

Airbag for piping systems

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Master of Science in Product Design and ManufacturingSubmission date:December 2010Supervisor:Morten Kjeldsen, EPTCo-supervisor:Håkon Hjort Francke, Flow Design Bureau AS

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Problem Description

Background and objective.

Transients in piping systems occurs whenever a change of operation e.g. in valve stroke or pump/ turbine speed takes place. For very fast changes of operation the pressure amplitude of the transient, or the reflected wave, can have deleterious effects on the mechanical integrity of the system.

The energy contained by the transient, and hence the pressure amplitude, can be influenced by scattering and attenuation. Scattering can be achieved by letting said transient propagate between zones of different impedances, i.e. the product of density and speed of sound. Attenuation is achieved by introducing loss mechanisms in the region of the pipe where the transient propagates. Both scattering and attenuation can be achieved by introducing gas into a liquid pipe flow. Gas bubbles in liquids will have a huge impact on the speed of sound even for very small Gas Volume Fractions. Attenuation is achieved since the compression-expansion cycles on the gas bubbles will essentially be fast and isothermal; and hence not isentropic or loss-free.

It s evident that a transient in an original liquid pipe flow can be dramatically affected if (small amounts of) gas is added the pipe flow. The objective of this study is therefore to verify or justify that an air injection scheme for controlling transients in piping systems can be established. The following tasks should be considered in the project work:

1 Establish an analytical model for the calculation of the effect of adding air into a pipe flow where transients occur. This should be accomplished either by developing an in-house MOC transient flow solver using Matlab, LabView or similar software, or by using commercially available software such as Flowmaster.

2 Provide a design study for gas injection. The study should have an emphasize on how (potentially) favorable bubble size distributions can be achieved.

3 Provide a design study for a sensor- analysis/processing- air injection actuation set-up, the airbag, that can affect hazardeous transients of a certain amplitude and/or nature.

4 Provide a design study of an experimental set up (prototype) where both item 2 and 3 above are demonstrated.

All tasks should include a literature survey with a special emphasis on state of the art within the scientific community.

Assignment given: 23. August 2010 Supervisor: Morten Kjeldsen, EPT

Norwegian University of Science and Technology NTNU

Department of Energy and Process Engineering



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"Airbag for piping systems"

Airbag for rørsystemer

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The candidate is free to choose whether to deliver the thesis report in English or Norwegian.

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Within 14 days of receiving the written text on the diploma thesis, the candidate shall submit a research plan for his project to the department.

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Two -2 – copies of the thesis shall be submitted to the Department. Upon request, additional copies shall be submitted directly to research advisors/companies. A CD-ROM (Word format or corresponding) containing the thesis, and including the short summary, must also be submitted to the Department of Energy and Process Engineering

Department of Energy and Process Engineering, August 2010

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Abstract

Pressure transients are caused by a change in the volumetric flow in a pipeline system, and can have severe consequences for rapid changes of the volumetric flow. A sudden closure of a valve is a common source of a pressure transient, and the pressure increase in front of the valve depends on the flow rate and the wave propagation velocity in the fluid and pipe. A gas-liquid mixture can have a very low wave propagation velocity, even for small air contents, and is effective in terms of damping due to the compressibility effects of the gas bubbles. With these alterations of the fluid properties the pressure transient will be weakened with reduced amplitude and an increased period, which are beneficial effects for the pipeline system.

A simple experiment was carried out to investigate the practical solutions for the air injection system, and the results showed that the presence of air was beneficial in terms of a reduction of the amplitude and increased damping of the pressure transient. However, a few aspects should be revised in a refinement of the experiment. The air flow rate and duration were uncertain because of water accumulating in the air hose, and the timing of the gate valve closure was challenging.

Simulations of various models of pipe systems were carried out in Flowmaster. The models are sufficient for simulation of ordinary pipes with a rapid closure of a valve, but fall short at modeling an air-water mixture. This is because only the reduced wave propagation is taken into account, and not the effects of the bubbles.

Sammendrag

Dynamiske trykkendringer forårsakes av at volumstrømmen i rørsystemet endres, og kan ha alvorlige konsekvenser for raske endringer av volumstrømmen. En ventil som lukkes raskt er et vanlig opphav til slike trykkendringer. Trykkøkningen foran ventilen er avhengig av volumstrømmen og lydhastigheten i rør og fluid. En blanding av gass og væske kan inneha svært lave lydhastigheter, selv for små luftmengder. Dessuten er blandingen effektiv med hensyn til demping av trykkendringen på grunn av kompressibilitetsegenskapene til luftboblene. Disse egenskapene bidrar til en svekkelse av den dynamiske trykkendringen ved at amplituden reduseres og perioden økes, noe som har fordelaktige innvirkninger på rørsystemet.

Et forenklet eksperiment ble gjennomført i laboratoriet for å undersøke de praktiske løsningene for et system til luftinjeksjon. Resultatene viste at forekomsten av luft hadde positive innvirkninger som reduksjon av amplitude og en økt demping av trykkvariasjonene. Noen løsninger bør imidlertid justeres for å forbedre eksperimentet. Volumstrømmen og mengden av luft var usikre på grunn av vann som samlet seg i luftslangen. Dessuten viste det seg å være vanskelig å lukke sluseventilen til rett tid i forhold til luftinjeksjonen.

Ulike modeller av rørsystemer ble modellert og simulert i Flowmaster, med varierende utfall. Modellene er tilstrekkelige for å simulere vanlige rør med rask lukking av en ventil, men kommer til kort ved modellering av en blanding av luft og vann. Dette er fordi det kun er den reduserte lydhastigheten som er inkludert, og ikke den faktiske effekten av luftboblene.

Acknowledgments

The work presented in this thesis has been carried out at the Hydropower Laboratory, Department of Energy and Process Engineering at the Norwegian University of Science and Technology, NTNU. The objective has been to investigate the possibility of an air injection system to reduce and control pressure transients, both by analytical models in Flowmaster and a simplified laboratory experiment, and to suggest solutions for a more advanced experiment.

During the work with this thesis many people have contributed with advice and encouragement, which is much appreciated. I would like to thank my supervisor Morten Kjeldsen for motivation and theoretical support, in addition to research advisor Håkon Hjort Francke and PhD student Jørgen Ramdal for crucial help with the experiment. Bård Brandåstrø also deserves credit for help with selecting and purchasing equipment for the air injection setup. I am very grateful for valuable help in the laboratory, and would like to thank Joar Grilstad, Trygve Opland, Halvor Haukvik and Per Eivind Helmersen for always being helpful.

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Ingrid Kristine Vilberg Trondheim, December 22, 2010.

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Nomenclature

Symbol	Description	Unit
A	Cross-sectional area	$[m^2]$
a	Wave propagation velocity	[m/s]
a_L	Wave propagation velocity of the liquid	[m/s]
D	Diameter	[m]
E	Young's modulus	[Pa]
e	Wall thickness	[m]
g	Gravitational acceleration	$[m/s^2]$
h	Hydraulic pressure	[mWC]
H	Head	[m]
K	Compressibility modulus	[Pa]
K	Loss coefficient	_
k	Polytropic exponent	_
L	Pipe length	[m]
\dot{m}	Mass flow rate	[kg/s]
p	Pressure	[kPa]
Δp	Pressure increase	[kPa]
Q	Volumetric flow rate	$[m^{3}/s]$
T	Period	[s]
T_C	Valve closure time	[s]
T_R	Reflection time	[s]
t	Time	[s]
v	Velocity	[m/s]
V	Volume	$[m^{3}]$
V_G	Volume of gas	$[m^3]$
V_L	Volume of liquid	$[m^3]$
Z	Characteristic impedance	$[Ns/m^3]$

Greek symbols	Description	\mathbf{Unit}
α	Gas volume fraction	_
λ	Friction factor	_
γ	Isentropic exponent	_
ho	Density	$[kg/m^3]$
$ ho_G$	Density of the gas	$[kg/m^3]$
$ ho_L$	Density of the liquid	$[kg/m^3]$

Chapter 1

Introduction

1.1 Background

Pressure transients occur with any change of volumetric flow in a pipeline system, but the severity of the pressure transient depends on the rate of change of the volumetric flow and the design of the pipeline system. An intense pressure transient can have destructive effects on pipes and equipment, where high pressures can cause pipe ruptures or deformations and low pressures can result in a pipe collapse. Additionally, a pressure transient causes movement and vibration, which could interact with the resonance frequency of the pipe system and have fatal consequences.

Gas is normally undesired in a liquid in a pipeline system, but the presence of gas bubbles will reduce the amplitude and increase the damping of a pressure transient because of the reduced wave propagation velocity and the compressibility effects of the gas bubbles. This is a beneficial effect that could be utilized in emergency situations with a severe pressure transient.

1.2 Objective

The approach angle of this thesis was initially a theoretical study and design of an experiment to investigate the possibilities of an air injection system to control and reduce pressure transients. However, in agreement with supervisor Morten Kjeldsen, the main focus of the thesis has been the realization of a simplified experiment to test the selected solutions for air injection and to verify that an air injection system is beneficial for pressure transients. Consequently, some tasks in the original project description have been given less priority. The design study for gas injection is to some extent described in the theoretical section and also in the description of the advanced experiment. The design study for the setup of the actuation process and sensor analysis is mentioned in the description of the advanced experiment, but has not been thoroughly considered.

