consideration is the primary function of the valve. This TM proposes that the most flexible air valve design would include two air valve assemblies. One valve assembly contains a combination air valve with a vacuum valve and air release valve sized for normal operation and hydraulic transient control. The second air valve assembly will be a large ported air valve sized for a line break scenario.

The design and locating of air valves has become deterministic in modern design. Simple tables and nomographs have replaced analysis for locating and sizing air valves within a system. Even though this paper has proposed multiple changes in how air valves are considered in design; there is very little empirical evidence of the benefit of these proposed changes. A fundamental
and relative obvious argument can be made outlining the cost savings of the reduced air valves, but in a large system, the collective cost savings for each valve may not meet the savings provided by a manufacturer for specifying a common valve size and type. Another issue to consider is that air valves are obvious points of cross contamination and are known to be remote and un-maintained appurtenances in most water conveyance systems. Given this basic argument, it is critical that air valves be considered with more care and diligence in future design.
CHAPTER 6 – CONCLUSIONS AND PROSPECTIVE WORK

Throughout the world, societies have become dependent, although in large part unaware of the dependency, on inexpensive, reliable, and safe conveyance systems that deliver raw and potable drinking water as well as that collect wastewater. The dialectic nature of the aging conveyance infrastructure to remain inexpensive while sustaining the safety and reliability of service requires a fundamental change in how we consider and assess the design and use of piping systems.

This thesis makes two basic research contributions to the field of pipeline and conveyance system design. The first is outlined in the Integrated Design Approach (IDA) where hydrodynamic loading is used as a means to inter-relate and assess the design, in many ways conflicting design, approaches of the various components necessary within a pipeline system. The IDA also allows for more comprehensive planning for when, where and what type of components and/or materials should be considered in relation to the inter-dependencies of the components as defined by the magnitude and frequency associated with hydrodynamic loading.

The second contribution is the development of an analysis procedure to quantify the performance limits and sensitivity of cement mortar lining and air valves in the presence of a hydrodynamic load. The analyses that are developed in this work are directly applicable to the improvement of the present design practice and can be integrated into the standards and manuals of practice.

The present North American pipeline and appurtenance design standards were thoroughly reviewed and areas where a temporal parameter, resulting from the consideration of hydrodynamic loading, were identified. These areas included early design decisions such as the type of pipeline material, the type of joint, gasket/seal material, and type of lining material. Each
of these component’s design and performance limits is identified in their respective design
guidelines and controlling standard, but a means to inter-relate the interdependences of the
various performance limits is not. This work provides that roadmap that inter-related and/or
identified the interdependences of the various components that can be introduced to the standard
of practice. Through the classification that is characterized by the magnitude and frequency of a
hydrodynamic load, these interdependent relationships define that needed roadmap. This
concept can easily be introduced into the design guidelines to refine the material, components
considered for design as well as introduce a quantifiable temporal consideration for the future
analysis of fatigue, and probability of failure.

6.1 Hydrodynamics Analysis Applied in Design

The second significant contribution of this work is the use of hydrodynamic loading to formulate
a model to assess the performance limits of cement mortar lining and the development of an
analysis procedure to refine the proper selection of air valves in a transmission pipeline system.

1. It is common in the design and specification of pipe to consider an additional thickness of
cement mortar lining when a corrosive (acidic) raw or treated water is conveyed. What is
not considered, in the standards and design guidelines, is the presence of a hydrodynamic
force and the influence of the lining thickness on the ability to resolve the impulse of that
force. A simple model was developed using Darcy’s law in the presence of a
hydrodynamic load. The model found that the duration of the loading introduced a
potential for failure of the lining. This simple model resulted in a once overlooked
interdependency of cement mortar lining thickness and hydrodynamic loads. This alone is
a significant contribution to the industry, but put in context of this work, shows the need
and value of considering a hydrodynamic load to design a conveyance system.
2. The present design procedure for air valves does not adequately consider the range of hydrodynamic loading in which an air valve may encounter over its useful life. Many times the air valve is designed for a principal purpose, irrespective of the range of pressures it may be called to operate within. This work outlines a procedure using hydraulic transient modeling software to help select the type and primary and secondary purpose of the air valve and select the size and the location of the valve. In the analysis presented in Chapter 7, it should that the proper sizing and control of both the air vacuum port and the air release port resulted in the need for one (1) air valve, where if the AWWA M51 procedure were followed, five (5) valves would have been required. This analysis procedure is a significant contribution and again put in context of this work, shows the need for the consideration of hydrodynamic loading in design.

6.2 Prospective Work

The intent of this work is to evolve the standards of practice and design guidelines that are presently used in the industry. These standards and guidelines have developed over the years and have repeatedly shown to produce high quality products. Due to the increasing cost of operating, maintaining and/or replacing the conveyance infrastructure throughout the world, the need to refine and identify areas where excessive, and therefore expensive, factors of safety have been utilized needs to occur. Of course in this pursuit for design efficiency, the level of quality should not be compromised because the comfort of “knowing” that water will flow when the faucet is turned on is immeasurable. Through the use of hydrodynamic analysis, the author believes that this design efficiency can be achieved and quantifiable, simply models can be developed to identify and improve on the standard of design. Three steps are required.
1. Using the hydrodynamic loading, the interdependencies of each component in a composite design must be evaluated and identified so that models can be formulated. It should be noted that the research, analysis, and development of the models must break from the practice of sizing and refinement of individual components to the identification of composite system interdependencies. At this stage the interdependencies, not the models, are introduced to the design guidelines. This will demonstrate the value and stimulate the research for the further development of the concept.

2. The new models must then be tested and vetted to identify limitations of practical use. The simple models such as the cement mortar-lining model, once vetted, can be introduced into the design guidelines. The more complex numerical models and/or physical models can be run with multiple design parameters, these design parameters are defined by the context of the model, to develop nomographs or reference catalogs, lookup tables, or databases. These are published.

3. The final step is to develop a comprehensive design procedure to outline the use and application of the hydrodynamic design approach to allow design of each component and allow specification in design.

The author is presently contributing to the re-drafting of multiple standards and manuals of practice for the American Water Works Association and American Society of Civil Engineers. In this work on the standards and manuals, the author has concluded that the practicing arm of conveyance design is failing to evolve as quickly as the numerical methods and analysis techniques developed within the universities. This disconnect along with the understanding that it is in human nature to treat the unknown or unfamiliar with skepticism and/or suspicion, leads
the author to believe the only path forward for this work is the three (3) steps. On to step one I go.
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