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Planning, start-up and testing of a pipe flow loop for the investigation of transient characteristics

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Abstract

This thesis concerns the design, planning, initial testing and evaluation of a pipe flow loop. The pipe flow loop will later be used in experiments on fluid transients, meaning that the design must be suitable for this purpose. The design of the flow loop is mainly based on simulations in LVTrans and FloMASTER. The design was completed and tested in May 2017 in combination with some simple experiments. The experimental setup mainly worked as expected and the initial testing was completed with satisfying results. Experience from the initial testing is the foundation for experimental procedures developed for further experimental work in the flow loop. Analysis of the results obtained from the initial testing has revealed some deficiencies in the experimental setup. The deficiencies have been identified, and measures for improving the flow loop has been suggested.

Sammendrag

Denne masteroppgaven dreier seg om design, planlegging, innledende testing og evaluering av en rørsløyfe. Denne rørsløyfen skal brukes til videre eksperimentelt arbeid på transiente rørstrømninger, og må dermed være designet for dette. Designet er hovedsaklig basert på simuleringer i programmene LVTrans og FloMASTER. Designet ble ferdigstilt og testet i mai 2017 i kombinasjon med noen enkle eksperimenter. Denne innledende testingen gikk i grove trekk bra. Erfaringer fra testingen har lagt grunnlaget for utviklingen av eksperimentelle prosedyrer for bruk av rørsløyfen i senere eksperimenter. Analyse av resultatene har avslørt noen mangler og svakheter i det eksperimentelle oppsetettet. Disse manglene og svakhetene er påpekt og tiltak for å rette opp i disse har blitt foreslått.

Acknowledgment

First and foremost, I would like to thank my supervisors Morten Kjeldsen and Bjørnar Svingen for their thorough guidance during the work with this master thesis. I would also like to thank Ingrid Vilberg and Carl Bergan for their help with the measurements. I am also grateful to Olav Olsen from Torsion Tool Company for his help with all practical challenges related to the experiments, and for making sure that the experiment worked in fulfilment with all requirements. Finally, thanks to IRIS - The International Research Institute of Stavanger for facilitating a safe and pleasant working environment during the initial testing of the flow loop.

EMH

(Eirik Myrvold Hansen)

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Nomenclature

ϵ	Relative pipe roughness [-]
λ	Friction coefficient [-]
μ	Poisson ratio of steel [-]
a	Wave propagation speed or speed of sound [m/s]
A_D	Amplitude of disc
A_{outlet}	Area of slit outlet in rotating valve [m^2]
c_1	Correction coefficient [-]
D	Internal pipe diameter [m]
E	Elastic modulus of pipe material [Pa]
e	Pipe wall thickness [m]
f	Frequency [Hz]
f_L	Pipe friction factor [-]
g	Gravitational acceleration [m^2/s]
H	Piezometric/Hydraulic head [m]
h_L	Head loss [m]
K	Bulk modulus of elasticity of fluid [Pa]
K_L	Loss coefficient [-]
n	Number of superimposed sines [-]
Q	Flow rate [m^3/s]
r	Radius of disc [m]
r_0	Average radius of disc [m]
Re	Reynold number [-]

rho	Density of water $[kg/m^3]$
t	Time [s]
T_M	Transmission constant of motor [-]
T_N	Natural period of pipe [s]
T_R	Reflection time [s]
T_{flush}	Flushing time [min]
v	Fluid velocity
Z	Characteristic specific acoustic impedance $[kg/m^2s]$

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Institutt for energi- og prosessteknikk

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MASTEROPPGAVE

for

Student Eirik M. Hansen

Våren 2017

Planlegging, oppstart og test på rørsløyfe for undersøkelse av transiente strømningskarakteristikker

Planning, start-up and testing on a pipe flow loop for the investigation of transient characteristics

Bakgrunn og målsetting

En god forståelse av transiente strømninger i rør og tunnel-systemer er helt sentralt for å oppnå sikker drift av vannkraftverk. Påførte transienter brukes også aktivt for å oppnå kunnskap, blant annet ved hjelp av såkalte frekvens-respons-målinger. Sistnevnte metode brukes for vurderinger av reguleringsstabilitet, og for optimalisering av kontrollsystemer.

NTNU Vannkraftlaboratoriet har hatt aktivt forskning på transient rørstrømning, og spesielt nevnes phd arbeidene til Bjørnar Svingen, Li Ping Ju og Roar Vennatrø, og Hermod Brekke og Abdel Rhich. De tre førstnevnte på transient friksjon, og de to sistnevnte på frekvens respons målinger. Det er også verdt å nevne arbeidene på selveksiterende svingninger, bl.a. ved Bjørnar Svingen sin hovedoppgave i 1991.

Denne Masteroppgaven er også knyttet til et pågående PhD arbeid ved Ingrid K Vilberg som fokuserer sitt arbeid på effekt av gass på transienter i vannkraftsamarbeid.

Kostnadene knyttet til fysisk testing og tilvirkning av komponenter er støttet av FDB, Forskningsrådet og RFF Vest.

Oppgaven bearbeides ut fra følgende punkter

1 Litteraturstudium. Det legges vekt på å sette dette arbeidet i kontekst med tidligere arbeider ved Vannkraftlab, bl.a. med nevnte phd oppgaver, og i vannkraftsammenheng.
2 Modellering av IRIS sin strømningssløyfe og vurdering av forskjellige pumpeløsninger.
3 Designinnspill og assistanse på følgende komponenter.
- Returkopling (høydeknekk)
- Element for generering av transiente (oscillatorisk) strømning.

4 Designinnspill på målesystem/ analyse.
5 Gjennomføring av innledende tester.
6 Det forutgående prosjektarbeid og framtidig arbeid i denne master skal beskrives i et faginnlegg,"paper", som skal presenteres ved 7th International Symposium on Current Research in Hydraulic Turbines (CRHT-VII) at Katmandu University, april 2017.
7 Skrive en rapport

Oppgavens fysiske gjennomføring (punkt 5) skjer ved IRIS i Stavanger. Kontaktperson ved IRIS formidles ved FDB. Tilvirkning av komponenter (punkt 3) gjøres av Torsion Tool Company og i regi av FDB.

" - "

Senest 14 dager etter utlevering av oppgaven skal kandidaten levere/sende instituttet en detaljert fremdrift- og eventuelt forsøksplan for oppgaven til evaluering og eventuelt diskusjon med faglig ansvarlig/veiledere. Detaljer ved eventuell utførelse av dataprogrammer skal avtales nærmere i samråd med faglig ansvarlig.

Besvarelsen redigeres mest mulig som en forskningsrapport med et sammendrag både på norsk og engelsk, konklusjon, litteraturliste, innholdsfortegnelse etc. Ved utarbeidelsen av teksten skal kandidaten legge vekt på å gjøre teksten oversiktlig og velskrevet. Med henblikk på lesning av besvarelsen er det viktig at de nødvendige henvisninger for korresponderende steder i tekst, tabeller og figurer anføres på begge steder. Ved bedømmelsen legges det stor vekt på at resultatene er grundig bearbeidet, at de oppstilles tabellarisk og/eller grafisk på en oversiktlig måte, og at de er diskutert utførlig.

Alle benyttede kilder, også muntlige opplysninger, skal oppgis på fullstendig måte. For tidsskrifter og bøker oppgis forfatter, tittel, årgang, sidetall og eventuelt figurnummer.

Det forutsettes at kandidaten tar initiativ til og holder nødvendig kontakt med faglærer og veileder(e). Kandidaten skal rette seg etter de reglementer og retningslinjer som gjelder ved alle (andre) fagmiljøer som kandidaten har kontakt med gjennom sin utførelse av oppgaven, samt etter eventuelle pålegg fra Institutt for energi- og prosessteknikk.

Risikovurdering av kandidatens arbeid skal gjennomføres i henhold til instituttets prosedyrer. Risikovurderingen skal dokumenteres og inngå som del av besvarelsen. Hendelser relatert til kandidatens arbeid med uheldig innvirkning på helse, miljø eller sikkerhet, skal dokumenteres og inngå som en del av besvarelsen. Hvis dokumentasjonen på risikovurderingen utgjør veldig mange sider, leveres den fulle versjonen elektronisk til veileder og et utdrag inkluderes i besvarelsen.

I henhold til "Utfyllende regler til studieforskriften for teknologistudiet/sivilingeniørstudiet" ved NTNU § 20, forbeholder instituttet seg retten til å benytte alle resultater og data til undervisnings- og forskningsformål, samt til fremtidige publikasjoner.

Besvarelsen leveres digitalt i DAIM. Et faglig sammendrag med oppgavens tittel, kandidatens navn, veileders navn, årstall, instituttnavn, og NTNUs logo og navn, leveres til instituttet som en separat pdf-fil. Etter avtale leveres besvarelse og evt. annet materiale til veileder i digitalt format.

☐ Arbeid i laboratorium (vannkraftlaboratoriet, strømningsteknisk, varmeteknisk)
☒ Feltarbeid

NTNU, Institutt for energi- og prosessteknikk, 15. januar 2017

Bjørnar Svingen
Faglig ansvarlig/veileder

Medveileder(e):
Morten Kjeldsen, Daglig leder, Flow Design Bureau AS.

Chapter 1 Introduction

A good understanding of transient flows in pipes and ducts is essential to ensure safe operation of hydropower plants. Forced transients can be used actively to improve this understanding through frequency response measurements. Flow Design Bureau AS will facilitate experiments involving such transients in a flow loop located outside of Stavanger, Norway. This flow loop is the property of IRIS – The International Research Institute in Stavanger. It is about 1400m long, yielding ideal conditions for large-scale experiments. A satellite photo of the entire test facility is provided in figure 1.1. Initial testing of the flow loop will be carried out in May 2017.

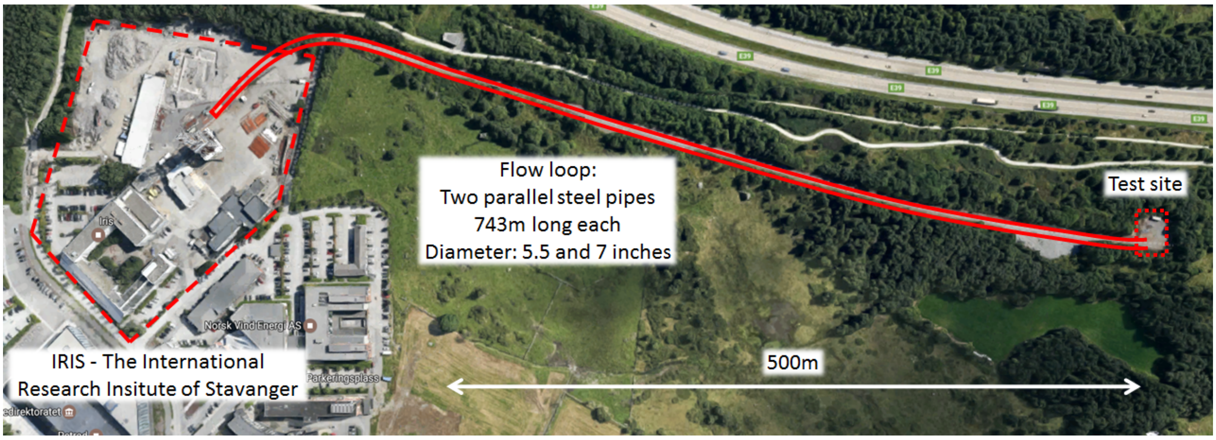


Figure 1.1: Satellite photo of the test facility

There are two different aspects of transient flows that will be subject to further investigation in this flow loop, namely transient friction and the effect of free gas on transients. The experiments on transient friction will be done by Bjørnar Svingen to supplement the results from similar experiments he has conducted in the past. Many good models exist for transient friction in pipes, but experimental data on this scale is scarce. Given the length of the pipes, this flow loop is ideal for providing such valuable experimental data. PhD candidate Ingrid Vilberg will do the experiments on free gas. The presence of free gas will lower the wave propagation speed and have a dampening effect on the transients. In her experiments, she will investigate the active use of this effect by purposely injecting air into the system.

This project is funded by Flow Design Bureau, RFF Vest and The Research Council of Norway. Torsion Tool Company, represented by Olav Olsen, is responsible for the manufacturing of the necessary components for this experiment.

1.1 Problem

To ensure that the described experiments can be successfully executed it is necessary to design a versatile flow loop that can generate different types of transient flows. The goal of this master thesis is to design such an experiment, perform initial testing and evaluate the performance of the experimental setup. To achieve this, a few more components must be designed or selected, such as pumps, accumulators and valves. To ensure that the chosen components yield the required system response, a model of the flow loop will be created in a 1D simulation software for transient pipe flows (such as FloMASTER or LVTrans).

The included formal problem description has been worked out in a very early phase of this project. This means that not all problems and challenges were known at the time this was developed, but have emerged as the project has progressed. This situation resulted in a very general and incomplete formal problem description that needs further specification and some additional points. All the work done in this thesis has been in compliance with the initiator of this project, namely FDB AS.

The work done in this thesis includes the following points.

1. Literature study. This work should be put in context with other work done at the Waterpower Laboratory at NTNU, and in a hydro power context in general.

2. Modelling of the flow loop and consideration of pump solutions.

3. Design input on the following components.

- Pipe connection.
- Component for generation of oscillatory flow.

4. Design input on measurement system and analysis. The analysis should be used to reveal the success of the various components and the established experimental procedures.

5. Carry out initial testing of the flow loop. This initial testing should be used to develop procedures for preparation of the flow loop and experimental procedures.

6. Describe the previous work done in project thesis, and the future work of this master thesis in a paper that should be presented at the 7th international symposium on current research on hydraulic turbines at Kathmandu University in April 2017.

The candidate will be the person with the deepest and longest lasting involvement in this project, at least during the work on this thesis. This means that the candidate will be given the role as project manager during these months.

1.2 Literature

It is important to clarify how this ongoing project can be of any scientific importance. To do that, it is necessary to provide a context in the form of previous studies. More precisely, it must be investigated in what ways this project relates to past projects, particularly in the Waterpower Laboratory at NTNU.

Some work has been done on transients in pipes at the water power laboratory in the previous years. These studies may be divided into two categories, namely theoretical and experimental. Some studies are also a combination of the two, where an experiment is used to validate a model, simulation or a computer program. Some studies also make great use of frequency response measurements to assert the dynamic behaviour of a system.