The pressure increase due to a sudden valve closure is dependent on the initial velocity of the fluid and the wave propagation velocity. The method described in this report will attempt to utilize the effects of air bubbles in water and the reduced wave propagation velocity to suppress intense pressure transients in pipe systems by injecting air into the pipe. This will result in a lower wave propagation velocity in a certain segment of the pipe. A simplified experiment has been carried out to investigate air injection methods and the selected solutions, and the results will be compared to analytical models in Flowmaster.

With regard to the original problem description, a design study of the setup of a more advanced experiment is presented. The plans for the advanced experiment are based on the theoretical study and experiences from the simplified experiment.

1.3 Outline

This thesis gives a general introduction of the theoretical aspects of pressure transients and the effects of gas bubbles in the liquid. The emphasis is given to sudden valve closures, as it is closely related to a turbine shutdown in a hydropower plant. Furthermore, the selected experimental solutions are presented as a simplified experimental study in Chapter 3. Several analytical models of pipe systems from Flowmaster are presented, including models of the laboratory setup. Additionally, a more advanced experiment is presented in Chapter 5. Results from the experiment and the Flowmaster simulations are presented in Chapter 6, followed by a discussion of the results and suggestions to further work.

Chapter 2

Theoretical background

An introduction to the topics in question is presented in this chapter. The theory is conveniently divided in two parts; namely system dynamics and effects of gas in a liquid.

2.1 System dynamics

System dynamics deals with the analysis and calculations of oscillations and surges caused by a change in the volumetric flow in a pipeline system. The fluid in the pipe can be considered rigid or elastic, which makes an apparent difference in the analytical approach. A rigid analysis does not take the elastic properties of the fluid and pipe material in to account. The wave propagation velocity is considered infinite and the fluid incompressible. However, a rigid analysis is insufficient for longer pipes, where the elasticity in the fluid and pipe material give rise to pressure transients in the pipeline when the fluid is compressed in front of a closed valve [7]. This section will focus on elastic analysis of pipe flow.

2.1.1 Pressure transients in piping systems

The governing equations for a pipe flow are the continuity equation and the equation of motion:

$$\frac{\partial H}{\partial t} + \frac{a^2}{g} \frac{\partial v}{\partial x} = 0 \tag{2.1}$$

$$g\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + \lambda \frac{v \left| v \right|}{2D} = 0$$
(2.2)

- H Piezometric head. H=h+z.
- h Hydraulic pressure [mWC]
- z Elevation [m]
- a Wave propagation velocity, [m/s]
- g Gravitational acceleration, $[m/s^2]$
- v Velocity, [m/s]
- λ Friction factor
- D Pipe diameter, [m]

The wave propagation velocity, a, is the speed of sound in the medium and is of great importance in the analysis of pressure transients. The speed of sound is given by Korteweg's formula [15], which is valid for a thin-walled pipe:

$$a = \sqrt{\frac{K/\rho}{1 + (K/E)(D/e)}}$$
 (2.3)

Here K is the compressibility modulus, ρ is the mass density, E is Young's modulus of the pipe material, D is the inner diameter of the pipe and e is the wall thickness.

Pressure transients are caused by unsteady flow in pipelines, which occurs from a change in operation, or a change in the boundary conditions of a system. Any change of volumetric flow in a pipeline system will cause a retardation or acceleration of the fluid masses, resulting in a dynamic change of pressure. Some of the most common operational changes are [15]:

- Valve closure or opening
- Pump startup or shutdown
- Load changes in hydraulic turbines

An uncontrolled pump trip, often caused by a power failure, will cause a low local pressure which can cause cavitation and have severe consequences. A rapid closure of a valve in extensive petroleum pipeline systems can cause an intense



Figure 2.1: Simple pipe system with valve

pressure transient. A sudden turbine shutdown in a hydropower plant will result in an abrupt closure of the guide vanes, which can be modeled as an instant closure of a valve. A common and illustrative example is a reservoir and a pipe with a valve in the end, as seen in Figure 2.1. A sudden closure of the valve will give rise to a pressure transient; also known as the water hammer phenomenon.

The water in the pipe has an initial constant velocity of $v_0 = \sqrt{2gH}$. The valve is then closed instantaneously, causing the water next to the valve to decelerate which results in an immediate pressure rise in front of the valve. The information of deceleration is propagating with the speed of sound through the pipe until it reaches the reservoir and the flow is stationary in the entire pipe. When the water has come to rest, there is an excess pressure in the pipe versus the reservoir causing the water to flow back to the reservoir with a velocity $-v_0$ until the pressure is equalized. As the valve is still closed there will be a negative pressure next to the valve, and this information will again propagate with the speed of sound to the reservoir and the water is again brought to rest. In order to equalize the pressure difference the water starts to flow into the pipe again, with a velocity v_0 . As the valve is still closed, the process will be repeated. The pressure variations in a frictionless pipe can be seen in Figure 2.2. In reality the friction effects cause the pressure oscillations to dampen eventually.

The situation described above constitutes one period of a pressure wave. The period of the pressure wave is $T = \frac{4L}{a}$, as shown in Figure 2.2. The time it takes for the pressure wave to propagate from the valve to the nearest water surface and back is called the reflection time, and is given as $T_R = \frac{2L}{a}$. The intensity of the pressure transient is dependent on the closure time, T_C of the valve and is significantly reduced if the closure time is increased. Two different scenarios are commonly used to distinguish the valve closure procedures:



Figure 2.2: Pressure variation in a frictionless pipe

- **Instant closure,** where $T_C < T_R$, will result in a severe pressure transient, a water hammer.
- **Increased closure time,** where $T_C >> T_R$, will result in a reduced amplitude of the pressure transient.

Unacceptably high pressures in a pipe can lead to permanent deformation and ruptures of the pipe and components, while a low pressure may result in a collapse of the pipe. A pressure transient can cause vibrations and movements of the pipe, and it is essential that the frequencies of the vibrations do not coincide with the natural frequencies of the system in order to avoid resonance.

The maximum pressure will arise in front of a valve when the valve is closed faster than the reflection time. The Joukowsky equation presents the increased pressure in front of a valve for an instantaneous closure [7]:

$$\Delta p = \rho a \Delta v \tag{2.4}$$

Here Δp is the pressure increase in front of the valve, ρ is the fluid density, a is the wave propagation velocity and Δv is the change of velocity of the fluid flow. The Joukowsky equation proves to be a sufficient estimation for the pressure increase when $T_C < T_R$.

For cases with a controlled closure of a value, where $T_C >> T_R$, an estimation of the increased pressure in front of the value is given as [7]:

$$\Delta p = \rho a \Delta v \frac{T_R}{T_C} \tag{2.5}$$

 T_R and T_C are the reflection time and the closure time, respectively.

2.1.2 Control and reduction of pressure transients

Several methods for reduction and control of pressure transients exist, where an increased closure time of the valve is a normal precaution. A controlled and slower closure of the valve results in a more gradual deceleration of the flow and the pressure oscillations in the pipeline are held to a minimum. The same effects can be achieved with a stepwise closure of the valve where the first 80 % of the valve closure is done rapidly while the remaining stages of closure are much slower. A similar procedure for the opening process includes a gradual initial opening [11].

Estimations of the pressure increase in front of a valve which is closed instantaneously and slowly is given in Equations 2.4 and 2.5, respectively. By applying $T_R = \frac{2L}{a}$ and $\Delta Q = \Delta v \cdot A$ to Equation 2.5 we get:

$$\Delta p = 2\rho \frac{\Delta Q}{T_C} \frac{L}{A} \tag{2.6}$$

Here ΔQ is the change in volumetric flow, L is the pipe length and A is the cross sectional area of the pipe. From Equation 2.6 it can be seen that in order to reduce the pressure rise in front of the valve, the ratio $\frac{L}{A}$ should be decreased. This would mean to decrease the pipe length L to the closest water surface, and a surge shaft in hydropower plants is a commonly used method to obtain this. A surge shaft can be located between the turbine and both the head and tail water, in order to change the rapid movements of the pressure transient into a slower mass oscillation between the surge shaft and the reservoir. In situations where a surge shaft is not feasible because of the topography, an air cushion chamber can be a solution in hydropower plants. Similar solutions are also used in other pipeline systems in the water industry, where air vessels and air cushion surge chambers are commonly used [11]. An example of the use of an air vessel to reduce a pressure transient can be seen in a Flowmaster model in Appendix D.

While technologies like the surge chambers and air vessels transform the rapid movement of a pressure transient to a slower moving mass oscillation, other techniques can be applied to reduce the amplitude of a pressure transient. For systems where unacceptably low pressures may occur, for instance after a pump failure, air valves can be installed to admit air into the pipeline and avoid a serious vacuum. On the other hand, pressure relief valves can be used to reduce the high pressure that occurs after a valve closure by releasing fluid from the pipe. The valve opens automatically and discharge a sufficient amount of liquid to reduce the pressure [11].

2.2 Effects of gas in a liquid

The effects of adding gas in a liquid are described in this theoretical part, as well as gas injection techniques and methods for measuring bubble size and gas content.

2.2.1 Bubble theory

The gas content of a gas-liquid mixture is normally described by a volume fraction, or void fraction given as:

$$\alpha = \frac{V_G}{V_G + V_L} \tag{2.7}$$

Here V_G is the volume of gas and V_L is the volume of liquid. With a certain gas content the gas-liquid mixture will obtain modified physical properties. A mixture of gas and liquid is considered as a fluid with density close to that of the liquid, while the compressibility of the gas-liquid mixture is mainly defined by the compressibility the gas [13].