1.2.1 Transient friction

In a transient flow, the friction loss will have an unsteady component. The friction loss is governed by a lot of different complex aspects of the flow that are not yet fully understood. This includes e.g. the behaviour of turbulent structures in the flow when subjected to a transient pressure wave [3]. Whenever such complexity is present, it necessary to introduce a model. The complexity can usually be modelled in many different ways, but high accuracy models are in general computationally demanding. A substantial amount of research has been done in this field, a lot of which in the last two-three decades[12]. This includes much theoretical work on different models, as well as experimental work for validation of the models. Several projects concerning this subject have also been conducted at the Waterpower Laboratory at NTNU.

The current status of the research on transient friction can be summed up as follows. Modelling of the transient friction is typically achieved by a 1D model. This means that averaged values of variables such as velocities are used at a cross section. A model is considered to be successful if it is both accurate and simple. A model is not simple enough if it requires vast amounts of computational power to acquire accurate results. The existing models are decent for some flow phenomenon, such as the classical case of the transient flow following an immediate valve closure. For other situations including e.g. surge, the models are less successful. The models also become computationally demanding when more complex systems are considered. In short, this means that more work needs to be done on modelling. It is necessary to provide experiments for validation of the models, which is how this project is important. As rapid changes in the operation of hydropower plants becomes more common, transients in pipes and ducts are also becoming more common. This means that good models for such transients are an instrumental part of both the design and operation of hydropower plants [4].

Being such a vast field of study, it is impossible to enlight the reader on all the research going on in the area of transient friction in pipes. Hence, the goal of the following section is to put this experiment in context with the work of some selected scientists, especially at NTNU.

Bjørnar Svingen

Svingen's thesis [13] investigates fluid-structure interactions(FSI) in piping systems. Svingen created a computer program based on the FEM discretization and validated the program by experiments. To investigate such interactions, the flow must be excited in some way. This is done utilising a sine shaped disc valve. Svingen's excitation mechanism ensures that the system is excited with a sine function.

The thesis is relevant to the field of fluid transients because FSI is considered a dynamic phenomenon. The problem is that the models used for transients in pipes often consider the solid uncoupled from the fluid. This is often not the case. One of the goals of the thesis is to establish this coupling. Given that the pipes are thick and can withstand pressures up to 300bar, the effects of FSI might be small in the flow loop in Stavanger. Svingen will be involved in experiments in this test facility at a later time, but his main focus will be on transient friction.

Svingen has also looked more directly at transient friction. He has proposed an explicit model for the friction losses in transient or oscillatory pipe flow, including the Rayleigh damping.

$$h_{loss} = f\frac{Q|Q|}{2DA^2} - \frac{\lambda_f}{\rho A}\frac{\partial^2 Q}{\partial x^2} \tag{1.1}$$

where λ_f is the frequency dependent friction factor.

Pål Tore Storli

In his thesis, Storli also investigates transient friction. His objective was to "to try to find a simple but accurate representation of the frictional losses in transient flow in pressurised pipes" [12]. In his work, Storli developed a correctional model for a particular case, concerning the flow that follows a sudden valve closure in a single pipeline at a low Reynolds number. This correctional model is an example of a model that could be validated by experiments in the flow loop. This will probably not be done but exemplifies how the future experiments can be related to previous work done at the Waterpower Laboratory.

Pingju Li

Pingju Li [9] has also carried out experimental work on oscillatory flow in conduits. His work was related to transient friction in non-circular ducts, both with and without added roughness. Li excited the flow with the pump. He also investigated the velocity profile by laser measurements, providing valuable experimental data for validation of 2D models in particular. The importance of Li's work sorts in the same category as the future experiments, namely that it provides data for validation of models. However, Li's experiments are very different from the one in question, regarding both scale and measured data. This means that Li provides validation for other types of models or simulations than that of the upcoming experiment.

Roar Vennatrø

Vennatrø [17] did somewhat similar experiments as Li at the Waterpower Laboratory in 2000. His work was similar in the sense that it was experimental work on transient flow in pipes, with a focus on measuring the velocity profile. His measurements, however, were performed in smooth circular pipes. His work revolved around both oscillatory and water hammer types of flow. Vennatrø also developed a computer program that he matched with the results of his experiments.

Ove Bratland

A less obvious contribution is that of Ove Bratland (Waterpower Laboratory, 1985). He investigated a principle for valve activation by hydraulic signals through hoses. Even though transient friction was not the main focus, Bratland developed a new simulation model for laminar transient pipe flow that also accounts for the frequency dependence. Bratland validated his simulation model with experiments conducted by Holboe and Roleau (1967), proving that the availability of good experimental data is important.

Erik Brodin

In 1998, inspired by Bratland, Brodin finished his experimental work on pulsatile flow in pipes with an unsteady component. He aimed to create a damping model, suitable for handling turbulence. His experimental setup consisted of 468m long flow loop. This is certainly long, but the pipe diameter was only 12mm. This lead to almost complete damping of resonance peaks in the system, showing the importance of having larger diameter pipes.

Current research at the Waterpower Laboratory, NTNU

At the moment of writing this thesis, another test rig for investigation of the transient friction phenomenon is under construction at NTNU. This test rig will focus on the transient friction component related to the surge in pipes or conduits of hydropower plants.

1.2.2 Frequency response measurements

Frequency response measurements quantify the output frequency spectrum of a system as a response to some stimulus. According to Balchen et al. [7], this is a great way to assert the general system dynamics. Such an analysis yields the eigenvalues and where in the system maximum amplitudes occur. This information will be very valuable concerning many aspects in the setup of the experiment. Due to the power of frequency response measurements it has been utilised in many scientific studies. Here, some studies related to hydropower are presented.

Hermod Brekke

In his thesis [1], Brekke developed a damping model for unsteady turbulent flow. This was a part of a more comprehensive structure matrix method. This method is a general frequency analysis model for piping systems such as hydropower plants. The main motivation behind his work was problems with stability in various Norwegian hydropower plants. In the developed model, Brekke involved the friction loss in the pipes as a complex parameter $K(i\omega)$, including both the steady and the unsteady model. His method was validated by extensive experimental work with frequency response measurements at six different hydropower plants in Norway.

Abdel-ilah Rhrich

Abdel-ilah Rhrich did his thesis with the title "Stability and transient performance studies of governed hydro turbine systems" in 2008 at the Waterpower laboratory at NTNU. In his thesis, Rhrich focused on modelling techniques of governed hydro turbine systems used for stability and transient performance studies. For the transient performance studies, the dynamic flow was modelled based on both water hammer and rigid column theory. Rhrich also establishes the application range for each of the models. Rhrich performed frequency response measurements at Tonstad power plant as a part of his thesis. This was done in order to "map the dynamic characteristics of the complex water conduit system to extract information about the interaction between upstream surge tanks" [6, p. 90].

Chapter 2 Theory

2.1 Flow classifications

It is necessary to define some of the terms used to characterise a flow. Wylie/Streeter [18, page 1], divides the flow regime into *steady* and *unsteady flow*. In an unsteady flow, the flow variables may change in time at a certain location. For flows in pipes, the term *water hammer* can be used to describe the unsteady flow preceding a sudden change in fluid motion. It is also necessary to divide between *uniform* and *nonuniform* flow. In a nonuniform flow, the average velocity might change along the pipe length. The term water hammer is typically used for the nonuniform flow, while the term *surge* is commonly utilised for the uniform flow. If the flow conditions are repeated in every period, the flow is termed *steady-oscillatory*, *pulsatile* or *periodic*. If the amplitudes of an oscillatory flow build up and amplify, the system is in *resonance*. The resonance indicates an energy storage in the system. This occurs at the natural periods of the system (fundamental or harmonic).

The oscillatory flow can be described by equation 2.1.

$$Q = Q_0 + \Delta Q sin(2\pi f t) \tag{2.1}$$

where Q_0 is the average flow, ΔQ is the amplitude in the flow, f is the frequency of oscillation and t is time.

2.2 Differential equations

All the flows considered in this thesis is confined within a pipe and governed by the continuity equation and the equation of motion [10, page 2].

$$\frac{\partial H}{\partial t} + \frac{a^2}{g}\frac{\partial v}{\partial x} = 0 \tag{2.2}$$

$$g\frac{\partial H}{\partial x} - \frac{\partial v}{\partial t} + \lambda\frac{v|v|}{2D} = 0 \tag{2.3}$$

where H is the piezometric head, a is the wave propagation speed, v is the fluid velocity and λ is a friction coefficient. These equations can be solved by the Method of Characteristics. This a widely used method that transfers the partial differential equations into ordinary differential equations that are much easier to handle numerically. Simulations in both LVTrans and FloMASTER are based on the solution of the governing equations by this method [10, page 17].

2.3 System parameters

It is necessary to calculate some of the system parameters for the flow loop. These parameters will be used in simulations.

Wave propagation speed

The wave propagation speed is important for the dynamics in the flow loop. It is given by the equation 2.4 [18, page 6]. This equation is valid for pipes with expansion joints throughout, yielding negligible axial unit stress.

$$a = \sqrt{\frac{K/\rho}{1 + (K/E)(D/e)c_1}} \tag{2.4}$$

where K is the bulk modulus of elasticity of the liquid, E is the modulus of elasticity of the pipe material, e is the wall thickness, ρ is the fluid density, and D is the internal diameter of the pipe. The coefficient c_1 is given by equation 2.5. This coefficient is valid for pipes with thick walls that can expand freely. Thick walls imply a diameter to wall thickness ratio smaller than 25. [18, p. 23]

$$c_1 = \frac{2e}{D}(1 + \mu) + \frac{D}{D + e} \tag{2.5}$$

where μ is the Poisson ratio of the pipe material.

Natural period

The natural period, T_N is an important parameter of the piping system, both during design and operation. It is given as the time it takes for a pressure wave to propagate two times back and forth trough the system, yielding the intuitive formula for the pipeline period as seen in equation 2.6 [15, page 10]. The reflection time, T_R, is given as half the natural period of the pipe.

$$T_N = \sum_{n=1}^{N_P} \frac{4L}{a} \tag{2.6}$$

where N_P is the number of pipe segments, L is the length of each pipe segment, and a is the wave propagation speed.

The magnitude of a pressure change generated by a 'rapid event' is dependent on the reflection time. If an event, such as valve closure or a pump trip, occurs slower than time pipeline period, the full Jukowsky pressure will not be developed. This is because the reflected pressure wave modifies the pressure before the full potential pressure change generated by the event is imposed on the system. Hence, the criteria for valve closure time is determined by 2.3

- If $T_C \gg T_R$ amplitudes are dampened.
- If $T_C < T_R$ amplitudes can reach Joukowsky pressure.

2.4 Waves

Two different kinds of waves will be present in this system, and some understanding of these waves is necessary to understand the system behaviour. The waves can be divided into travelling and standing waves.

Travelling waves

A change of boundary or flow conditions at one section of the pipe cannot be immediately transferred to another part of the pipe. The interface between the part of the fluid that is affected, and the fluid that is unaffected is referred to as a transient pressure wave. This wave will travel through the system at the wave propagation speed. This wave will reflect at either a boundary or as the wave passes into a zone of different impedance. [18, page 222]

Standing Waves

Whenever reflections are present, the combination of forward waves and reflected waves yields a standing wave in the system [18, page 222]. Such waves represent the energy storage in the system. Resonance occurs whenever the valve frequency in the system matches one of the natural frequencies. The resonance appears as a standing wave in the system. It will first occur when the flow oscillates with the fundamental frequency of the system, theoretically known to be twice the reflection time given by equation 2.6. This is illustrated for a simple system in figure 2.1. However, standing waves are not limited to these frequencies, but will always be present in the system to some degree.

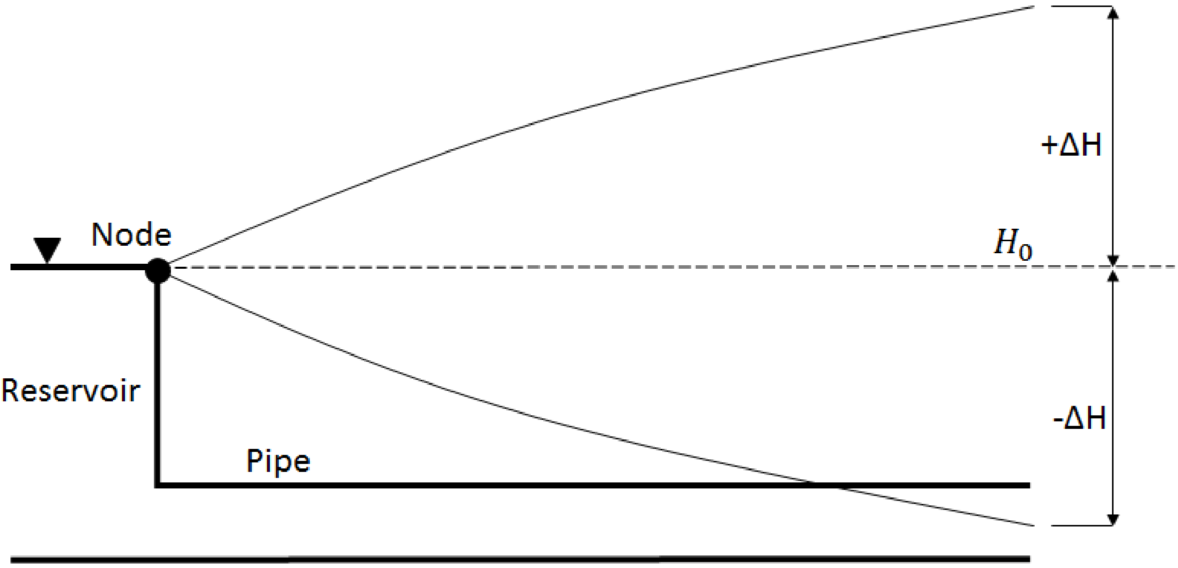


Figure 2.1: Standing wave at fundamental period in simple pipeline [18, p.237]

The periods of the remaining harmonics occur as fractions of the fundamental period. This illustrated by the fifth harmonic in figure 2.2.

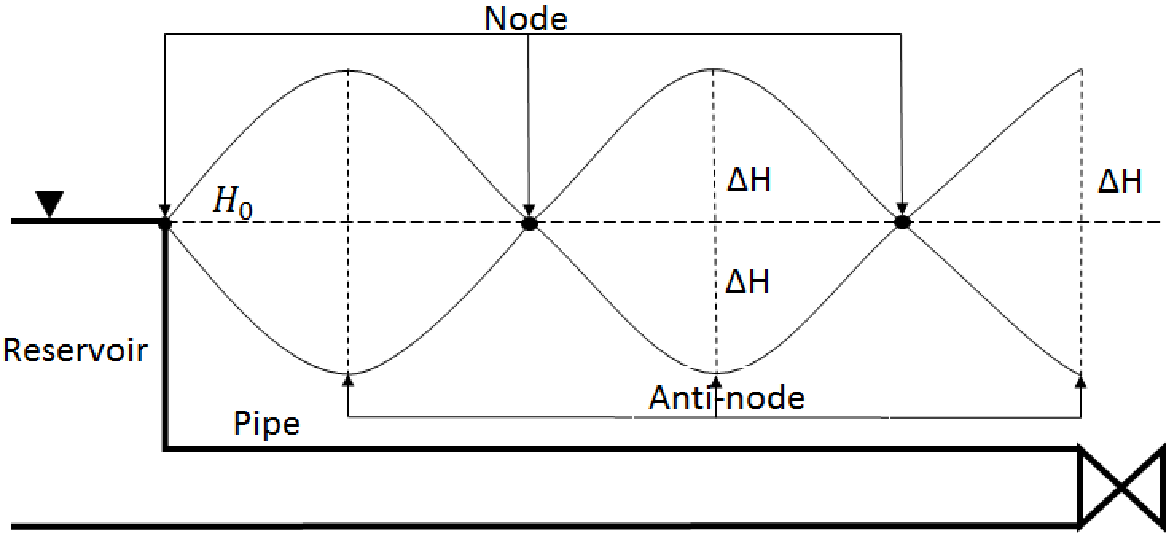


Figure 2.2: Standing wave at the fifth harmonic in simple a pipeline [18, p.238]

Impedance

The characteristic acoustic impedance of a medium is defined by formula 2.7 [18, page 209].

$$Z = \rho a \tag{2.7}$$

where ρ is the fluid density and a is the wave propagation speed.

The affect of a change of impedance on the travelling waves is illustrated in figure 2.3. At time 1, the wave propagating through the system has not yet reached the smaller pipe, and the amplitude of this wave is given by the wave speed in pipe 2. When the wave reaches pipe 1, the impedance increases due to the increase in wave speed. This change in wave speed is in accordance with formula 2.4 as the pipe diameter decreases. This results in a higher amplitude of the travelling wave, and a positive wave is reflected. This reflected wave will increase the head in pipe 2.

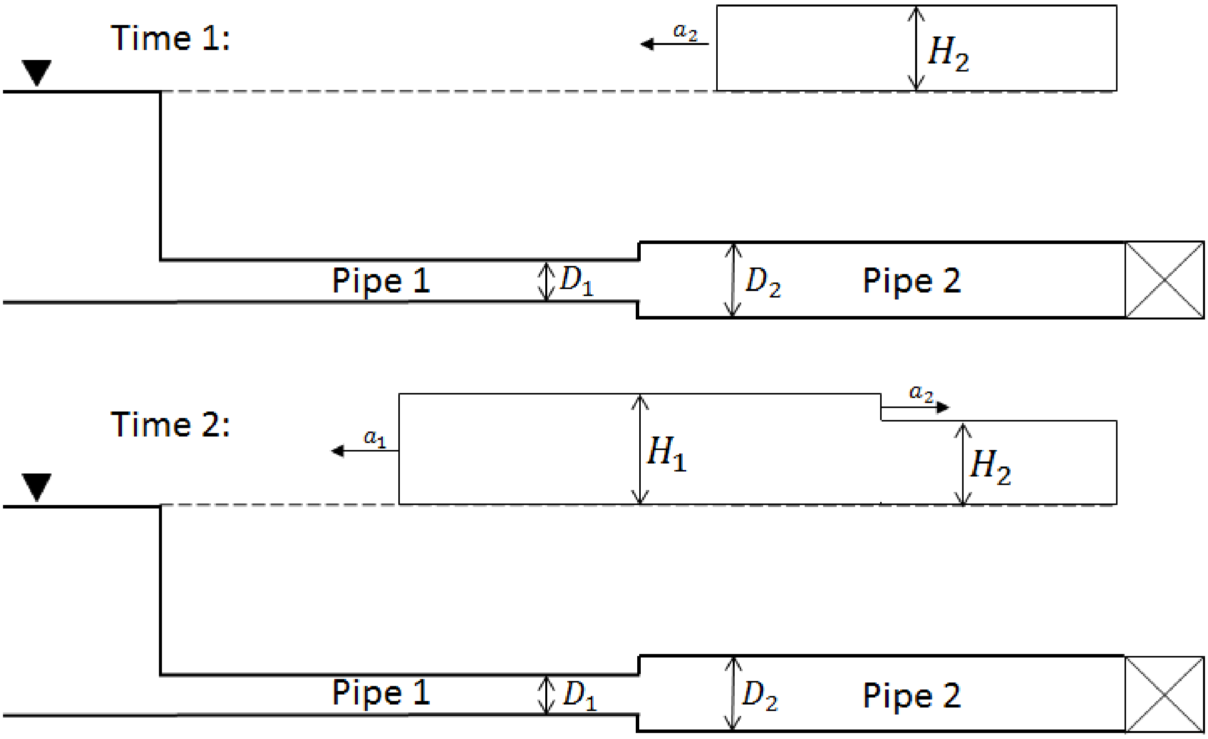


Figure 2.3: Illustration of how a change of wave speed affects the travelling waves

To sum up, there will be a wave reflection when the wave enters a zone of different impedance. However, the entire wave will not be reflected, some will also be transmitted.

2.5 The water hammer transient

The water hammer transient typically follows what Thorley [15] refers to as a 'rapid event. This is usually a rapid valve closure. If such an event occurs, a pressure wave will travel through the system at the wave propagation speed. This pressure wave is related to a coinciding change in flow. When the pressure wave reaches a boundary, it will be reflected back to the origin of the rapid event. The altered flow conditions will cause a new pressure wave, with opposite sign, to travel through the system until the boundary is reached again. Here it is reflected again, and moves back to the origin of the rapid event. The magnitudes of the pressure waves will eventually be dampened out by the presence of friction in the flow loop [10, page 8-16]. The magnitude of the change in the head can be approximated with the Joukowsky equation [15, p. 7]:

$$\Delta H = \frac{a}{g}\Delta V \tag{2.8}$$

This behaviour can be illustrated by the time development of the head for a simple system. This system consists of a valve and reservoir, interconnected with a straight pipe with no friction. If the valve is closed immediately, the Joukowsky head will develop. The period of the pressure signal will be the natural period of the system.

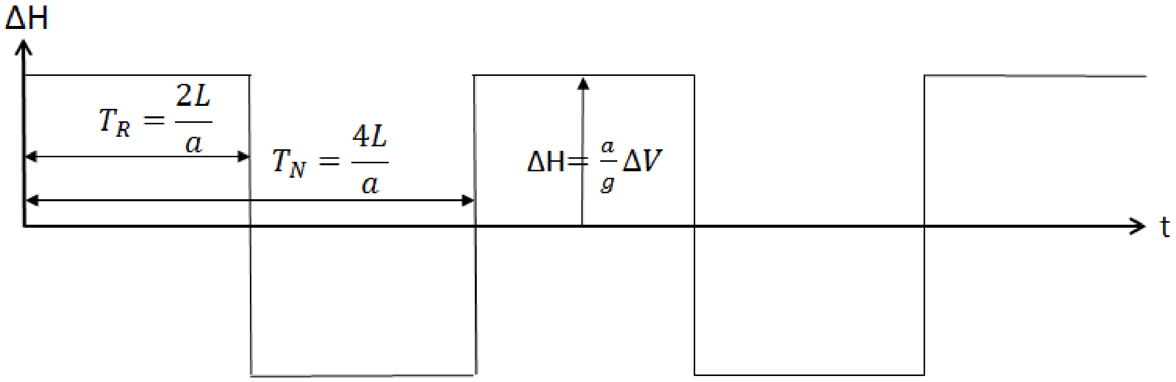


Figure 2.4: Time development of the water hammer transient at the valve

2.6 The effect of free gas

There is a high chance of air as a free gas entering the system before, or even during the experiments. The removal of this free gas can be challenging due to the extent of the pipes. It is shown that free gas will affect the transient behaviour of the system. Due to this, the effect of free gas on the transients must be explained, at least in some detail.

The gas can mainly be present in the flow in two different ways, either as bubbles moving with the flow or as pockets of air trapped somewhere in the system.

It is proven that even small amounts of free gas in a liquid flow will greatly affect the wave propagation speed, a [18, p. 9]. This is illustrated in figure 2.5 that shows the effect of up to one volume percentage of air present in the water as bubbles. The plots are generated from formula 2.4, by changing K and ρ. This will effectively affect a transient pressure wave by two different concepts, called scattering and attenuation.

Attenuation is due to the relatively high compressibility of the gas bubbles present in the fluid. As a pressure wave propagates through a fluid and gas mixture, the bubbles of gas will be compressed like a spring. This will again accelerate the water, which in turn cause more gas bubbles to be compressed. This process is not loss-free, and energy is dissipated as thermal energy. This effectively causes the wave to move at a lower velocity, compared to when moving through homogeneous mediums. [18, 10]

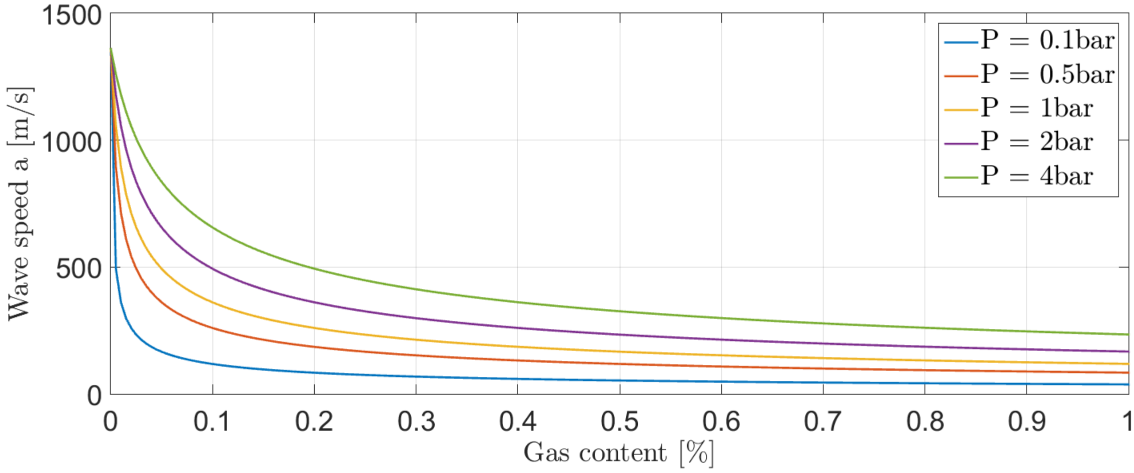


Figure 2.5: The effect of free gas evenly spread as bubbles on wave speed in a pipe flow

Scattering occurs as the pressure wave moves through zones of different impedance. According to formula 2.7, the impedance of the medium will change as the speed of sound is changed.

2.7 Losses

The losses in the system in a hydraulic system will be the sum of minor and major losses. Major losses are losses frictional losses such as the head loss through a long stretch of pipe. Minor losses are losses through components such as valves and bends.

The head loss through a pipe can be found by the head loss relation: [2, page 346]

$$h_L = f_L \frac{L}{D} \frac{V^2}{2g} \tag{2.9}$$

where f_L is the friction factor in a fully developed pipe flow, L is the pipe length, D is the internal pipe diameter, and V is the flow velocity.

For a turbulent pipe flow, the f can be calculated by the Colebrook equation [2, p. 357].

$$\frac{1}{\sqrt{f}} = -2log_{10}\left(\frac{\epsilon/D}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) \tag{2.10}$$

where f is the friction factor in a fully developed turbulent pipe flow, ϵ/D is the relative pipe roughness, and Re is the Reynolds number.

Minor losses are given by the minor head loss equation: [2, page 365]

$$h_L = K_L \frac{V^2}{2g} \tag{2.11}$$

where K_L is the loss coefficient.

2.8 Fast Fourier Transform

The Discrete Fourier Transform (DFT) is a very handy tool for converting a time domain signal to its frequency domain counterpart. This can be achieved very efficiently by the Fast Fourier Transform algorithm (FFT). The discrete Fourier transform is given by equation 2.12. [5]

$$X_k = \sum_{n=0}^{N-1} x_n e^{-i2\pi kn/N} \tag{2.12}$$

where N is the sample length, x is the discrete time domain signal. X_k is a complex number, and by representing it in the complex plane, one can obtain both the magnitude and the phase of X_k. If X_k has a magnitude at a frequency, it indicates the presence of that frequency in the time domain signal.[11]

An example of how the FFT can be used to identify the different frequencies and their magnitude is provided in figure 2.6. The two plots on the top show two sine signals with different amplitude and frequency. In the bottom left figure, these two sine functions are combined, rendering more complex signal. The FFT of this signal is plotted in the bottom right figure, showing how the complex signal is a sum of the two simple signals.

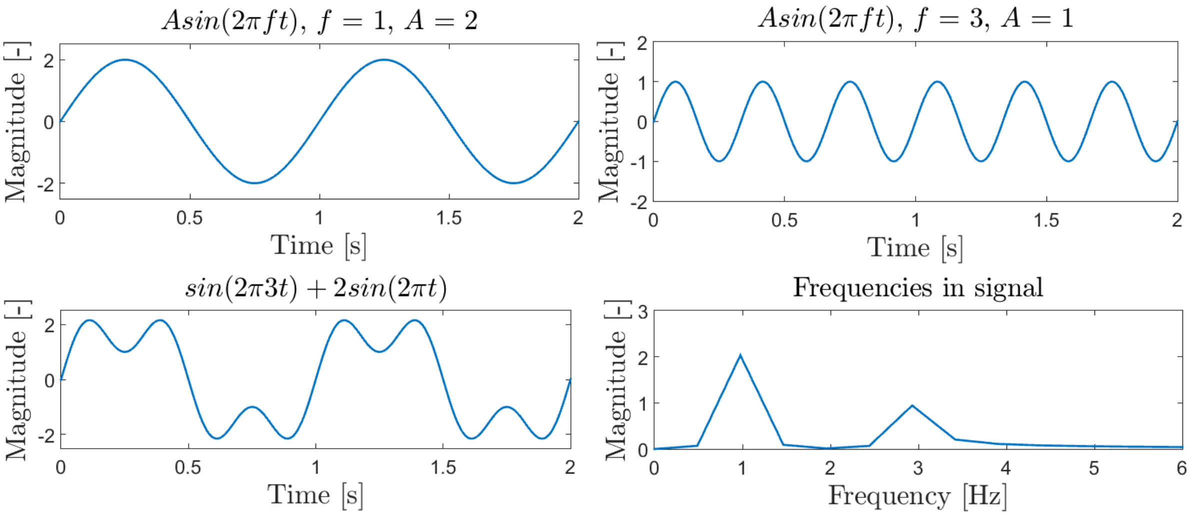


Figure 2.6: Example of how a Fast Fourier Transform can be utilized

The sampling frequency must be at least twice the highest frequency present in the signal. If this is not the case, the higher frequencies will fold over and overlap with the lower frequencies. This is called aliasing. [11]

Chapter 3 Design of the Flow Loop

3.1 Flow loop components

A lot of preliminary work on the design is necessary to ensure the proper mode of operation for the flow loop. This involves determining the necessary components and making sure that they fulfil the requirements imposed by the experiments.