Assuming that the gas is ideal and behaves according to the polytropic equation $pV^k = Constant$, the wave propagation velocity in a homogeneous bubbly flow can be expressed as [2]:

$$\frac{1}{a^2} = \left[\rho_L(1-\alpha) + \rho_G \alpha\right] \left[\frac{\alpha}{kp} + \frac{(1-\alpha)}{\rho_L a_L^2}\right]$$
(2.8)

Here ρ_L and ρ_G are the density of the liquid and gas phase, respectively, k is the polytropic exponent, p is the pressure and a_L is the wave propagation velocity of the liquid. A polytropic exponent of k = 1 implies an isothermal process and constant bubble temperature, while $k = \gamma$ means an isothermal process.

As presented in Table 2.1, the wave propagation velocity in air is much lower than in water, and the presence of air bubbles in the water will significantly alter the fluid properties. The wave propagation velocity for a water-gas mixture can be very low, depending on the void fraction of gas in the liquid and the pressure. This is showed in Figure 2.3. Note that the void fraction of gas is given as a percentage, and the wave propagation velocity in the water-gas mixture is even lower than in gas for relatively low void fractions.

A certain amount of gas is naturally present in a liquid as dissolved gas. When the pressure drops below vapor pressure, the gas will vaporize and form bubbles.

Medium	Speed of sound
Water	$1440 { m m/s}$
Stainless steel	$5800 \mathrm{~m/s}$
Water in steel pipe	1000-1200 m/s
Air	340 m/s

Table 2.1: Speed of sound in various media



Figure 2.3: Propagation velocity for water with varying air content [15]

2.2. EFFECTS OF GAS IN A LIQUID

This is known as cavitation and can cause serious damage, along with loud noise, when the bubbles collapse close to a surface. The vaporization will influence the dynamic behavior of the system, as the wave propagation velocity is reduced in that region. Consequently, the gas content in a liquid changes at varying pressures due to the vaporization and the wave propagation velocity is strongly dependent on the pressure. Low pressure regions will occur at the downstream side of a valve after a sudden closure or after a pump trip. The most serious situation occurs when the pressure is sufficiently low, resulting in column separation. Additionally, fluctuations of pressure due to turbulence can be a source of low pressure regions causing bubble formation.

The presence of dissolved air and gas in a liquid in a pipeline system is generally considered as undesirable. The air may accumulate in pockets and generate high shock loads when encountering valves or pipe bends [11]. However, for incidents where a reduction of the wave propagation velocity is favorable, a distribution of air bubbles may have beneficial effects. In the case of a pressure transient, the reduction of wave propagation velocity and the increased damping effects from the bubbles will result in a reduction of amplitude and an increased period of the pressure oscillations. Hence the presence of air bubbles may be advantageous in emergency situations.

2.2.2 Attenuation and scattering

Gas bubbles in a liquid will reduce the amplitude of a pressure transient due to the compressibility effects of the gas bubbles and the impedance differences in the medium.

Attenuation

Attenuation is achieved because of the compressibility effects of the gas bubbles which contributes to a reduction of the pressure amplitude through thermal damping. The compression and expansion cycle will depend on the oscillation frequency. For lower frequencies the bubble behavior is approximately isothermal, with the polytropic exponent k = 1. Due to the low oscillation frequency there is sufficient time for thermal conduction between the gas and liquid. On the contrary, at higher frequencies the bubble behavior is initially isentropic, with $k = \gamma$. There is insufficient time for heat transfer as a result of the high oscillating frequencies [2].

Nevertheless, conditions in the bubble are often assumed to be isothermal, because an adiabatic compression of the whole liquid-gas mixture results only in a small rise of temperature in the liquid. The higher temperatures in the bubble are reduced by heat transfer across the bubble boundary [3].

It was previously stated that the wave propagation velocity was reduced in a gas-liquid mixture, which is true for frequencies below the bubble resonance frequency. However, for frequencies higher than the bubble resonance frequency the propagation velocity can be higher than the sonic speed in pure water [3]. As this study deals with the frequencies below the resonance frequency, no further attention will be given to frequencies above resonance frequency.

Scattering

Scattering can be achieved by allowing the pressure transient to propagate between regions of different impedances. The theoretical introduction to impedance is mainly based on Bodèn et al. [1].

The characteristic impedance of a medium, Z, is given as:

$$Z = \rho \cdot a \tag{2.9}$$

 ρ is the fluid density and *a* is the wave propagation velocity. The impedance is of great importance when dealing with waves, as a wave generates a reflected and a transmitted wave when it encounters a change in the medium, like at the border between regions of different impedances.

Consider the case shown in Figure 2.4, where a wave is approaching a border between regions of different impedances. The reflection and transmission coefficients are defined as:

$$R = \frac{\rho_2 a_2 - \rho_1 a_1}{\rho_2 a_2 + \rho_1 a_1} = \frac{1 - \frac{\rho_1 a_1}{\rho_2 a_2}}{1 + \frac{\rho_1 a_1}{\rho_2 a_2}}$$
(2.10)

$$T = \frac{2\rho_2 a_2}{\rho_2 a_2 + \rho_1 a_1} = \frac{2}{1 + \frac{\rho_1 a_1}{\rho_2 a_2}}$$
(2.11)

Equations 2.10 and 2.11 show that the reflection coefficient is always real and can be both positive and negative, while the transmission coefficient is always real and larger than zero. As the reflection coefficient can attain both positive and negative values, the reflected wave will either be in phase or 180° out of phase. The transmitted wave will always be in phase with the incoming wave.



Figure 2.4: Wave approaching border between zones of different impedances.

A wave propagating from a region with high impedance to a region with lower impedance like from water to a water-air mixture, $\rho_1 a_1 > \rho_2 a_2$, will according to Equations 2.10 and 2.11 generate a transmitted wave in phase and a reflected wave 180° out of phase. This will be illustrated with a Flowmaster model of a propagating pressure pulse in Chapter 4.

2.2.3 Gas injection

Gas injection has been subject for many experiments and theoretical studies. Older studies have a more practical approach, which is of interest in this study.

One of the challenges regarding injection of gas is to obtain a favorable bubble size. According to Silberman [8], the bubble volume, V, is only dependent on the gas flow rate Q and the velocity of the liquid, U, for the case of gas injection into a pipe of moving fluid. The assumption is that the gas is injected as a jet from the orifice.

$$V_{bubble} = \frac{12,96}{\sqrt{\pi}} \left(\frac{Q}{U}\right)^{3/2} \tag{2.12}$$

The assumption of an approximately spherical shape of the bubbles yields the following expression for the bubble diameter, D. It is noteworthy that the bubble

volume and diameter depend neither on the diameter of the orifice nor the gas or liquid properties.

$$D_{bubble} = 2, 4\sqrt{\frac{Q}{U}} \tag{2.13}$$

These equations can be applied to find the size of the largest bubbles, whereas the size distribution of smaller bubbles must be obtained experimentally. Due to bouyancy the bubbles will have a rising velocity, which depends on gravity, the densities of gas and liquid and the viscous drag on the bubble.

The gas can be injected in two ways, namely as a jet from an orifice at the pipe wall or through submerged injection [9]. The injection can be optimized for several purposes by varying parameters like direction of injection with respect to the flow, where the gas can be injected in parallel, normal or opposite direction to the liquid flow.

Wall orifice injection

A wall orifice can have different geometries, but the jet will have an approximately circular cross section because of the surface tension. It is shown that gas injection from a wall orifice into a liquid cross flow results in the smallest bubbles [8].

Submerged nozzle injection

Gas injection through a submerged nozzle is beneficial in terms of a centered distribution of bubbles. A submerged injection is practical in terms of varying the direction of gas injection with respect to the flow. On the other hand, the submerged nozzle injector is intrusive on the flow and causes a change in the liquid velocity field.

2.2.4 Measurement techniques

Silberman's bubble size theory assumes a spherical bubble shape where only the size of the largest bubbles can be calculated, which leaves the question of bubble size distribution unanswered. It would be interesting to investigate the effect of bubble size on the reduction of the pressure amplitude, hence the size and distribution of the bubbles must be obtained. Size and distribution of gas bubbles in a liquid can be obtained by various measurement techniques, but a majority of
2.2. EFFECTS OF GAS IN A LIQUID

the gas content measuring techniques utilize the changed physical properties and bulk quantities [6]. Many of the following measurement techniques are widely used in cavitation research for investigating gaseous cavitation.

Acoustic techniques

The acoustic measurement methods make use of the modified acoustical property of the liquid with a certain gas content, as the gas will result in a reduced wave propagation velocity. This is utilized by ultrasonic flowmeters that measure the time it takes an acoustic signal to propagate a certain length.

While the wave propagation velocity needs to be known in the most basic setups of an acoustic measurement system, the more advanced commercial systems are independent of the wave propagation velocity because of a feed-back system [6]

The main assumption in many of the acoustic methods is a homogeneous distribution of gas, i.e. the gas bubbles are evenly distributed in the liquid. This assumption is not necessarily true.