3.1.1 Excitation valve

Two different kinds of valves are required in the system, one for creating oscillatory flow and one for creating the water hammer transient. The performance of the valves is instrumental for the success of the experiments.

The oscillatory flow excitation can be achieved in several ways, e.g. by governing the pump as demonstrated by Pingju Li [9], or by a valve as demonstrated by Svingen [13]. This cannot be achieved by any valve, a special design is required. Svingen's valve design has been proven to work in similar experiments, so his design is chosen for this experimental setup as well. The valve must be redesigned and manufactured to fit the flow loop.

General description of valve

The valve for oscillatory flow excitation is mounted at the outlet of the flow loop. It consists of a circular plate with a milled out slit in the middle serving as the outlet, a disc with a number of superimposed sine periods, as well as a motor to rotate the disc. The valve works by rotating the sinusoidal disc across the outlet, generating a sinusoidal variation of the outlet area as described by equation 3.1. The magnitude of the excitation can be varied by changing the amplitude of the sine function superimposed on the disc or by changing the slit outlet area. The frequency of the excitation can be varied by changing the speed of the motor that the disc is attached to, or by changing the number of superimposed sine periods on the disc.

$$A_{outlet} = A_0 + \Delta A sin(2\pi f t) \tag{3.1}$$

The valve for the water hammer transient is combined with the valve for oscillatory flow. Through the plate, there is another pipe with a standard ball valve. A more detailed description of this solution will follow in the description of the experimental setup.

Valve requirements and dimensions

The valve will be required to run at a wide range of frequencies. For these initial test, it was decided to investigate the lower range of the frequency spectrum. This means frequencies ranging from 0.1Hz to about 20Hz.

It is necessary to design the valve in such a way, that no undesirable non-linearity is introduced through the valve. This might be achieved by a design that creates a close to uniform velocity profile through the slit.

It is necessary to determine the disc parameters, namely n and A_D as well as the slit opening area, A_{slit} determined by the width w and the height h. This is done by simulations in LVTrans.

The slit outlet area, A_{slit} should be adjustable to enforce greater flexibility in the magnitude of the excitation. If A_{slit} is made smaller, the relative change in area will be bigger as the disc rotates across the slit. This can be achieved by partly closing off the slit by a plate.

Disc

The disc can be described mathematically in polar coordinates as:

$$r = r_0 + A_D sin(n\theta) \tag{3.2}$$

where r_0 is the radius of the circle on which the sines are superimposed, A_D is the amplitude of these sines, and n is the number of sines for one full rotation. r_0 is determined by the distance from the motor shaft on which the disc is mounted, and the slit. More precisely, the distance from the centre of the motor shaft until the closest edge of the slit is given by $r_0 - A$. n is determined by the required excitation frequency and A_D is determined by the required amplitude of the excitation.

The disc must be made of a suitable material. It is necessary for the disc the slide easily inside the groove at high rotational speed. The fit of the disc inside the groove must be rather tight, to avoid excessive spillage of water. It should have a low density, minimising vibrations when the disc is rotating. The material stiffness should be moderate, while still having high impact strength and excellent abrasion resistance. A low coefficient of friction will ensure that a low torque is required to rotate the disc. The material must also have a low water absorption.

Motor

An electric motor is required to rotate the disc. The required speed will be governed by the required frequencies, and the required torque will be governed by the friction force working on the disc while sliding through the slit. It is hard to do calculations on the required torque, so a good margin should be implemented. To reach the lower frequency range, a frequency of 0.1Hz is required on the shaft. This probably means that some transmission is required, depending on the motor specifications.

3.1.2 Pump

It is possible to provide flow and pressure to the system in multiple ways, e.g. by a reservoir or a pressure tank. A pump seems like the most convenient solution for this flow loop.

Selection of pump

The selection of pump is critical for the success of the experiments. It must be able to supply the required head and flow to the system. It is also required that it does not impose disturbing frequencies on the flow in the system. This might be the case if a reciprocating pump is chosen [16, page 10]. Hence a centrifugal pump will be better. It must also be able to withstand the demanding transient flow conditions imposed by the valves at the outlet of the system. This must either be handled directly by the pump(e.g. by increasing inertia of rotating parts) or by other measures to ensure steady operation of the pump.

The dimensioning of the pump is done by the later described simulation in LVTrans. The results from one of these simulations are shown in figure 6.7.

Pump Characteristics

The pump characteristic is necessary for including the pump in a numerical model that will be used prior to further experiments in this flow loop. The characteristic was found from a small experiment. One of the pumps was driven at full speed while pumping water through the entire flow loop. The valve at the outlet end of the pipe was slowly opened incrementally while letting the flow stabilise in between each adjustment. The flow and pressure at the discharge side of the pump was continuously monitored, yielding the pump characteristic seen in figure 4.6. The placement of the flow meter and the pressure sensor can be seen in figure 4.1. The pressure sensor is placed only a couple of meters downstream of the pump, so it should give the pump head quite accurately. The pump characteristic can be validated by calculating the flow delivered by the pump with the flow measured by the flow meter.

3.1.3 Suppression of transients

As discussed, it is necessary to dampen out the transients to ensure steady operation of the pump. This dampening should take place directly upstream of the pump because the presence of transients is required in the remaining part of the system. One way to achieve this is by releasing excessive energy from the system, or drawing required energy into the system. Examples of devices that provide this functionality include air vessels, accumulators, surge tanks/shafts, etc. For this system, an air vessel or an accumulator seems like the most reasonable solution. An air vessel is a tank, filled with both liquid and compressed gas. The gas may or may not be separated from the liquid by an elastic bladder. This prevents the gas from being sucked into the rest of the system, which is undesired during operation. An accumulator is just a small scale air vessel. [15, p. 24-29]. This component is considered a part of the pump solution as it will directly affect the pump's operation.

Dimensioning of accumulator

The dimensioning of the accumulator was done by a few simulations in FloMASTER. It was done by simulating the system at the same flow conditions as was expected during the experiments. An accumulator was inserted, and the system response was observed. The simulations were done with both the water hammer and oscillatory flow. The FloMASTER network for the oscillatory flow can be seen in figure 3.1. The network for water hammer simulations is very similar.

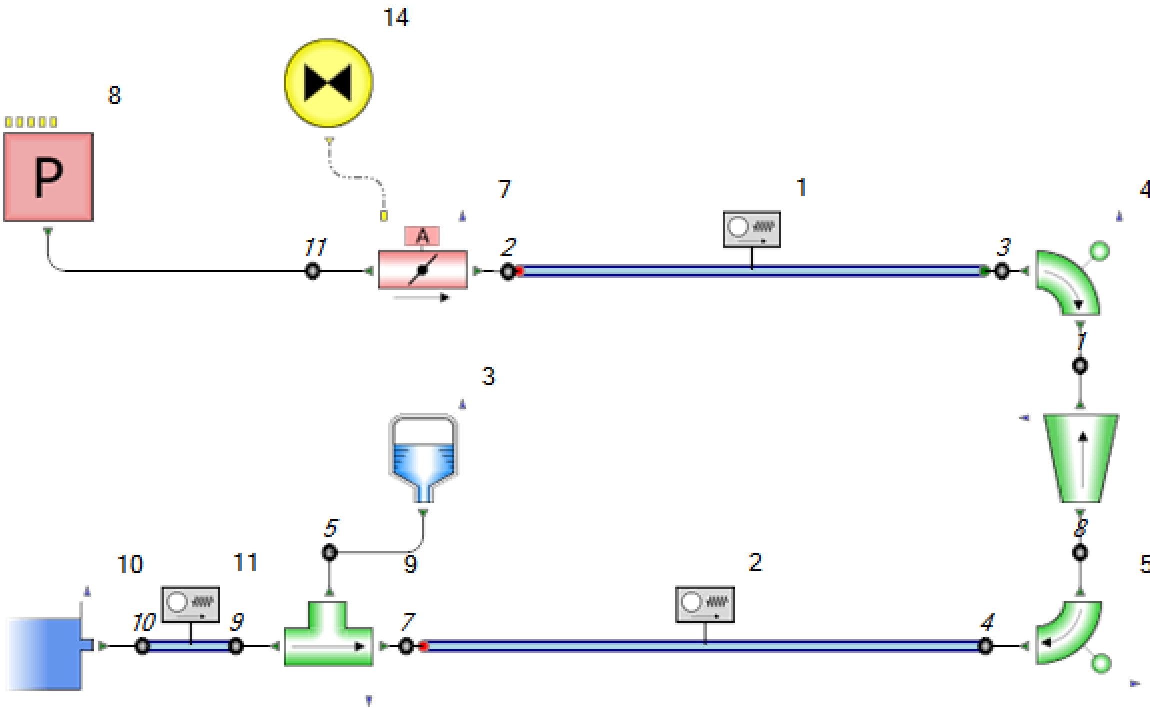


Figure 3.1: FloMASTER network for dimensioning of accumulator

In figure 3.2, the resulting pressure oscillations are compared at two different locations in the system. The red curve indicates the pressure felt by the pump with the accumulator, while the blue curve shows the pressure felt by the pump without an accumulator in the system. The qualitative difference is obvious, showing that the accumulator dampens out the oscillations, providing stable operating conditions for the pump. Quantitatively, the amplitudes of the pressure oscillations compare as follows:

- Amplitude in pressure fluctuations without accumulator: $0.41bar$
- Amplitude in pressure fluctuations with accumulator: $0.0016bar$

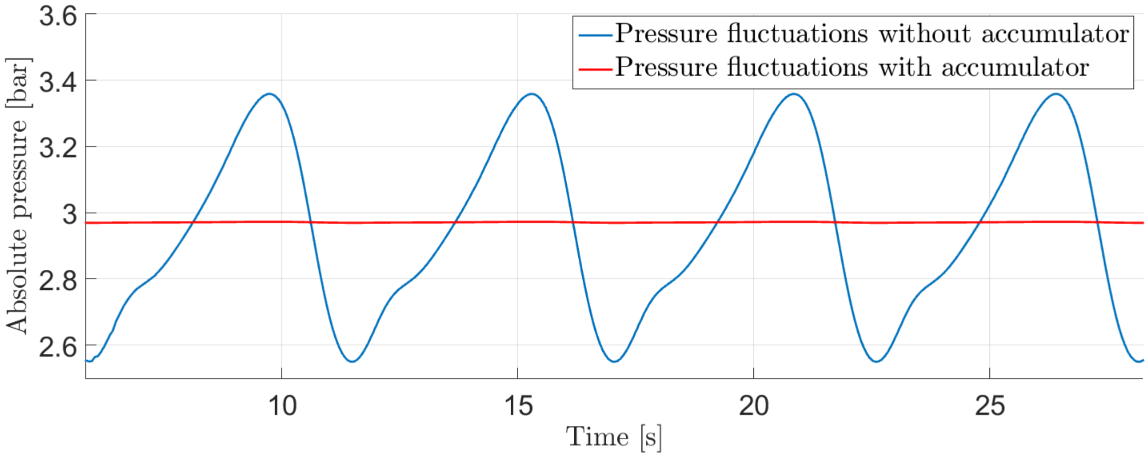


Figure 3.2: Comparison of pressure fluctuations during oscillatory flow conditions with and without accumulator in the system

In figure 3.3, the resulting pressure conditions following a rapid valve closure are compared at two different locations in the pipe. The purpose of the accumulator during such flow conditions is to effectively dampen amplitudes of the pressure waves before they can reach and damage the pump. The pump housing is not able to withstand large amplitudes in pressure, and a massive pressure wave can compromise its mechanical integrity. In figure 3.3, the red curve shows the pressure at a location close to the pump, while the blue curve shows the pressure at an arbitrary location in the pipe. This location is not very important because the amplitudes will(in theory) be the same throughout the system. The positive shift in the red curve compared to the blue curve is due to the pressure loss through the system. The results show that the amplitudes are effectively dampened. Quantitatively, the amplitudes compare as follows:

- Amplitude in pressure fluctuations inside pipe: $2.13bar$
- Amplitude in pressure fluctuations close to pump: $0.016bar$

Both of these simulations are done with an accumulator with a total volume of 20 litres, half filled with water and half filled with air at operating conditions. The simulations show that 20 litres are more than enough. It could be possible with an accumulator with a smaller volume, but some practical considerations must also be taken into account. It is not desired that air is accidentally sucked into the system, so a good margin in the water

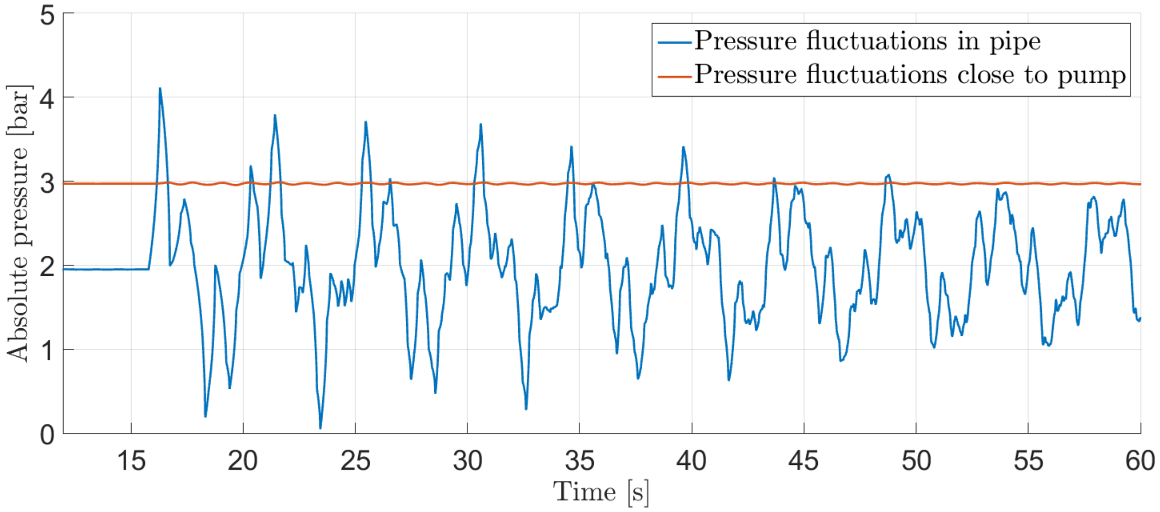


Figure 3.3: Comparison of pressure fluctuations in pipe and fluctuations close to pump in the time following a rapid valve closure

volume available in the accumulator should be implemented. If the air in the accumulator should be compressed by the pumps, the change in air volume from atmospheric conditions to operating conditions will be substantial. Treating the air as ideal gas yields that the compressed air volume will be about a third of the initial air volume if the gas pressure is changed from atmospheric to 3 bars. This means that some volume for the initial amount of air is also required. It is by no means impractical to build an accumulator with a total volume of 20 litres. However, a smaller accumulator volume creates some practical challenges. Hence a 20-litre accumulator is recommended.