Conductivity probes

The presence of air bubbles in a liquid will modify the electric properties of the liquid, i.e. the electrical conductivity. This is measured with conductivity probes, and the regions containing gas bubbles can be found, along with the void fraction.

Wire mesh sensors were used by Prasser et al. [4] to measure bubble size. The wire mesh are based on the same principles as the conductivity probes, but are combined to constitute a fine wire mesh. The measurement technique can obtain the bubble size distribution as well as the void fraction. This type of measurement technique is intrusive on the flow, but is relatively inexpensive compared to other available bubble size measurement systems.

Laser and optical techniques

Several laser and optical techniques to measure accurate bubble size exist, which have the advantage of not being intrusive on the flow. Although more expensive than electronic-based sensors, laser and optical techniques prove to be accurate and widely used measurement techniques.

A pilot study by Keller and Zielke [5] in 1976 investigated the applicability of the Scattered Light Counting Method for measuring the size and number of bubbles

in water and the void fraction. The Scattered Light Counting Method is based on the fact that air bubbles will scatter the incoming light from a laser beam and is registered by an optical receiver. The output signal is proportional to the scattered light intensity, and hence the bubble size can be determined.

Adaptive Phase-doppler Velocimetry (APV) is a laser technique that is commonly used in cavitation research [6]. The doppler signal is used to calculate the velocity, while the phase difference is used to calculate the bubble size. Other laser techniques involve laser doppler velocimetry (LDV) and the use of gamma rays.

Optical techniques also involve the use of a light source to make the bubbles visible. Both a back-light and a laser sheet can be used for this purpose, but neither of the methods provide any volume information.

Chapter 3

Pilot study

The pilot study was carried out to investigate the practical effects of injecting air into a pipe flow. As this was considered a demonstrational experiment the main focus was to test the chosen solutions and control that the air injection was beneficial for the pressure oscillations, rather than to obtain accurate results from the measurements. The different aspects of the experimental planning and procedure are presented in this chapter.

3.1 Laboratory set up

To be able to carry out the experiment within the given time frame, the existing pipe setups and solutions at the Hydropower Laboratory were used. A straight horizontal pipe with a gate valve was found most suitable for this purpose, and the pipe length was approximately 26 meters from the bend to the gate valve. The gate valve was controlled manually by a hydraulic aggregate. The pipe section can be seen in Figure 3.1.



Figure 3.1: Pipe location in the Hydropower Laboratory



Figure 3.2: Air injection setup

3.2. EXPERIMENTAL PROCEDURE

The air injection setup is shown in Figure 3.2, where the components are enumerated. The components are described in the following table:

Number	Component
1	Pressure regulator
2	Air tank of 12 liters
3	Manometer
4	Magnetic electric valve
5	Throttle valve
6	Flowmeter
7	Ball valve

Table 3.1: Air injection setup

The pressure tank was connected to the compressed air system in the laboratory through a pressure regulator. The compressed air from the laboratory was around 6 bar, but the pressure was adjusted to 4 bar with the pressure regulator. The magnetic valve was chosen because of the ability to open and close quickly, thus ensure an instantaneous air supply, and it was controlled electrically through a LabView program. The throttle valve was installed to enable a regulation of the air flow rate, while the actual air flow rate was measured in the flowmeter. A ball valve was placed between the flowmeter and the air hoses in order to shut off the air supply if necessary. A circular hose setup was put together to distribute the air through four existing inlets on the pipe.

3.2 Experimental procedure

The pressure in the system was provided from the upper water tank which is shown in Figure 3.1. The water level in the tank was kept constant at 3,5 m, in order to ensure a constant pressure in the system.

The reflection time T_R for the laboratory setup was approximately 0,07 s. To impose a closure time faster than the reflection time would be practically impossible, as well as possibly destructive for the system. However, the very last part of the closure process would always be faster than the reflection time, and that effect was investigated further in the experiment.

The implementation of a measurement and control system for the air injection would be too complex to include in this pilot study, thus a simpler solution was used. The effects of air added to a pipe flow of water were investigated, but the air injection was activated manually from a LabView program instead of being activated by pressure sensors and an actuation system.

Several parameters could be varied in this experiment with the existing setup:

- Volumetric flow rate of water, controlled by the gate valve
- Duration of air injection
- Flow rate of injected air, controlled by the throttle valve
- Pressure of injected air
- Number of injection points
- Location of injection
- Closure time of the gate valve

Only two of the parameters were varied in this experiment, namely the volumetric flow rate and duration of air injection. The flow rate of injected air was measured, but not used as a variable parameter. Two different cases were studied; one where the air was injected into the pipe prior to closure of the gate valve and a second case where the air was added right after the valve was closed. Three initial positions of the gate valve were used, namely 35 %, 20 % and 10 % opening, which corresponded to flow rates of 260 l/s, 160 l/s and 80 l/s, respectively. The closure time could be regulated manually on the hydraulic aggregate, and a closure time of approximately one second was used in the experiments. The gate valve was opened completely between the air injection tests to flush out air bubbles from the previous injection.

A LabView program made by Håkon H. Francke in Flow Design Bureau (FDB) was used for logging the measured values. The front panel can be seen in Figure B.1 in Appendix B. The duration of air injection was entered in the LabView program, which controlled the opening and closure of the magnetic value connected to the air tank. Durations of air injection of one to four seconds were tested in the experiment.

3.2.1 Challenges with the experimental procedure

Several difficulties arose during the experiment, mainly involving the air injection setup and the manual positioning and closure process of the gate valve. As seen in Figure 3.2, the air exiting the pressure tank had to pass through the rotameter and the air hoses before entering the pipe, which could cause a delay in the air injection process. Additionally, it was found that the air hose was partially filled with water, forcing the air to displace the water before entering the pipe. Thus the amount of air which actually entered the pipe was uncertain.

The gate valve was controlled manually by an aggregate, and it was challenging to find the exact valve opening ratio when repeating the experiments. However, with the use of the signals from the string-based position indicator that were logged in the LabView program, the valve opening ratio was nearly the same for each repeated experiment. A more challenging aspect with the closure of the gate valve was the timing of closure in connection with the air injection. As it was interesting to investigate the effects of air admission at different points of time, i.e. before, during and after the closure of the gate valve, the timing of the closure of the gate valve was decisive.



Figure 3.3: Schematic overview of the measurement setup

3.3 Instrumentation

An illustration of the laboratory system and the measurement setup is shown in Figure 3.3. The following measurement instruments have been used in this experiment.

3.3.1 Volumetric flow measurements

The flowmeter in the laboratory was used for measuring the volumetric flow of water. The flowmeter was placed on the vertical pipe from the water tank, and can be seen in Figure 3.1. The electromagnetic flowmeter was of the type Krohne Aquaflux IFS4000, and as seen in Figure 3.3 the volumetric flow was logged in the LabView program.

3.3.2 Position measurements

The position of the valve was measured with a string-based position indicator and logged in the LabView program. A linear releationship between the output volt signal and the actual position was assumed.

3.3.3 Air flow measurements

The volumetric flow rate of the injected air was measured prior to the experiment with a rotameter of the type Variable Area Flowmeter Glass Tube 10A1190. The maximum flow rate for the rotameter was 6,62 NL/s, and a percentage of the maximum flow rate was monitored manually. In order to obtain the actual flow rate, an equation which takes the pressure difference into account was used.

3.3.4 Pressure measurements

Two piezoelectric transducers and two regular pressure transducers were used for the pressure measurements. One of each transducer were used in two measurement points, fairly evenly distributed along the pipe length. The measurement points and the location of the air injection can be seen in Figure 3.3

GE Druck PLX 610

The regular pressure transmitters were of the type GE Druck PLX 610, and had a range of 0-10 bar of absolute pressure, which corresponds to an output range of 4-20 mA. Ideally, the pressure transducers should be flush-mounted with the inner wall of the pipe, but for simplicity the transducers were mounted on a connection.

Kistler 701A

The piezoelectric transducers were of the type Kistler 701A, which only gave an output signal when a change of pressure occurred. The transducers were connected to a charge meter, where the measuring range, sensitivity and voltage output was entered. The measuring mode was set to medium, the sensitivity for 0-250 kPa was entered and 100 kPa was set as the measuring range. The scaling of the voltage output was set to 10 kPa/V. It was decisive that the piezoelectric transducers were flush-mounted, and a transition had to be made to fit the special threads of the transducers.

3.4 Uncertainty

The uncertainty of an experiment implies the error limits in the measurements, often given with some level of confidence. The errors in an experiments are divided in two categories, namely systematic and random errors [14]:

- **Systematic errors** are repeatable and consistent errors in a measuring system. The main source systematic errors result from the calibration process, while others are loading errors from the setup of the measuring device.
- **Random errors** are caused by the lack of repeatability in the experiments. The measuring system, esperimental system or the environment can be a source of random errors. Uncontrolled variables should be eliminated to minimize the random errors in an experiment.

As this was a pilot experiment, uncertainty analysis has not been a main focus. However, a few factors should be mentioned and taken into account for further development of the validity of the experiment. The main sources of inaccuracy are described below.

Air injection setup

The exact amount of air injected in the pipe was unknown because of the extensive air hoses that connected the throttle valve to the pipe inlets, via the rotameter. Ideally the hose length should be minimized to avoid a time lag in the air injection. Additionally, the air hose was partially filled with water, which also contributed to a more uncertain flow rate of air.

Valve closure procedure

As the valve was controlled manually by the hydraulic aggregate, a precise positioning of the valve was difficult. The initial volumetric flow rate could be somewhat different for the repeated experiments. The closure speed of the valve could be regulated with a handle on the hydraulic aggregate, and some uncertainty was also related to that adjustment.

Flowmeter

The flowmeter proved to have a time lag in the output signal, and withheld quite high output values even after the gate valve was closed. Thus it was difficult to obtain a relationship between the valve closure and the volumetric flow of water.

3.4.1 Calibration of the measurement equipment

The GE Druck 610 pressure transducers were calibrated one month prior to the experiment, and a quick control of some calibration points with a dead weight manometer confirmed the calibration. Therefore a further calibration was found unnecessary.

The piezoelectric transmitters required a more comprehensive calibration, and the Hydropower Laboratory did not have the necessary equipment. Consequently, the piezoelectric transmitters were not calibrated, but the measuring sensitivity given from the manufacturer was entered in the charge meters.

The remaining measurement instruments were not calibrated as it was found unnecessary for this simplified experiment. It was not decisive to get accurate results in the demonstrational experiment, only to investigate the positive effect of air injection and examine the chosen solutions for the air injection setup. A further development of the experiment should include a more thorough calibration process of all the measuring equipment involved in the experiment.

Chapter 4

Computational models

The modelling procedures for the different pipe systems in Flowmaster are described in this chapter. Flowmaster is a commercial software for simulation of thermal and fluid systems, and uses the method of characteristics to calculate the pressure behavior in transient pipeline flow. The method of characteristics is described in Appendix A.

4.1 Simple pipe systems

Gas injection in a pipe segment will result in a reduced speed of sound in that segment, in addition to an increased damping due to the compressibility effects of the air bubbles. As the Flowmaster components were unable to include injected air bubbles, a simpler solution had to be used where only the effect of reduced wave propagation velocity was included. Several models were created in Flowmaster to illustrate the effect of a lower wave propagation velocity in a certain segment of the pipe, where a short pipe with a relatively low speed of sound was added to the system. The setup of a basic pipe model is shown in Figure 4.1 and the different components are described in Table 4.1. Here H is the liquid level above base level, z is the elevation, L is the pipe length, D is the pipe diameter and a is the wave propagation velocity.



Figure 4.1: Basic pipe setup

Number	Component
1	Reservoir, constant head. $H{=}50 \text{ m}, z{=}50 \text{ m}$
2	Elastic pipe. L=480 m, D= 0.5 m, a= 1200 m/s
3	Elastic pipe. L=20 m, D= 0.5 m, a= 1200 m/s, a= 200 m/s, a= 100 m/s
4	Butterfly valve, initially open. $z=10 \text{ m}$
5	Valve controller. $T_C = 0.5 - 20$ s.
6	Elastic pipe. $L=500 \text{ m}, D=0.5 \text{ m}$
7	Reservoir, constant head. $H=9 m$

Table 4.1: Components in the model of a simple pipe system

4.2 Pressure pulse

The gas injection will cause a reduced wave propagation velocity, which results in a lower impedance in a certain region of the pipe. Because of the impedance difference the incoming pressure wave will generate a reflected and a transmitted wave, as mentioned in Section 2.2.2. This phenomenon is illustrated with a Flowmaster model with a pressure source generating a pressure pulse which propagates through the pipe. This model is shown in Figure 4.2.

To be able to investigate the propagation of the pressure pulse, the minimum number of required reach lengths of the pipes was increased from 5 to 99, which is the maximum number of reach lengths. With this setting, the pressure could be monitored for every 5 cm of the pipes of 5 meter, and accurate results for the pressure pulse propagation could be obtained. In order to get even more accurate results, the pipe lengths would have to be reduced while the number of reach lengths were kept at the maximum level.



Figure 4.2: Flowmaster setup for pressure pulse calculations

Number	Component
1	Pressure source, p=100 kPa
2,3,5,6	Elastic pipe, L=5 m, D= $0,3$ m, a= 1200 m/s
4	Elastic pipe, L=5 m, D= $0,3$ m, a= 300 m/s
7	Pressure source, p=200 kPa
8	Pressure controller, p=100 kPa, p=400 kPa for 2 ms

Table 4.2: Overview of components in the pressure pulse model

4.3 Experiment

The pipe setup of the simplified experiment described in Chapter 3 has been modeled in Flowmaster. The simulation model of the experimental setup has been tested both with and without air injection, and the length of the pipe segment with reduced wave propagation velocity was varied in order to simulate durations of air injection from one to four seconds. The model can be seen in Figure 4.3, and the components are defined in Table 4.3.

The length of the pipe with reduced wave propagation velocity was found by multiplying the velocity of the water with the duration of air injection. For the case with one second of air injection, the corresponding pipe length with air was 2,26 m. The flow rate of air was measured to be approximately 0,93 l/s, which implies a void fraction of 0,6 %. The wave propagation velocity was calculated from Equation 2.8, and was found to be approximately 170 m/s. A closure time of one second, approximately the experimental closure time, was used in the Flowmaster model.

As the pipe setup in the laboratory included several bends and other connected pipes, it was interesting to investigate the effect of these bends and pipes on the



Figure 4.3: Flowmaster model of the laboratory setup

Number	Component
1	Reservoir, constant head. $H=3,5 \text{ m}, z=12,1 \text{ m}$
2	Elastic pipe. L=8 m, D= 0.6 m, a= 1100 m/s
3	Elastic pipe. L=1,1 m, D=0,6 m, a=1100 m/s
4	Elastic pipe with blank end. L=12 m, D=0.6 m, a=1100 m/s $$
5	Elastic pipe. L=1,8 m, D=0,3 m, a=1100 m/s
6	Elastic pipe. $L= 5,2 \text{ m}, D=0,3 \text{ m}, a=1100 \text{ m/s}$
7	Elastic pipe. L= 2,26 m, D= 0,3 m, a=170 m/s
8	Elastic pipe. L= $18,54 \text{ m}, D=0,3 \text{ m}, a=1100 \text{ m/s}$
9	Gate valve, initially 35 % open.
10	Valve controller. $T_C=1$ s
11	Elastic pipe. $L = 10 \text{ m}, D=0.3 \text{ m}.$
12	Reservoir, constant head. H=5

Table 4.3: Components in the model of the experimental setup

solution. Thus, different models both without pipe 12 in Figure 4.3 and the bends have been analysed. The results from these simulations will indicate whether or not a different pipe setup should be used for the more refined experiment.

4.4 Valve characteristics

According to Flowmaster's user guide, the valve loss characteristics could be decisive in terms of giving a reasonable result for the pressure increase in front of the valve. The valve loss curves in Flowmaster are only extended to around 10 % opening, and an infinitely high loss coefficient is used for the remaining closure. This could result in too high pressures in some situations [12].

Flowmaster's definition of the pressure/flow equation for control valves is:

$$p_1 - p_2 = \frac{K\dot{m}_2 |\dot{m}_2|}{2\rho A^2} \tag{4.1}$$

Here p_1 and p_2 are the pressure at node 1 and 2, K is the dimensionless loss coefficient, \dot{m}_2 is the mass flow rate at node 2, ρ is the fluid density and A is the valve area. For the simulations of the laboratory system with a valve opening ratio of 0,35, the loss coefficient K found from Equation 4.1 is 0,45.

The valve loss coefficient for the gate valve in the Flowmaster simulations of the laboratory setup is given as a curve for loss coefficient versus opening ratio and is shown in Figure 4.4. The figure shows that the loss coefficient for a valve opening of 0,35 is approximately 4, which is about 10 times as high as the calculated K of 0,45. It can be seen that a valve opening of 0,77 corresponds to the loss coefficient of 0,45, and the valve opening of 0,77 was used to obtain the same flow rate as the experiments, namely 260 l/s.

4.5 Limitations

Although the Flowmaster models were illustrative, they have some shortcomings. The gas injected into a real pipe with a flow of water will naturally be transported downstream with the water, which the Flowmaster models with a pipe segment of lower wave propagation velocity are unable to take into account. The increased damping due to the compressibility effects of air bubbles will not be included in the Flowmaster model, as only the decreased wave propagation velocity is taken into account. Additionally, the case of a rapid valve closure followed by an air



Figure 4.4: Gate valve loss coefficient [12]

4.5. LIMITATIONS

injection cannot be investigated in Flowmaster, as the pipe with reduced wave propagation velocity must be included in the model from the start.

Chapter 5

Planning an advanced experiment

The previously presented simplified experiment gave some indications of possibilities and improvements of the existing setup. With regard to the original problem description, idea for a more advanced experiment is presented in this chapter, including suggestions for measuring bubble size and distribution and the actuation system for the air injection setup.

The air injection system is supposed to be used for unintended and rapid closures of valves and emergency situations to avoid the dangerous consequences of a severe pressure transient. The system must be able to identify the situations where a pressure transient occur and inject air as fast as possible to reduce the amplitude of the pressure transient. This is a rather challenging procedure which requires an extensive control and actuation system, in addition to the actual injection setup to obtain a favorable bubble size and distribution.

A decition must be made of whether or not the existing pipe setup in the laboratory should be used. Further refinement of the simplified experiment will probably give some indications to this choise, but the existing pipe setup worked well for the pilot study. In order to be able to visually inspect the air injection and photograph the bubbles, a part of the pipe test section should be made of plexiglass, preferably close to the air injection setup.