3.1.4 Pipe connection

A connection between the two pipes is also necessary. The purpose of this connection is to connect the 7" and the 5.5" pipe inside the IRIS facilities. There are a few requirements to this component.

The connection is considered a passive component of the flow loop but should be designed so that it does not interfere with the experiments any more than necessary. Ideally, this would not be a point of reflection for the pressure waves, but due to the different diameters of the pipes, this is inevitable. The wave speed will change when the diameter changes, causing a change in impedance that again causes a wave reflection, according to formula 2.4 and formula 2.7. This reflection can, however, be minimised by removing all other factors that cause a change in impedance. This includes the stiffness of the material of the connection. This means that the connection should have the same stiffness as the pipes, meaning that it should be made of steel.

It was also considered to design the connection to make it a suitable location for trapping and venting out air. This was not prioritised for this initial round of testing of the flow loop. The connection is not the naturally highest point of the pipes, limiting the effect of using this as an air venting location.

3.1.5 Other components

A few more components are necessary to provide the required mode of operation for the experiments. This includes the measurement system, water recycling system, pipes and a solution for flushing and filling of the pipes. These components are not discussed here, as they are either an existing part of the test facility, or they have been decided by one of the other involved parties.

3.2 Development of LVTrans network

LVTrans is a program for simulations on liquid flow in piping systems programmed in LabVIEW. It is specialised for use on transients in hydropower plants but is also applicable for this flow loop. LVTrans is based on the method of characteristics yielding close to the analytic solution. [14]

The LVTrans network will particularly be used for the pump and the rotating valve requirements. The development of the network is rather straightforward. Existing components can be dropped into the working environment and linked together in the user interface. Realistic dimensions and parameters are given for all necessary components. The resulting network can be seen in figure 3.4. The valve itself exists as a predefined component in the LVTrans user interface. The pump can be modelled by a reservoir for design purposes.

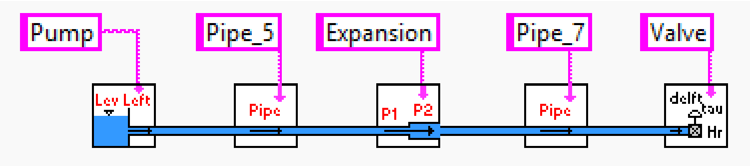
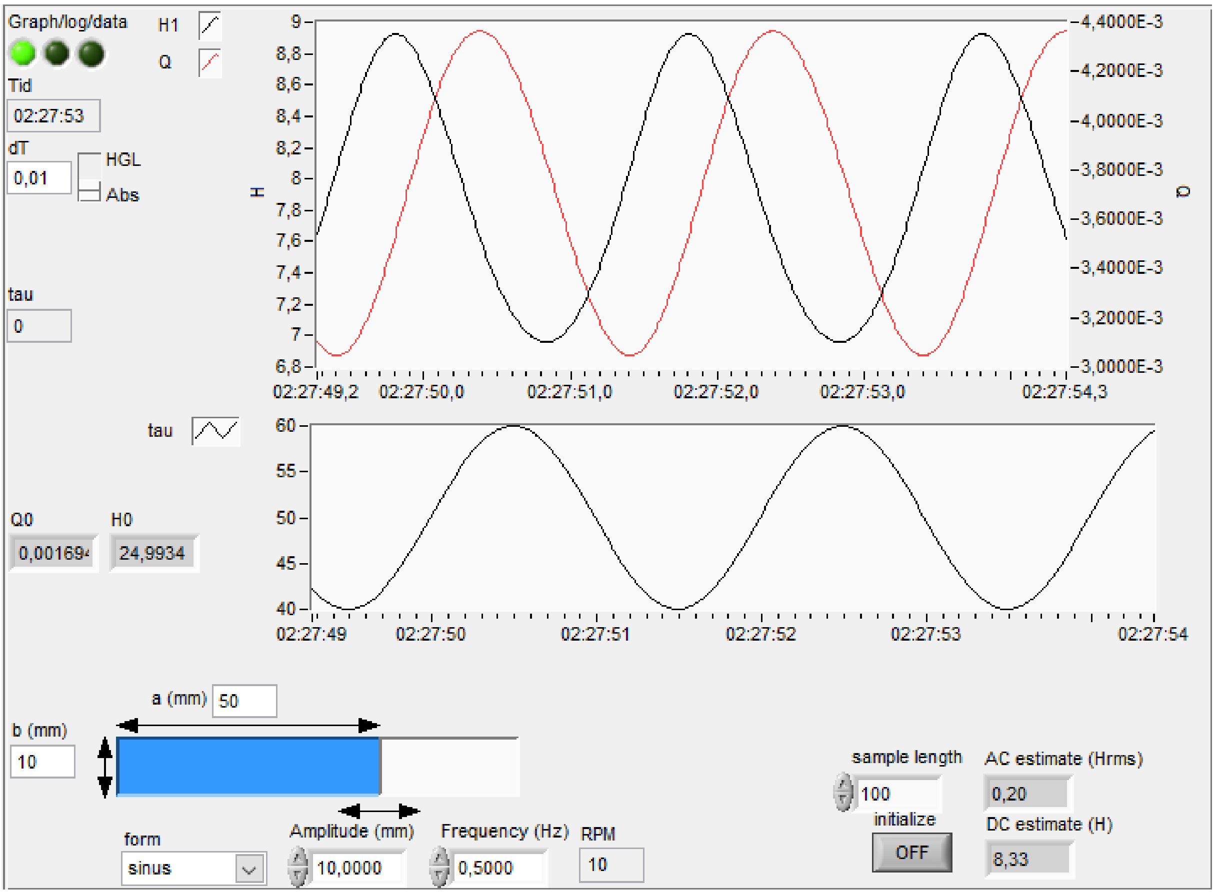


Figure 3.4: LVTrans network

3.2.1 Dimensioning of rotating valve and pump

Since the dimensions of the rotating valve and the pump are subject to the same flow conditions, their dimensions can be found simultaneously by one simulation. The dimensions of the slit and the amplitude of the disc, as well as the constant pump head, can be varied by trial and error until satisfying flow conditions are met. For the valve, this is done directly in the user interface, as seen in figure 3.5.

For the initial test, there are no strict flow requirements. The valve should generate head amplitudes of a few meters, say 5 metres. There are no special requirements to the velocity through the pipes. Cost is, of course, a limitation, so moderate flow rates and heads are preferred, to avoid excessive pump cost. In this case, moderate flow rates are in the range of 2-10l/s.



Tid: 02:27:53

dT: 0,01

tau: 0

Q0: 0,00169

H0: 24,9934

a (mm): 50

b (mm): 10

form: sinus

Amplitude (mm): 10,0000

Frequency (Hz): 0,5000

RPM: 10

sample length: 100

initialize: OFF

AC estimate (Hrms): 0,20

DC estimate (H): 8,33

Figure 3.5: LVTrans interface for rotating valve

The model should later be verified with the experiments. This can be done by operating the flow loop with the same head and valve settings as in the simulations and comparing the results directly. This type of comparison is done in figure 6.7.

The results from the simulations are presented in the next chapter.

3.3 Project management

In addition to the technical work presented in this thesis, the management of this project has been a substantial part of the candidate's work. This has involved much organising work, such as meetings, communication with all the involved parties and other minor tasks such as writing smaller informative texts about the experiments to invoke the interest of other parties in the research community. The author has efficiently served as a junction between all the involved parties.

This process commenced with a start-up meeting in Stavanger in early February, where all the involved parties participated. The process was then started, and it was decided that a rugged design of the flow loop should be ready by March 1st. It was agreed that the experiments should take place in early May. During this phase, it was necessary to establish what kind of behaviour that was required in the system, and what kind of components that was required to achieve this. This work consisted of gathering information from all the parties and creating solutions that worked for everyone. This e.g. involved finding a compromise between the desired flow velocities in the system, and a possible solution when considering the cost of the pumps. During this process, it was important to efficiently distribute information between the involved parties to make sure everyone was informed about the progress of the project and the relevant limitations that emerged along the way. This process lasted until the end of April.

The following phase was closer to the time of the experiments. During this phase, it was necessary to revise parts of the design. This was often due to practical limitations in the manufacturing of the different components, or limitations imposed by the owner of the flow loop. It was important to coordinate the revisions in such a way that all parties remained satisfied with the design. It was also necessary to establish what kind of experiments that was desired to conduct during the initial tests.

The next phase was during the week of testing and execution of experiments in Stavanger. During this phase, it was important to make sure that all parties knew what to do so that the testing and experiments could be conducted in a safe and efficient manner.

The final phase revolves around writing this report, making sure that all the necessary information and recommendations for further use of the flow loop are thoroughly documented.

Chapter 4 Experimental Setup

4.1 Flow loop components

The following components have been chosen for the flow loop, based and simulations, practical and economical considerations. An overview over the flow loop with all the components is shown in figure 4.1.

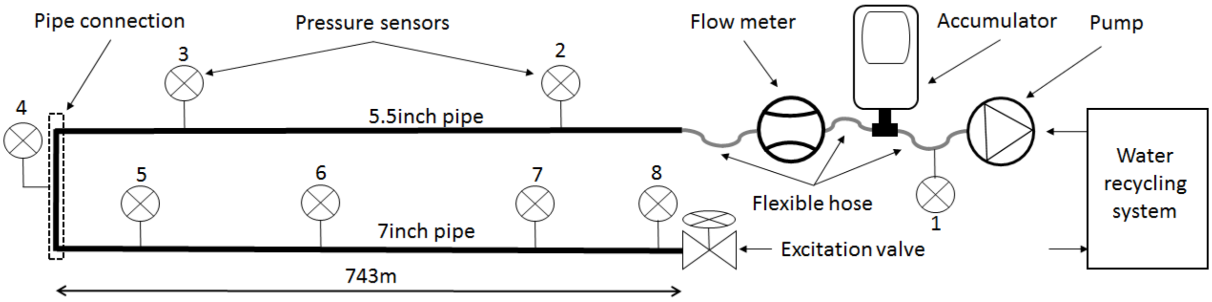


Figure 4.1: Overview of the components in the flow loop

4.1.1 Excitation valve

The following dimensions were obtained from the LVTrans simulations for the valve and the disc.

- A_D = 10mm
- n = 1, 3 and 4
- A_{slit}: w = 50mm and h = 10mm, $A_{slit} = 5 \cdot 10^{-4} m^2$

A 3D model of the suggested design is shown in figure 4.2.

The final valve solution is pretty similar to what was proposed. Some adjustments were necessary to ensure fit with preexisting components at the site. One of these adjustments were to move the outlet a bit to the side, to avoid conflict between the rotating disc and the ball valve. A picture of the combined rotating valve and water hammer valve can be seen in figure 4.3.

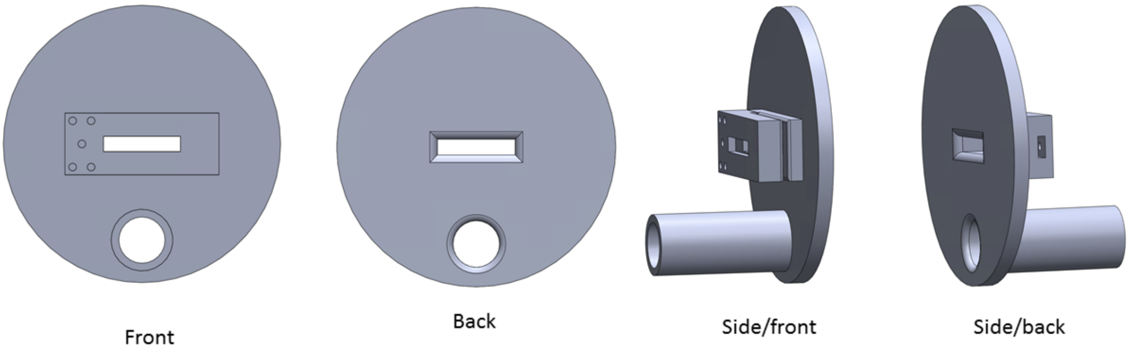


Figure 4.2: 3D model of suggested valve

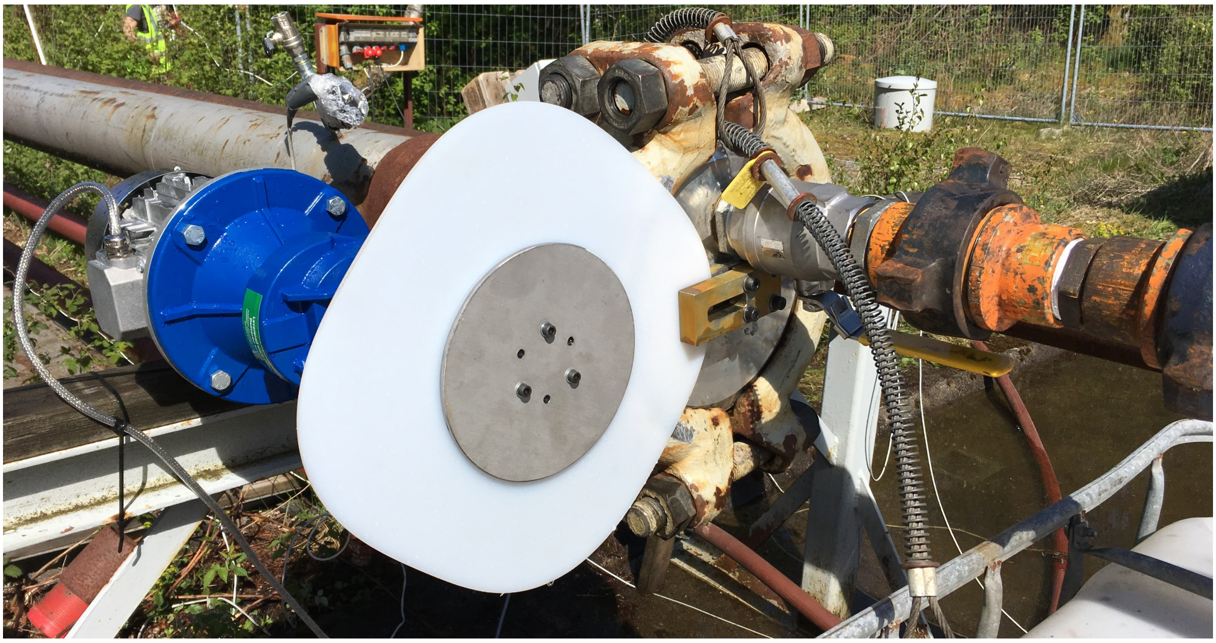


Figure 4.3: Photograph of the final valve design

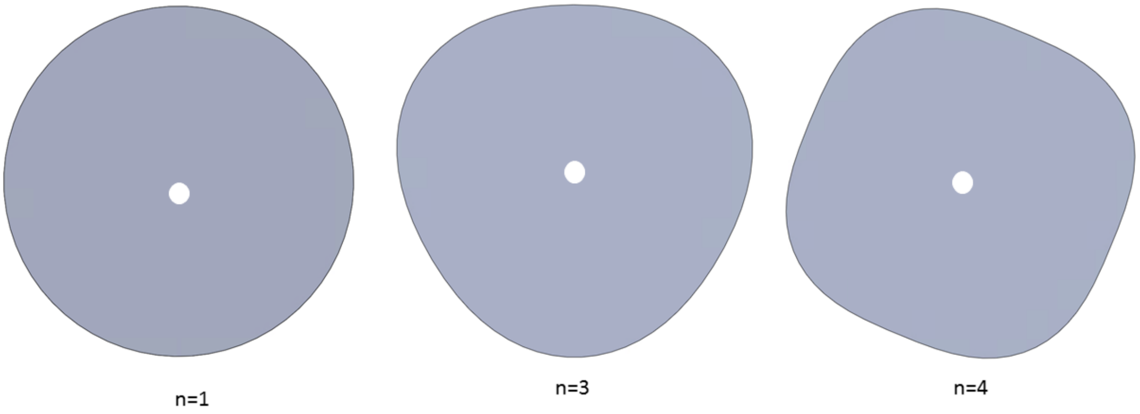


Figure 4.4: 3D model of suggested discs

The suggested discs for the initial tests are shown in figure 4.4

The material chosen for the disc is PE1000, a polymer with very suitable characteristics.

Motor

A 2.2kW AC motor was chosen for the rotation of the valve. The motor is controlled by a frequency converter that outputs frequencies ranging from 0.5Hz to 100Hz. The frequency is controlled by a potentiometer. A picture of this component is shown in figure B.9 in the appendix. The total transmission constant from frequency converter to the motor shaft, T_m, was found to be 0.158. This yields the frequency ranges for the various discs seen in table 4.1.

Number of sine periods on disc	Frequency range
n = 1	0.079-15.800Hz
n = 3	0.237-47.400Hz
n = 4	0.316-63.200Hz

Table 4.1: Frequency range for each of the discs

With this frequency range, the rotating valve should be able to excite the system at all the required frequencies for these initial tests.

4.1.2 Pump

The LVTrans simulations showed that the system requires a head of 20m to obtain the required flow rates. The final choice of the pumps ended up being very cheap and simple, as simulations show that only very moderate pressures and flow rates are required for these initial experiments. The pump specifications are included in table 4.2

A picture from the test site 4.5 shows the pumps and how they are connected to the rest of the system. Note that the pumps are connected in parallel to supply a higher flow rate

Model	FGP15A
Max capacity	233 l/min
Suction head	$9m$
Total head	$23m$
Max output	$3hp/2.24kW$
Max speed	$3000rpm$

Table 4.2: Pump specifications

to the system. The pumps can easily be reconfigured to be connected in series, providing a lower flow rate and higher pressure to the system.

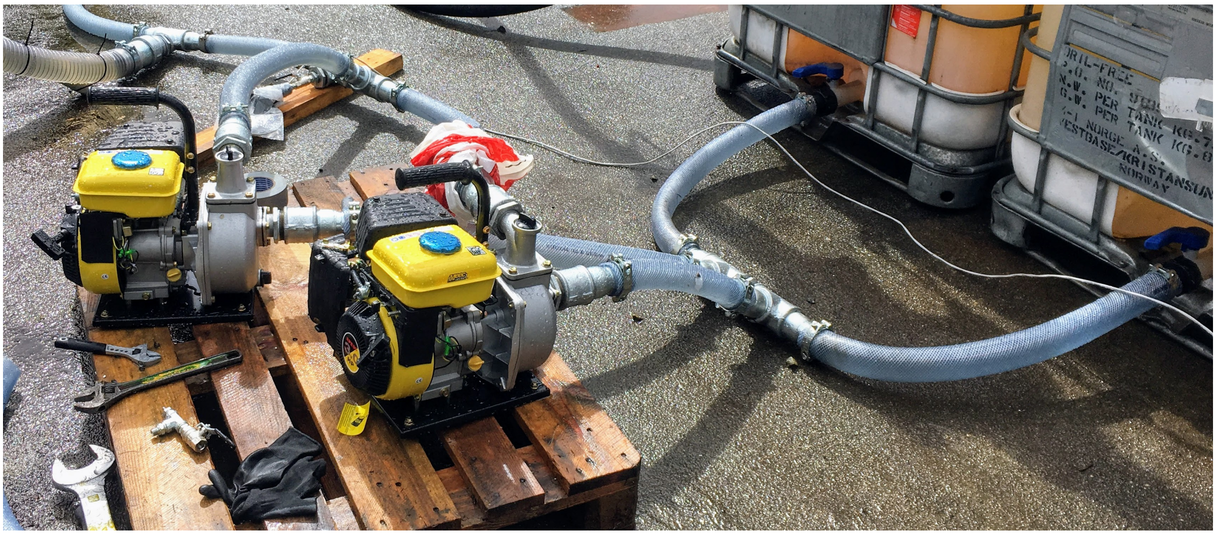


Figure 4.5: Pumps connected to water supply tanks

Pump Characteristics

The pump characteristic has been obtained through an experiment, but some processing of the results are necessary to obtain a useful characteristic.

Much noise is present in both the flow and pressure signal. This noise can be seen as the blue in the upper plot in figure 4.6, while the red curve is the pump characteristic with all the noise filtered away by averaging. The complete pump characteristic was not obtained experimentally due to some limitations in the experimental setup. The flow meter was not able to measure flow rates lower than $60l/min$, meaning that the full characteristic was not obtained for flow rates lower than this. Since the pressure loss through the system was always present, the full discharge of the pump was also not investigated. However, a complete pump characteristic for the entire flow range is provided in the lower plot in figure 4.6. This characteristic is obtained by using both interpolation and extrapolation. Extrapolation should always be used with caution, and the provided pump characteristic should be validated before it can be considered accurate. It does match the experimental curve inside the experimental range, but the values outside of this range are uncertain. For the lower flow range, the pressure is known for zero flow but is followed by a range of unknown values until the lower measuring limit of the flow meter is reached. The

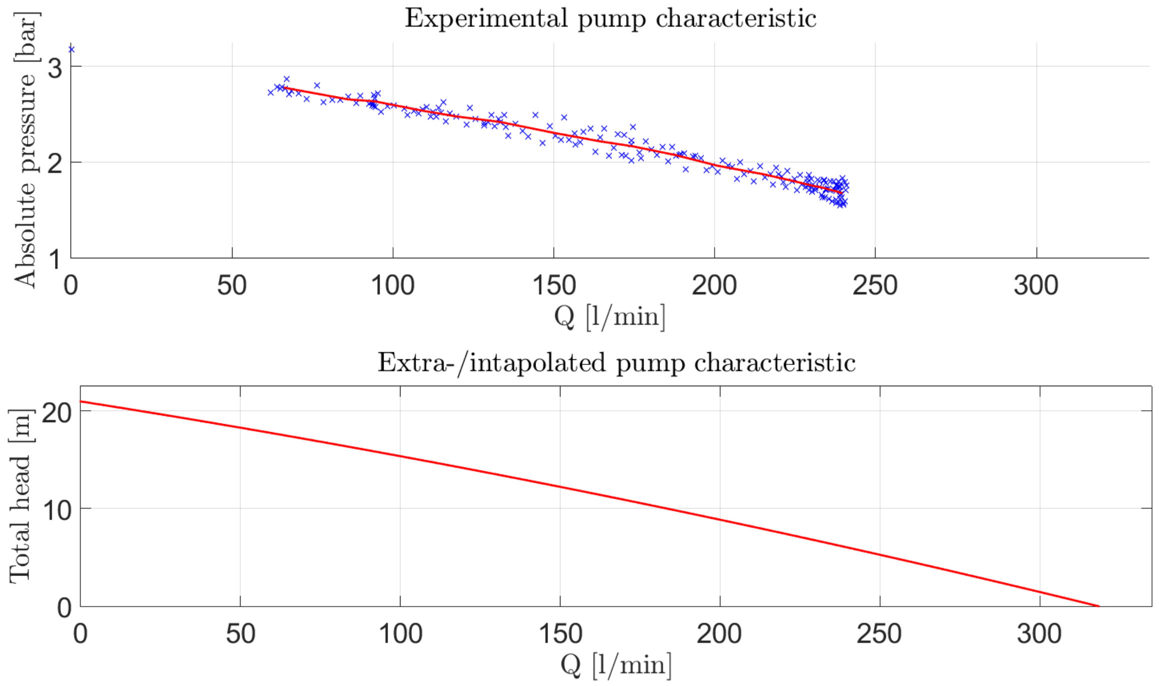


Figure 4.6: Pump characteristic for full speed. Upper plot for experimental but limited range, lower plot for extrapolated full range values

uncertainty in this range is considered to be quite small. The uncertainty in the upper range of flow is considered to be much bigger. Note that both the maximum discharge and the maximum head deviates from the values given by the manufacturer in table 4.2. This could indicate that the extrapolation is wrong, but also that the specifications provided by the manufacturer are inaccurate. The deviations are the following:

- Total head for zero flow deviates about $2m$
- Maximum discharge deviates about 100 l/min

These deviations can be due to many factors. Some strange behaviour was sometimes observed by the pumps, e.g. a sudden change in speed. In general, the pumps struggled to supply a steady flow rate to the system, even though no cause of changing flow conditions were apparent in the system. More tests should probably have been conducted to exclude some of this uncertainty. However, the pump was able to provide a flow rate very close to the maximum flow rate specified by the manufacturer while still overcoming the steady head loss through the system. This indicates that the pump can provide a higher maximum flow than the manufacturer indicates, but the exact maximum flow rate is not known.

For easy implementation in further simulations, a polynomial for the pump characteristic is provided. A second-degree polynomial yields good fit with experimental data.

$$H = -4.39 \cdot 10^{-4} Q^2 - 5.1 \cdot 10^{-3} Q + 20.96 \tag{4.1}$$

where H is the total head and Q is the discharge.

Validation of pump characteristic

The pump characteristic can be validated by comparison with measured data. More precisely, the obtained pump characteristic can be used in combination with pressure measurements to calculate the discharge of the pumps. This discharge can be compared to the flow measured by the flow meter. The results may also be used to assess the performance of the flow meter.

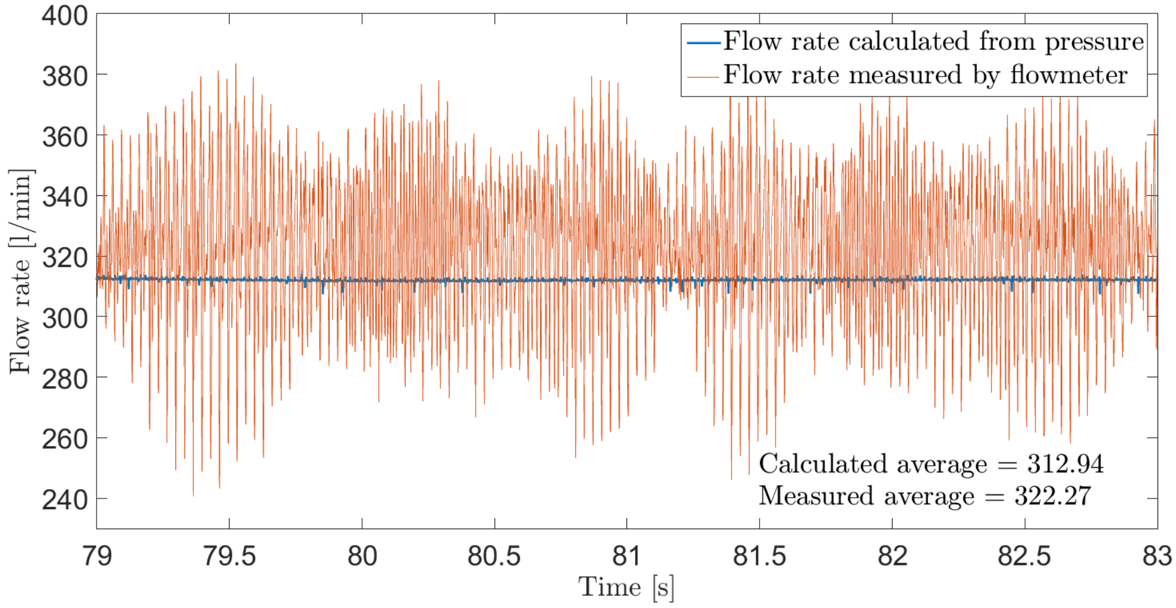


Figure 4.7: Validation of pump characteristic

For the steady flow case shown in figure 4.7, the results show that the measured average and the calculated average deviates a bit. It is possible that the pumps were running at slightly different speed than during the experiment for generating the pump characteristic.

Another interesting result is when the calculated flow is compared to the measured flow for the oscillatory flow. This is done in figure 4.8.

Here, there is an obvious phase shift in the signal produced by the flow meter compared to the flow signal calculated from the pressure. The flow meter is a turbine flow meter, and it was maybe expected that there would be some delay in this when the flow is changing rapidly as the turbine does not respond immediately to the change in flow. The amplitudes are also different, but this may be due to the placement of the flow meter. The flow meter will measure the flow coming from both the accumulator and the pump, while the calculated flow only accounts for the pump discharge.