5.1 Bubble size and distribution

During the development and design stages of the air injection system it is important to investigate the effects of the different parameters described in Section 2.2.3 in order to obtain a favourable bubble size and distribution.

The air could either be injected through a wall orifice or a submerged nozzle. Injection through a wall orifice would be the best solution for future applications of the air injection system as it does not interfere with the pipe flow, and should be used in the advanced experiment.

In addition to the basic air injection, a control and actuation system for the air injection and a device to ensure a favourable bubble size and distribution is required. As mentioned in the theoretical part, the bubble size depends on the gas flow rate and the velocity of the liquid. A uniform bubble size and diameter is desirable in order to calculate the effects of the bubbles.

The decision of what kind of measurement system to use to measure the bubble size and distribution in addition to void fraction will depend on many factors. The availability of such a system is of course decisive, as it would be favourable to make use of the measurement equipment at the university. If a measurement system had to be purchased, the price and accuracy of the device would be deciding factors. A laser based measurement system should be considered as they have proven to be accurate and practical.

5.2 Control and measurement systems

As the title of this thesis implies, the air injection system was motivated by an airbag for cars. The airbag must inflate in a fraction of a second in order to make the car passenger's deceleration as smooth as possible. However, it is the actuation process which is the most interesting in this context. The inflation sensors receive information from an accelerometer constantly monitoring the car's acceleration and deceleration, and will actuate the inflation process as soon as critical values are detected [10].

The control system for the air injection is thought to function in the same way, constantly measuring and monitoring the pressure in a pipe and actuating the air injection as soon as the monitored parameters reach critical limits. As for the car airbag, the actuation time is crucial and should be minimized in order to achieve the desired effect on the pressure transient.

The continuous measurements and rapid actuation process require an extensive and advanced control system. An outline for the control and actuation system is given below, where solutions to the control process are suggested.

The air injection system could be activated by a sensor connected to the pressure measurement system, and the system would be activated as soon as unacceptable pressures were registered. Depending on the highest acceptable pressures in a system, the limit for actuation of the air injection system should be set. Alternatively, the sensor could be linked to a position indicator on the valve and activated the air injection system if the registered closure time was too fast.

The optimal duration of air injection would have to be calculated and customized for each application, as the optimal amount of air will depend on the system. The optimalization process could be carried out with a simulation of the system in question, which requires a precise and comprehensive computer program. As for the experimental setup in the laboratory, the amount of air corresponding to one second of air injection proved to have significant effects on the pressure transient and could provide a good starting point for the injection process. As the pressure would be monitored continuously, the air injection system would be activated once more if the pressure was still exceeding the set limit for the system.

Chapter 6

Results

Results from the experiment and simulations in Flowmaster are presented in this chapter. The results are further discussed in the following chapter.

6.1 Experimental results

After the completion of the test rig, a number of experiments were carried out. The parameters in question were the duration of air injection and the flow rate of the water which was controlled by the opening of the gate valve. The air flow rate was measured by a rotameter and calculated to be 0,93 l/s. As previously mentioned, some uncertainty was involved in the actual flow rate of air as the air hose was partially filled with water.

Two different cases were investigated. In the first case the air was injected prior to closure of the gate valve in order to practically investigate the effect of bubbles in the liquid. Air was admitted right after the valve closure in the second case to demonstrate the immediate effect of air injection on a pressure transient. Both cases were repeated with no air injection and duration of air injection of one, two and four seconds.

As the gate valve was operated manually, some uncertainties are related to the initial valve position before closure and the timing of the valve closure.

Tests were performed at a liquid flow rate of 260 l/s, where the flow rate was achieved by a 35 % opening of the gate valve. The following figures show the pressure oscillations, closure process of the gate valve and duration of air injection.

Figure 6.1 shows the pressure oscillations following the closure of the gate value without any air injection. The maximum peak-to-peak value, Δp_{max} , of the amplitudes was 139,6 kPa and the period T, was 0,153 seconds, as illustrated on the figure.

The period of the pressure oscillations from the experimental results was approximately 0,153 seconds. The period is also given by the equation $T = \frac{4L}{a}$, which gives a period of 0,147 seconds assuming a wave propagation velocity of 1100 m/s.



Figure 6.1: Pressure oscillations after closure of the gate valve

Figure 6.2(a) and 6.2(b) show the pressure oscillations after one and three seconds of air injection, and it can be seen that the amplitude of the pressure oscillation was reduced and the period increased due to the presence of air.



(a) Pressure oscillations with one second of air injection



(b) Pressure oscillations with three seconds of air injection

Figure 6.2: Air injection prior to closure of the gate valve

As previously mentioned the air injection system will be activated when unacceptably high pressures are registred by a sensor, and hence the air will be injected after a valve closure. This case has been investigated here, where the manual closure of the valve was followed by an air injection of varying duration. Figure 6.3(a) shows the pressure oscillations with one second of air injection. The pressure oscillations after the valve closure for cases with and without air incjection are compared in Figure 6.3(b). It can be seen that the presence of air clearly reduce and dampen the pressure oscillations.

The same case was also tested with air injection of two and four seconds, and the results are presented in Table 6.1. Q is the flow rate, Δt_{air} is the duration of air injection, Δp_{max} is the maximum peak-to-peak value and T is the period. Δp_{max} and T are shown in Figure 6.1. $\Delta p_{air}/\Delta p$ is the ratio of the pressure amplitude 0,5 s after air injection and the pressure amplitude without air injection at the corresponding time. This ratio was chosen to illustrate the increased damping due to the air injection.

The cases where the air was injected prior to closure of the gate valve are marked with *. The valve was closed too late in these cases, and the air was injected either before or during the closure of the valve.

Q, [l/s]	$\Delta t_{air}, [s]$	Δp_{max} , [kPa]	$\Delta p_{air}/\Delta p$	T, [s]
270	0	139,6		$0,\!153$
	1	104,8	0,31	$0,\!198$
	2*	77,4	$0,\!30$	0,23
	4*	112,9	0,21	$0,\!38$
160	0	202,6		0,153
	1	203,9	0,42	$0,\!175$
	2*	174,9	0,31	0,392
	4*	$273,\!6$	0,27	0,40
80	0	119,1		$0,\!153$
	1*	75,3	$0,\!60$	0,18
	2*	80,0	0,30	0,41

Table 6.1: Overview of experimental results



(a) Pressure oscillations with 1 second of air injection



(b) Pressure oscillations with and without air injection

Figure 6.3: Air injection immediately after closure of the gate valve

6.2 Results from Flowmaster simulations

6.2.1 Simple pipe systems

An instant closure of the valve in Figure 4.1 was simulated in Flowmaster. The reflection time, T_R , for this system with a uniform wave propagation velocity was 0,833 s. The valve was closed in 0,5 s, which gave rise to a water hammer. In addition to simulations of the standard pipes, the simulation was repeated for pipe setups with a short pipe with reduced wave propagation velocity of a=200 m/s and a=100 m/s. The pressure in front of the valve was investigated for the three cases, and the results are shown in Figure 6.4.



Figure 6.4: Pressure variation after instantaneous valve closure

As mentioned in the theoretical section, an estimate of the pressure increase in front of the valve after an instantaneous closure is given by $\Delta p = \rho a \Delta v$. By applying this equation to the pipe system described above the pressure increase would be $\Delta p = 7,48 \cdot 10^3$ kPa, for the ordinary pipe. The result from the

Type	L, [m]	a, $[m/s]$	$p_{max}, [10^3 k Pa]$	T, [s]
Ordinary	500	1200	7,97	$1,\!67$
Modified	480 + 20	200	6,24	3,44
Modified	480 + 20	100	3,47	6,34

Table 6.2: Overview of comptational results for a simple pipe setup

Flowmaster simulation gave a pressure increase of $7, 59 \cdot 10^3$ kPa. Figure 6.4 shows that the pressure oscillates between positive and negative values. However, the pressure in normal water, which contains dissolved gas, will never reach negative values due to cavitation when the pressure drops below vapour pressure.

The period of the pressure oscillation, found by the equation $T = \frac{4L}{a}$, was 1,67 s for the ordinary pipe. From Flowmaster simulations with a sufficient time resolution it was also found that the wave period is 1,67 s.

It can be seen that the cases with a pipe segment of reduced wave propagation velocity have a reduced amplitude of the pressure oscillations, and also an increased period. The maximum pressure of the first amplitude is reduced to $6, 24 \cdot 10^3$ kPa for the case with a pipe segment with a=200 m/s, and further decreased to $3, 5 \cdot 10^3$ kPa for the pipe with a region of a=100 m/s.

Figure 6.4 and Table 6.2 shows how the amplitude was reduced and the period increased as the wave propagation velocity in a certain pipe segment was reduced. It was also interesting to see how the amplitude and period was affected by different placement of the short pipe segment with reduced propagation velocity, and the simulations were repeated for different locations of the pipe with reduced wave propagation velocity. Figure 6.5 shows the pressure oscillations for the ordinary pipe and for a pipe with reduced propagation velocity in a section in the middle, as the forth case presented in Table 6.3. Table 6.3 presents an overview of the results from different placements of the pipe segment, all for a propagation velocity of 200 m/s in the short pipe segment. The placement is given in terms of x, which is the distance along the pipe starting from the upstream reservoir. The case with x=480 m and L=20 m is equal to the case presented in Figure 6.4, but is included here as a comparison.

The Flowmaster simulations for the ordinary pipe proved to correspond well with the estimated pressure increase for $T_C < T_R$, but the same was not the case for the results with an increased closure time. A closure time of 5 seconds was applied to the pipe model shown in Figure 4.1. According to the equation for estimated pressure rise, $\Delta p = \rho a \Delta v \frac{T_R}{T_C}$, the pressure infront of the valve would increase with 1, 17 \cdot 10³ kPa. The results from the Flowmaster simulations, however, gave a pressure increase of $4, 21 \cdot 10^3$ kPa. The same overestimation was found in all



Figure 6.5: Comparison of normal pipe and pipe with reduced propagation velocity in the middle

Placement	L, [m]	$p_{max}, [10^3 \text{ kPa}]$	T, [s]
x=495	$5 \mathrm{m}$	8,76	$2,\!17$
x=490	10 m	7,33	2,63
x=480	20 m	6,24	3,44
x=240	20 m	12,13	0,8
x=245	10 m	12,37	0,8

Table 6.3: Varying placement and length of pipe with reduced wave propagation velocity

cases for increased closure times, and the resulats are presented in Table 6.4. Δp_{est} is the estimated pressure increase and Δp_{FM} is the pressure increase from the Flowmaster simulations.

Closure time, [s]	Δp_{est} , [10 ³ kPa]	Δp_{FM} , [10 ³ kPa]
3	1,96	5,76
5	1,17	4,21
10	0,59	1,83
20	0,29	0,74

Table 6.4: Pressure increase for $T_C > T_R$

Pipe setups including pipe segments with lower propagation velocity were also tested with an increased closure time, and the same tendencies as the rapid closure were found; namely a reduction of amplitude and an increased period. The cases with increased closure time were not investigated closer.

6.2.2 Pressure pulse

The results from calculations of the pressure pulse described in Section 4.1 can be seen in Figure 6.6, where the pulse propagation is shown with intervals of 4 ms. A pipe segment with a reduced wave propagation velocity is included from x=10 m to x=15 m, which is emphasized with red lines in the figure. The reduced wave propagation velocity will cause an impedance reduction in that pipe segment.

The pressure pulse, originating from the right pressure source, propagates towards the pipe segment with reduced wave propagation velocity. As the pressure pulse reaches the border between the different impedances, it generates one transmitted and one reflected wave. The transmitted wave will always be in phase with the incoming wave, while the reflected wave will either be in phase or 180° out of phase, depending on the impedance zones.

Figure 6.6 (a) shows the incoming wave approaching the pipe segment with lower velocity, with $\rho_2 a_2 < \rho_1 a_1$. According to the reflection coefficient, Equation 2.10, the reflected wave will be 180° out of phase. Figure 6.6 (e) illustrates how the reflected wave is positive for the case where $\rho_3 a_3 > \rho_2 a_2$ at the left border of the region of lower impedance. It can be observed that two pressure pulses are propagating between the pressure source and the region of reduced impedance.



Figure 6.6: Propagation of a pressure pulse in a pipe with regions of different impedances

6.2.3 Experiment

The Flowmaster simulations of the experimental setup can be seen in Figure 4.3 in Chapter 4. As the pressure was measured 6,7 m from the gate valve, values from the corresponding location in the modeled pipe were used.

The results from a simulation without air injection is shown in Figure 6.7(a). It can be observed that the pressure amplitudes are higher than the measured values. The period of the oscillations, accoring to the equation $T = \frac{4L}{A}$, was approximately 0.15 s, while the period of the simulated oscillations was 0.12 s.

Figure 6.7(b) shows the pressure oscillation with a pipe length of 2,26 m of reduced wave propagation velocity placed approximately 18,5 m from the valve. The length of the short pipe section corresponds to one second of air injection. The same tendecies as in Figure 6.5 can be seen, but the pressure peaks are not as evident in this case. The period of the oscillations is reduced, but the highest pressure peak are further apart. The cases for air injection of two and four seconds gave similar results, and are thus not presented.

Furthermore, the model of the laboratory setup was tested without the pipe with a blank end to check the influence from the ramifications. The results from the two simulations were almost identical, and the presence of the blank ended pipe did not appear to affect the results from the simulations significatly.



(a) Pressure oscillations with ordinary pipes



(b) Pressure oscillations with pipe segment of reduced propagation velocity, corresponding to one second of air injection

Figure 6.7: Results from simulation of laboratory setup with and without air injection

Chapter 7

Discussion

7.1 Experimental results

The results from the experiment showed that the pressure oscillations were significantly reduced by the air injection. Figure 6.1 shows the pressure oscillations without air injection. The distinct pressure increase as the valve closes is due to the deceleration of the water masses, while the pressure oscillations are due to the compression of the water close to the valve. The mean pressure in the pipe with steady state conditions was approximately 173 kPa of absolute pressure, caused by a pressure head of 9 m from the system. The pressure loss is due to friction losses in the bends and pipes. The pressure transient arise from the very last part of the closure, which is faster than the reflection time of the system. The period of the pressure oscillations was 0,153 seconds, which corresponds well with the estimated period of 0,147 seconds.

Figure 6.1 shows the pressure oscillations for one and three seconds of air injection. Because of the reduced wave propagation velocity and the increased damping due to the compressibility effects of the bubbles, the pressure oscillations are reduced. The amplitude was reduced and the period of the oscillation was increased. The amplitude of the oscillations with three seconds of air injection was somewhat higher than the corresponding amplitude for one second of air injection. This could be due to uncertainty in the measurements, and further experiments would clarify this.

In order to investigate the effects of injecting air immediately after the closure of the gate valve, several attempts were made to time the closure of the gate valve so that the air was injected right after the closure. This proved to be challenging with the existing solutions for the LabView program, hence half of the results presented in Table 6.1 are from cases where the valve was closed too late and air was injected before or during the closure. Ideally, the closure process in Figure 6.3(a) should have been started earlier, but it is still a good illustration of the effects of air injection.

The comparison shown in Figure 6.3(b) illustrates the effects of the air injection, where the amplitude was reduced immediately after the air injection and the oscillation dampen out quickly. The same case was also tested for air injections of two and four seconds, which were presented in Table 6.1. One second of air injection seemed to have the most influence on the amplitude of the pressure oscillations, while the period was decreased significantly for the cases with air injection of four seconds. Results from the other flow rates showed the same tendencies.

The pressure increase as the valve closes is apparent in all the figures, but the figures show a variation of the pressure peak. The different pressure scenarios could be caused by the uncertainty of initial valve position and the closure speed of the valve, which are both manually adjusted. It was found difficult to repeat the tests as the manual adjustments were quite uncertain.

The overall results from the pilot study proved that an air injection system is possible, that it has beneficial effects on pressure transients and that injection setup worked relatively well. The actual flow rate of air proved to be uncertain because of the long air hose from the valve to the pipe inlet, and because water accumulated in the air hose. Water in the air hose could be avoided by placing check valves at the pipe inlets.

7.2 Flowmaster simulations

It can be seen that the placement directly in front of the valve was beneficial in terms of a reduced amplitude and an increased period. On the other hand, placements in the middle of the pipe section lead to an increased amplitude and reduced period, which worsen the effects from the pressure transient. This is an effect from the impedance difference, which is also demonstrated in the results from the pressure pulse propagation presented in the following section. This will be further discussed in the following chapter.

Although the experimental results indicated that the presence of air can have beneficial effects on the pressure transient, the results from simulations in Flowmaster were somewhat divergent. The simulations were accurate for the rapid
7.2. FLOWMASTER SIMULATIONS

valve closure, and a modified pipe with a short region of reduced wave propagation velocity directly in front of the valve proved to have beneficial effects on the pressure oscillations, as seen in Figure 6.4. However, the effects were only caused by the reduced wave propagation velocity, as the Flowmaster model was unable to account for the increased damping because of the compression-expansion cycle of the bubbles. The pressure amplitudes were reduced equally in all three cases, because of friction in the pipe.

For the cases with an increased closure time, the results from the Flowmaster model did not correspond well with the estimated pressure rise from Joukowski's equation for $T_C > T_R$. The pressure in front of the valve was overestimated compared to Joukowski's equation for a number of different closure times. This could be due to the valve characteristics of the gate valve in Flowmaster, as mentioned in Section 4.4, where the valve loss data for the last 10 % of the closure is uncertain. Valve characteristics will be discussed further in this section.

The Flowmaster models provided good results for cases where the region of reduced propagation velocity was placed next to the valve, but this was not the case when the modified region was placed away from the valve. From the results presented in Table 6.3, it can be seen that the maximum pressure in front of the valve is increased for the cases where the modified pipe was placed away from the valve. Additionally, the period was decreased to less than half of the period for the ordinary pipe, which results in a worsening of the pressure transient.

This can be explained by the propagation of a pressure pulse, as shown in Figure 6.6. The impedance difference causes a reflected and a transmitted wave. The reflected wave is 180° out of phase and propagates back to the valve. When the region of reduced impedance is placed away from the valve, the distance of travel for the pressure pulse is reduced to the distance to the pipe with reduced impedance, and thus the period is decreased.