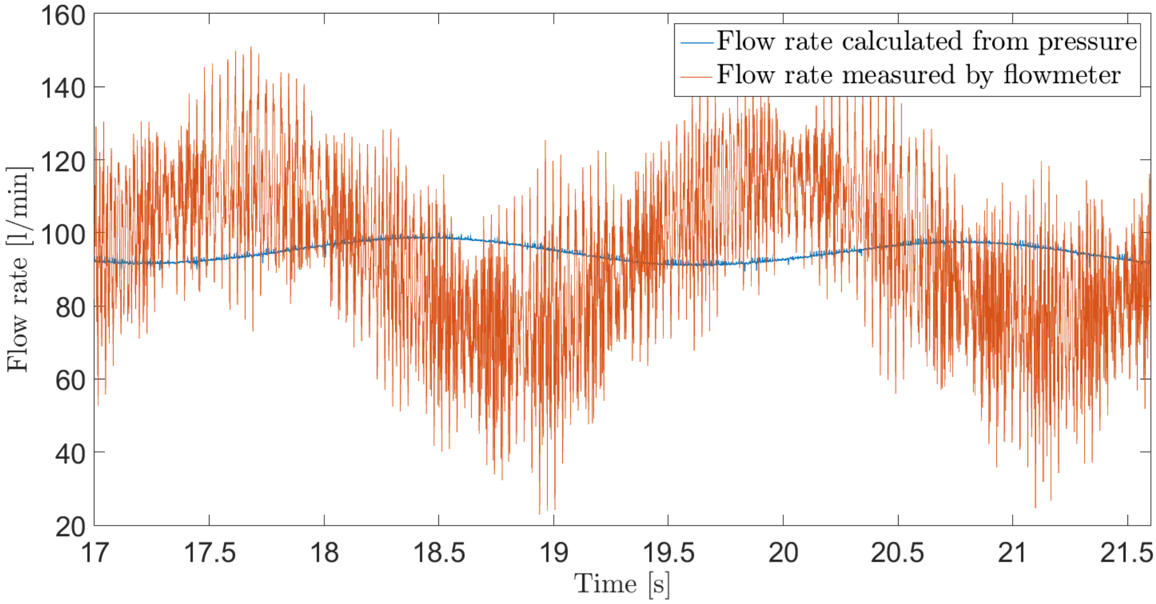


Figure 4.8: Validation of pump characteristic for oscillatory flow

4.1.3 Accumulator

The accumulator used during the tests are shown in 4.9. It follows the recommendations made considering total volume and placement. It consists of a long and thin plastic tube, with a valve placed at the upper end. The pipe is transparent to visually monitor the water level inside. It is also reinforced to withstand high pressures.

Figure 4.9: Photograph of the accumulator from test site

4.1.4 Pipe connection

The resulting component was in practice decided by the components already available at the facility. No photograph is provided, but a 3D model is provided in figure 4.10 The component consists of four 90 degree bends, interconnected with pipe connections. The diameter is also smaller than both of the pipe diameters. Both of these factors yield a higher head loss than necessary through the component. All the parts are made from steel, providing the desired stiffness. A hole is drilled in this component to fit a pressure sensor. This component also contains a valve, connecting the flow loop to the IRIS pumps and the external water supply used during filling.

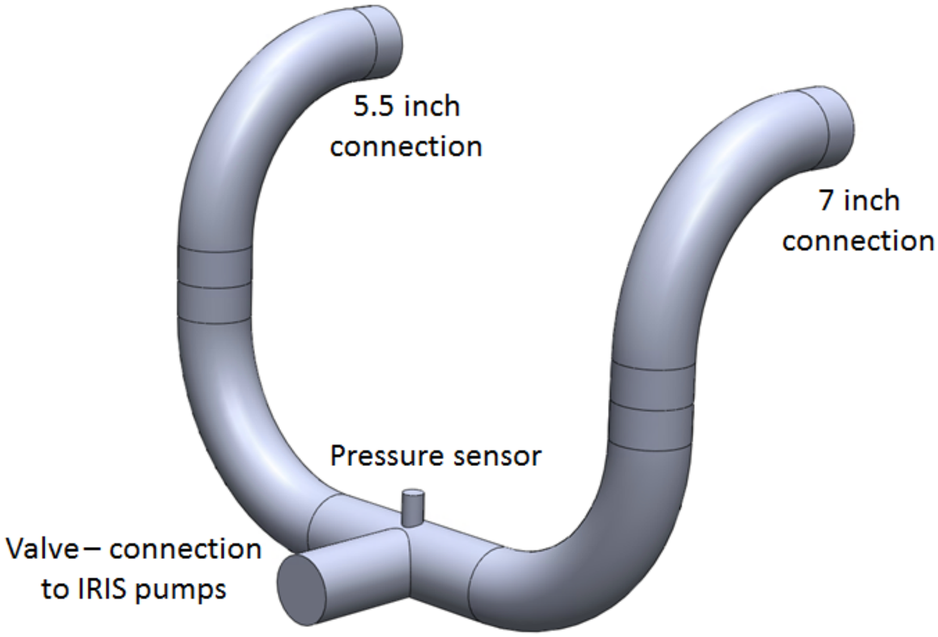


Figure 4.10: 3D model of the pipe connection

4.1.5 Measurement system

In total, eight pressure sensors are mounted on the pipe. They are all spread out quite evenly, as seen in figure 4.11. The red part of the plot indicates where the pipes were lifted up from the ground. There is a total of six sensors with a measuring range of 0-10bar and two sensors with a measuring range of 0-5bar.

Figure 4.12 shows how both pressure and flow are measured. The sensors are all connected by cables to a signal cable running along the full length of the pipes. This signal cable has several connection points along the way, making the connection to the signal cable possible without excessive cable lengths. The sensor output is a 4-20mA signal that is transferred by the signal cable back to the test site. Here, the signal is transformed to a 2-10V signal before it is recorded by a Voltage Input Module. This module converts the analogue signal to a digital signal. It is connected to a router by a network cable that makes wireless transmission to a computer possible. This is convenient because it makes it easier to monitor the flow conditions in the flow loop while making adjustments to e.g. pumps or valves. The signal is logged in a LabView program, outputting .tdms files.

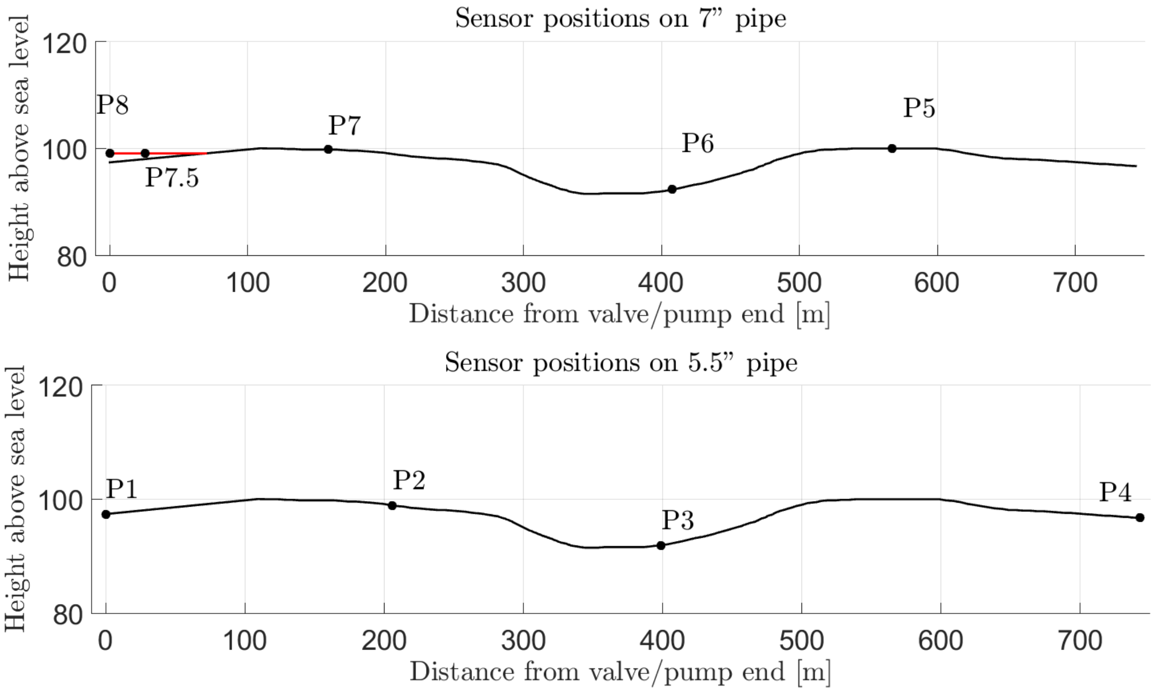


Figure 4.11: Sensor position and height profile of the pipes

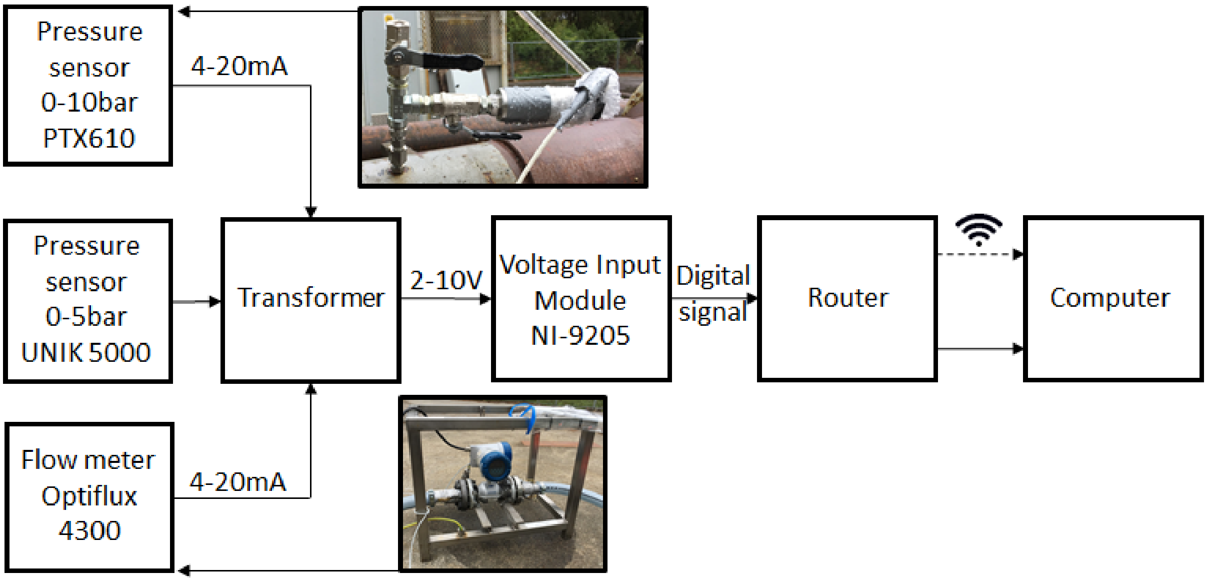


Figure 4.12: Overview of the measurement system

The measurement system is mainly designed by PhD candidate Ingrid Vilberg, with the aid of PhD candidate Carl Bergan. The LabView program is also created by Vilberg, based on a solution by Flow Design Bureau AS. Vilberg also did the calibration of the pressure sensors.

4.1.6 Water recycling system

To avoid unnecessary and excessive spillage of water on the test site, a water recycling system was also made. The basic setup of the system can be seen in figure 4.13.

Figure 4.13: Water recycling system

The recycling system consists of two $1m^3$ tanks, for storage and gathering of spill water. The two tanks are both connected to the pumps. For the experiments with the oscillatory valve, a huge collection bucket is also placed on top of the tanks. A picture of this can be seen in figure B.7 included in the appendix. The bucket has a tube in the bottom that goes down into one of the tanks. For the experiments with the ball valve in place, a tube is connected directly to the valve and guides the water into one of the tanks.

4.1.7 IRIS facilities

Some equipment is also provided by IRIS. This includes the two pipes and some pumping equipment. An overview of the entire test facility is provided in figure 4.14

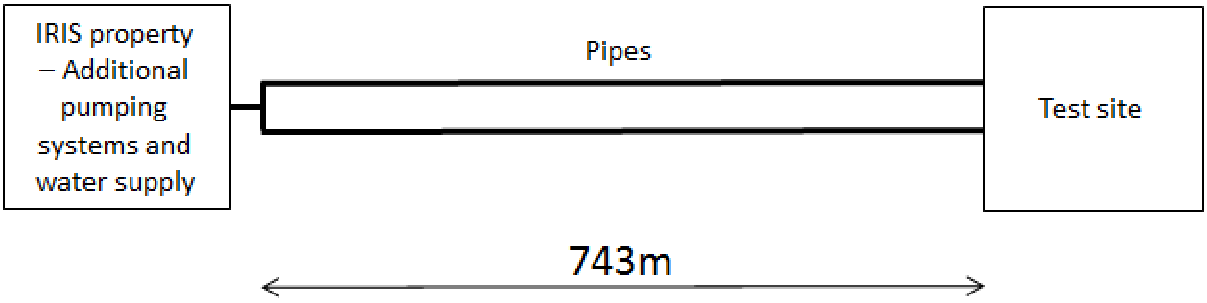


Figure 4.14: Overview of the entire test facility

Pipes

The pipes consist of two 743m long pipes made of carbon steel. The dimensions of the pipes are provided in figure 4.15. Along the pipes, there are various outtakes for venting and sensor installation.

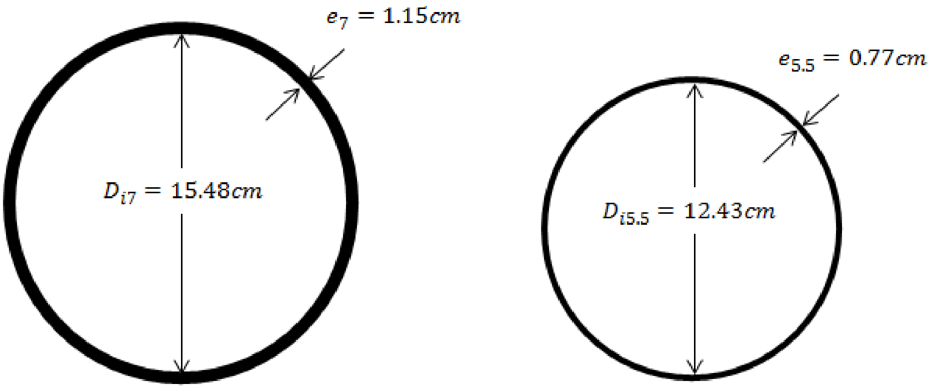


Figure 4.15: Pipe dimensions

Pumps

IRIS also has a few pumps at their disposal. It is necessary to utilise these pumps during filling and flushing of the pipes. One of the pumps is a very powerful triplex pump, capable of pressurising the system at more than 300bar. The flow rate available in the flow loop was at the time restricted to 1400l/s.