A simulation of a pipe with a reduced wave propagation velocity is shown in Figure 6.5. The pressure oscillations were not as regular as the oscillations from an ordinary pipe, and even though the period was reduced it is further between the highest pressure peaks because of quicker oscillations with smaller amplitude. The pressure peaks are significantly higher than the maximum pressure for the ordinary pipe, and the beneficial effects of reduced wave propagation velocity are not present.

Results from the Flowmaster simulations of the experimental setup showed the same tendencies as the previous simulations. An initial valve opening of 0,77 was used in the Flowmaster simulation in order to obtain a similar flow rate as the experiments, due to the loss coefficient of the gate valve. The valve was closed with $T_C > T_R$, and a higher pressure increase was found from the simulations

than from the measured values, as with the previous Flowmaster simulations. Figure 6.2.3 show the pressure oscillations for simulations with no air injection and one second of air injection. The pressure increase becuase of the deceleration of the water masses as the valve closes could also be seen here, and the principles are similar in the experimental and simulated results. The maximum pressure of the case with air injection is lower than the case of no air injection, but as in the previous simulations the oscillation is quick and irregular.

Additionally, the pressure increase was overestimated in the simulation of the experimental setup. Where the pressure increase in the laboratory was approximately 170 kPa, the simulations resulted in a pressure increase of nearly 2000 kPa. As previously mentioned, this could be due to the valve characteristic of the gate valve in Flowmaster. For improvements of the Flowmaster model, the valve characteristic of the gate valve in the laboratory should be found experimentally and implemented in the Flowmaster model.

Chapter 8

Conclusion

The purpose of this thesis was to investigate the possibilites of air injection to reduce pressure transients in piping systems. The initial project definition included a theoretical study of gas injection and the actuation process of the system, along with simulations of analytical models in Flowmaster. However, in agreement with supervisor Morten Kjeldsen, the main focus of the thesis has been shifted in a more experimental direction, as it was found more beneficial for the project to investigate the experimental solutions and challenges at an early stage of the development.

The experimental results showed that the presence of air had positive effects on the amplitude and period of a pressure transient, and the possibilities of an air injection system for control and reduction of pressure transients should be futher investigated. The experimental setup worked well, and the chosen components can be included in a future experiment. The main challenges in the simplified experiment were the uncertain flow rate of air and the duration of air injection, due to water in the air hose. Additionally, the timing of the gate valve closure in connection with the air injection proved to be difficult. Before an advanced experiment is carried out, a further development of the simplified experiment and a parameter study should be completed.

The simulations of the computational models in Flowmaster proved to be insufficient in terms of modelling the increased damping due to the compressibility effects of the air bubbles, and also for the cases where different locations for the air injection were demonstrated. Thus, along with a comprehensive parameter study and improvements of the existing setup in the laboratory, further development of a computational model is decisive for the future development of the air injection system. The air injection system should be investigated thoroughly in a computational model in for instance LabView or Matlab, where the effects of air injection, damping due to the copressibility effects of the bubbles and the transportation of bubbles down stream is included.

Chapter 9

Further work

Suggestions for the advanced experiment include a setup for investigating effects of bubble size and ensure a favourable bubble size distribution, in addition to an extensive measurements system to monitor the pressure development and activate the air injection system if necessary. A further study of the parameters involved in the simplified experiment should be carried out prior to the advanced experiment.

Some changes and refinement are needed on the existing setup in order to complete the parameter study. This includes a check valve mounted on the pipe inlet to avvoid water in the air hose, and a reduction of the hose length between the throttle valve and the pipe inlet. With a procedure where the air flow rate regulated by the throttle valve is calibrated with the flowmeter prior to the experiment, the length of the air hoses could be held to a minimum. Additionally, the gate valve should be controlled automatically through the LabView program in order to obtain a more accurate closure process. This could be achieved by an electric valve that regulates the oil supply to the hydraulic aggregate controlling the gate valve.

Sevaral parameters could be investigated after the refinement of the simplified experiment. The parameter study should include:

- Flow rate of injected air
- Time lag of air injection after gate valve closure
- Number of injection points
- Location of air injection

• Pressure of injected air

Further tests could also be performed with varying duration of air injection and closure time of the gate valve. Emphasis should be given to reduce the uncertainties in the experiment, and calibration of all the measurement equipment should be carried out.

As the Flowmaster simulations proved to be insufficient in terms of modelling the presence of air in the pipe, one has the option of developing a new program based on the method of characteristics in LabView or Matlab. The air injection system should include the effects of increased damping due to the compressibility effects of the bubbles in addition to the transportation of bubbles downstream, which the Flowmaster simulations were unable to take into account. If Flowmaster are to be used further, it should be considered to measure the valve characteristics and implement it in the Flowmaster model. This could give better results for the cases with increased closure time.

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Appendices

Appendix A

Method of Characteristics

The Method of Characteristics (MOC) is a numerical method which transfers partial differential equations with no general solution into total differential equations [15]. This approach is widely used in calculations and analysis of system dynamics, like pressure pulsations and the water hammer phenomenon. The governing equations for any change in pipe flow are the equations of continuity and motion, Equations 2.1 and 2.2:

The partial differential equations of continuity and momentum, Equations 2.1 and 2.2, are transformed into ordinary differential equations and combined linearly by the use of an unknown multiplier [15]. After some alterations, the characteristic equations are defined as:

$$C^{+}:\begin{cases} \frac{g}{a}\frac{dH}{dt} + \frac{dV}{dt} + \frac{V|V|}{2D} = 0\\ \frac{dx}{dt} = +a \end{cases}$$
(A.1)

$$C^{-}: \begin{cases} -\frac{g}{a}\frac{dH}{dt} + \frac{dV}{dt} + \frac{V|V|}{2D} = 0\\ \frac{dx}{dt} = -a \end{cases}$$
(A.2)

The characteristic lines, $\pm a$, are straight lines in the xt plane where Equations A.1 and A.2, the compatibility equations, are valid. The system is solved numerically, and a computational grid for the numerical solution can be seen in Figure A.1. The independent variables V and H are known at the points A and B at t = 0, thus the compatibility equations A.1 and A.2 that are valid along the characteristic lines can be integrated with point P as a upper limit. This results

in two equations with two unknowns for point P, which provide the conditions at P [15].



Figure A.1: Calculation grid in the xt plane

An integration of the compatibility equations along the characteristic lines yields two equations that define the pressure head and flow in a pipeline:

$$C^{+}: H_{P} = H_{A} - B(Q_{P} - QA) - RQ_{P} |Q_{A}|$$
(A.3)

$$C^{+}: H_{P} = H_{B} - B(Q_{P} - QB) - RQ_{P} |Q_{B}|$$
(A.4)

B is defined as the pipeline characteristic impedance and is given as $B = \frac{a}{gA}$. *R* is the pipeline resistance coefficient and is given as $R = \frac{f\Delta x}{2qDA^2}$.

The computational grid shown in Figure A.1 and the equations above are ideal for a computer program in Matlab or Fortran, for instance, where different boundary conditions can be applied. Some of the basic boundary conditions are a reservoir, pump or a valve. The commercial software Flowmaster is also based on the method of characteristics, but does not require any programming. A more detailed introduction to the method of characteristics is given in Fluid Transients in Systems by Wylie and Streeter [15].

Appendix B

LabView program

The measured values were logged with a LabView program made by Håkon H. Francke from Flow Design Bureau (FDB). The front panel of the program can be seen in Figure B.1 below.

Håkon Hjort Francke		
F.D.B Bureau		AQ OK LVE CLOSED
RECORD CONTROL RECORD	Save to folder: C:Documents and Settings: table Settop:New Folder Filename Var Injection Testing q=181 4s	AL AIR NUAL JECTION
SETUP MAIN		
XY Graph 0 2500 40 255 25 20 25 20 5 20 5 40 5 40 5	500 750 1000 1500 1500 000 220 2500	Air 2000
	Real Time	

Figure B.1: Front panel of LabView program

The input values are the duration of logging the measurement values, and start

and stop time of the air injection. Additionally, the air injection can be turned on and off in order to measure pressure transients both with and without air injection.

Appendix C

Complementary results

C.1 Ideal air injection procedure

Figure C.1 is the result from a test with a flow rate of 160 l/s, where the gate valve is closed right before the air injection starts. As the actuation system would have the same procedure, it would be desirable to obtain a closure and air injection process like this.



Figure C.1: Closure of the gate valve followed by one second of air injection

C.2 Flow rate in the rotameter

In order to find the real flowrate of air, an equation which takes the pressure difference into account was used. The equation can be seen below, where Q is the actual volumetric flow rate, Q_{meas} is the measured flow rate, p_1 is the standard pressure, p_{in} is the incoming pressure and p_{pipe} is the pipe pressure.

$$Q = Q_{meas} \frac{\sqrt{p_1 p_{in}}}{p_{pipe}} \tag{C.1}$$

Appendix D

Additional Flowmaster models

D.1 Air vessel

Figure D.1 shows a Flowmaster model which demonstrates the use of an air vessel upstream a rapidly closing valve to reduce a pressure trasient. The air vessel will transform the rapid movement into a slower surge between the upper reservoir and the air vessel.



Figure D.1: Flowmaster model of a pipeline with an air vessel to reduce pressure transients



Figure D.2: Pressure variation with and with air vessel

The model was simulated with and without the air vessel, and the results can be seen in Figure D.2. It is clear that the pressure in front of the valve was significantly reduced by the air vessel.