Chapter 5 Experimental Method

Since this is the first time these types of experiments are carried out in this test facility, it is necessary to establish some experimental procedures. This will ensure a more efficient process the next time experiments are conducted.

5.1 Preaparations

After installing all the necessary components, there are a few more preparations that are necessary to ensure good experimental results.

5.1.1 Pipe adjustment

The altitude varies along the pipes, as the pipes follow the landscape. Air is expected to gather in the peaks. To avoid this, the last 100m of the pipe upstream of the valve was lifted about 1m.

5.1.2 Procedure for air removal

Since the pipes are so long, there is always a possibility of having air entrapped somewhere in the system. As previously discussed, the air will have a noticeable effect on the outcome of the experiments. Excessive dampening of transients is of course not desired when doing experiments on transients, hence the desire to establish good procedures for its removal.

The air inside the pipes can be removed in two different ways.

- Venting of air at designated locations
- Flushing of the pipes at high flow rates

It is also possible to combine these measures.

Venting of air at designated locations

There are valves mounted along the pipe that can be opened to let air out of the system. It is also possible to mount the pressure sensors on the pipe with an extra valve to let air out at the sensor locations as well. In short, the procedure for venting the pipe in this way is a very simple four step process.

- Pressurise the system
- Vent at suitable locations
- Wait
- Repeat

The system can be pressurised in two different ways. By the powerful pumps owned by IRIS, or with the less powerful pumps used during the experiments. Both of these tactics were attempted, with varying results. The first venting was done by bringing the pressure in the system up to about 5bar and then closing off the system with all the valves to maintain the pressure. An overview of the valves can be seen in figure 5.1. Venting was then attempted at various locations along the pipe. The main venting was done at the two highest point on the pipes, as seen in figure 4.11. This was where the air was expected to settle. A lot of air was successfully vented during this initial venting, but the exact amount was not quantified. It later became evident that a lot of air was still present in the system, meaning that further action was needed. Note that when the pipes have been pressurised with the powerful triplex pumps, it is necessary to show caution when connecting the less powerful pumps back in the flow loop by opening valve 2 in figure 5.1. These pumps are not designed to withstand such pressures and can break. The pressure inside the pipe should be monitored, and if necessary lowered through other valves before connecting the pumps back in the system.

After this initial venting, the remaining air volume was in some way quantified. The system was brought from an internal pressure of a little over 1bar to an internal pressure of a little over 3bar. During this process, about 600 litres of water were pumped into the system, saying something about the remaining air volume. The 600 litres were measured by looking at the change in water level within the supply tanks.

Further venting through the same venting valves was performed many times throughout the experiments. At all these instances, the system was pressurised by the test pumps for the experiments. More often than not, some air was successfully vented. The amount, however, was very limited. The exact amount was not quantified, but the best guess would be in the range of a few litres for each time the pipes were vented. The most air was vented when the water in the pipes had been stationary for some time. This indicates that some of the air is carried along with the flow, and some of the air wanders slowly back to the highest altitudes when no flow is imposed.

Flushing of the pipes

There are two different goals to achieve when flushing the pipes. The first is to clean the pipes. The pipes are quite old and rusty and had not been used for a while when these experiments started. The second goal is to flush out the entrapped air from the system. Both of these goals can be achieved simultaneously. In figure 5.1, an overview over the different connections used while flushing is provided.

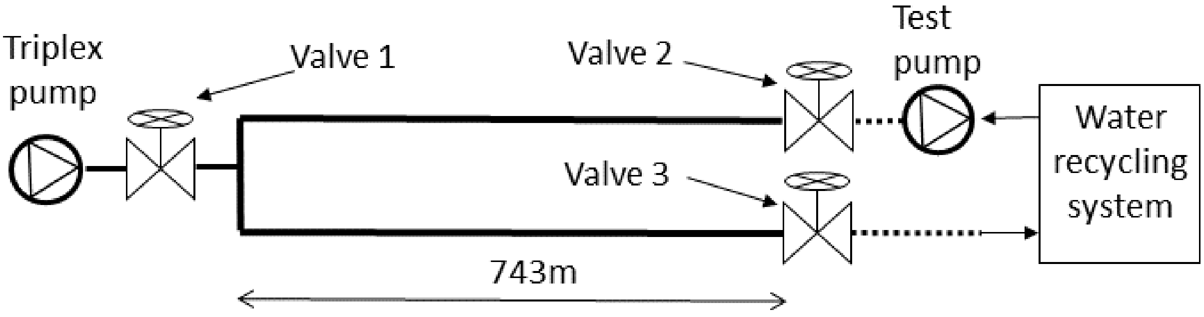


Figure 5.1: Overview of system setup during flushing

In total, the pipes were flushed three times. The initial flushing was done mostly at a flow rate of about 600 l/min for a prolonged period, and 1400 l/min for a very limited amount of time. For this flushing, valve 2 and valve 3 were connected with a hose, and the returning flow was deposited inside the IRIS facility. The primary purpose of this flushing was to clean the pipes and fill them up with water, with no focus on removing entrapped air. The second flushing had a greater focus on removing entrapped air, but at the time, no information about the current air content was available. Again, the pipes were flushed at a flow rate of 600 l/min for some minutes. The triplex pump was used for this flushing. Valve 1 was always open, while valve 2 was open while valve 3 was closed, and vice versa. After this, the first experiments were conducted in the test facility. The results from the experiments on the water hammer transient showed that too much air was still present in the system. It was at this time the first compression test was conducted. After unveiling that air was still present in the system, it was decided to do a third flushing of the pipes, but this time with higher flow rates for a prolonged time. This flow rate was 1400 l/min. After some minutes of flushing at this flow rate, it was observed that air started to come out in significant quantities. This flushing was maintained until no sign of air exiting the system was apparent from observation of the flow at the outlet. As a guide, the minimum amount of time this flushing is maintained should be the time it takes the flow to move from the triplex pump to the outlet on the other side. For a flow rate of 1400 l/min, the following flushing times can be calculated.

- Volume of 7" pipe $= 13.98m^3 \rightarrow T_{flush} = 10min$
- Volume of 5.5" pipe $= 9.01m^3 \rightarrow T_{flush} = 6min30s$

A subsequent compression test was performed, and this time only 60 litres was pumped into the system before the pressure stabilised, meaning that 90% of the air present before the flushing had been removed. Following experiments on the water hammer transient also showed that pressure waves propagated much easier through the system.

Following the third flushing, the pipes were vented a few times as previously described. The final compression test showed that only about 10 litres of water were pumped into the system before the pressure stabilised.

5.2 Conducting experiments

It is also important to establish good procedures for how the different experiments are conducted.

5.2.1 Procedure for water hammer experiments

The water hammer experiments are relatively easy to complete. It involves running the system with a steady flow rate and then closing the valve at the outlet end of the pipe. The valve should be closed quickly to achieve the greatest amplitudes in pressure. Great amplitudes are desired because of the dampening present in the system. The change in pressure will be in accordance with the Joukowsky head (later shown) if the wave speed is known and the valve is closed sufficiently fast. The flow should be regulated with the valve at the outlet end of the pipe. This is because a high static pressure directly upstream of the valve is favourable if cavitation is to be avoided. The greater the pressure loss through the valve, the higher the pressure upstream of the valve. Depending on the amplitude, the transient takes about 60s to dampen, so logging should be maintained for at least this period. It is also possible to monitor this during the experiments, to ensure that complete dampening is achieved, before saving the results.

An increase in static pressure following valve closure is expected. This is partly due to the change in flow conditions at the pump, giving increased head according to the pump characteristic. If this effect is not desired, the pumps must be shut off at the exact time of valve closure. This will also cause some dynamics in the system.

5.2.2 Procedure for experiments on oscillating flow

The experiments on oscillatory flow are a bit more extensive. They involve running the valve at various frequencies at constant flow.

Preparations

Before starting the experiments, some parameters should be estimated to create a reasonable plan for experiments.

- Frequency range
- Natural frequency
- Harmonics
- Amplitude of excitation
- Duration of data logging

The frequency range should be known because it determines what kind of rotating disc to utilise. The natural frequency and the harmonics should be known because these frequencies are the most interesting. If the experiments for some reason has a time constraint, the experiment can be done more efficiently by varying the step in the excitation frequency. This means having a fine step in excitation frequency around the natural frequency and the harmonics, and a bigger step for less interesting frequencies. The amplitude of the excitation is determined by the amplitude of the superimposed sine function on the disc and the adjustable slit outlet area. This configuration must be found by utilising the LVTrans model. It is also inefficient to maintain the same duration of the data logging for all the frequencies. The required duration decreases with increasing frequency. This is because a certain amount of similar periods are required for a good experiment. This amount remains constant and should be determined beforehand.

Running the experiments

When the experimental setup is complete, experiments should be initiated immediately and ran more or less continuously. This is because the valve is not tight meaning that there will be some leakage. This also means that the pumps should always be running to prevent suction pressures in the pipes that suck in air through the valve.

When starting the experiments, it is necessary to make sure that the signal is truly periodic before logging begins. This can be observed by monitoring the system in real time. Once the signal appears to be periodic, logging can start for the defined period. When the experiment is completed satisfactorily, the frequency can be adjusted according to the predetermined step in excitation frequency. This frequency is adjusted by the potentiometer that controls the frequency converter. When the frequency is changed, the system will take some time to adapt to the new frequency. This time will also decrease with increasing frequency. The stabilisation time is not specified here, but this can also be monitored in real time while conducting the tests.

While running the experiments, some water is lost due to spillage. Water is splashing around everywhere as the disc rotates. This means that the water level in the tanks should be monitored, and the tanks should be refilled when necessary.

The rotating disc might impose a safety hazard when it is running at full speed. Fragments of the disc might fly through the air at high speed if it suddenly breaks during operation. Standing directly next to the valve should be avoided. A person should always be in control of the frequency converter to shut the motor down if the disc suddenly breaks.

Parameter	Value
K	$2.15 \times 10^9 N/m^2$
E	$200 \times 10^9 N/m^2$
$e_{7"}$	$1.15 \times 10^{-2} m$
$e_{5.5"}$	$7.7 \times 10^{-3} m$
μ	0.3

Table 5.1: Pipe parameters

Changing disc

To enter a higher or lower frequency range, it is sometimes necessary to change the disc. While this is done, there will be more leakage than normal. To make the change easier, the pressure should be lowered to a minimum while still ensuring that no air is sucked into the system. This can be achieved by running only a single pump at a low speed.

Finishing experiments

When the experiments are finished, it is necessary to seal of the slit to avoid leakage. This is done by removing the disc and sealing the slit. The valves in both ends should then be closed, and the pumps shut completely down.

5.3 Experimental analysis

It is also important to establish methods for analysing the results. The analysis in this thesis is focused on evaluating the flow loop performance.

5.3.1 System parameters

It is necessary to obtain some system parameters to get accurate results from the design simulations. The required parameters can be calculated with formulas, but some parameters in the formulas are not under control. This may give inaccuracies in the calculated parameters, so the calculated parameters should be validated by experiments.

Wave speed

The wave speed can be calculated by formula 2.4, inserting the parameters provided in table 5.1.

This result can be validated by several experiments. One way is to measure the time delay a sensor experiences for the arrival of an elastic pressure wave following a sudden valve closure. The distance between the sensors is also measured, and by assuming a constant wave speed, the calculation is trivial. This approach is inspired by similar work done by Lari Kela and Pekka Vähäoja [8]. Figure 6.2 and 6.1 illustrates this approach.

In practice, this method was not as straightforward as in principle. In general, the pressure wave appeared to be more dampened, and not as sharp and distinct as desired. For the 5.5" pipe, the arriving pressure wave appears much more dampened, and the time of arrival is not entirely clear. Therefore, the peaks in the pressure signal are used instead, as illustrated in figure 6.2. The data here contained a lot of noise, so the higher frequencies of the signal have been filtered away in this plot.

The wave speed can also be found by investigating the amplitudes of the water hammer transient. If the amplitudes of the pressure wave are known, the wave speed can be calculated by manipulating the formula for the Joukowsky head 2.8.

Natural Period

Once the correct wave speed is obtained, calculation of the natural period of the system can be done by formula 2.6. This formula requires a known wave speed for the system. The wave speed is previously found both analytically and experimentally. However, it is not known whether the obtained wave speeds are accurate for the entire system, or just smaller sections of the pipes. It is possible to find the natural period of the system experimentally, as a verification of the already obtained wave speeds. It can be found by bringing the system to a steady state with some constant flow rate and then closing the valve at the outlet end of the pipe rapidly. This induces a water hammer transient with a frequency equal to the natural frequency of the system. This approach is illustrated graphically in figure 6.5.

Head Loss

It is relevant to know something about where the losses occur throughout the system. This comes in handy in later simulations and for designing a more efficient system concerning hydraulic losses.

The losses are determined experimentally. This experiment involves running the pumps at a constant speed, yielding a steady flow through the system. The pressure was logged at all the sensor locations throughout the system. The data was then processed. This involves finding the pressure loss between each of the pressure sensors while correcting for the change in altitude. The friction factor f_L can be calculated by formula 2.9. The roughness for each of the pipes can also be found by solving the Colebrook (2.10) equation numerically.

5.3.2 Analysing gas content by investigation of water hammer transient

In addition to the analysis on wave speed and natural frequency, it is possible to say something about the presence of free gas in the pipe by investigating the water hammer transient. This is shown by a comparison between the results from the water hammer experiments before and after flushing of the pipes. The frequencies of the results can then be compared. Any deviation will indicate a difference in wave speed and air content. It is also interesting to observe to what extent the water hammer transient is dampened throughout the system as a result of varying air content. This analysis should also be used to say something about the general influence of the presence of air as free gas, and to what extent it is a major problem.

5.3.3 Standing waves

Standing waves are an interesting wave phenomenon that can tell a lot about the system dynamics. The standing waves are found by investigating the results from the experiments on oscillatory flow. The amplitude in the pressure oscillations at each of the sensors are calculated. The amplitude is then investigated for all the sensors at the natural frequency and some of the harmonics. The standing wave can be illustrated by plotting the amplitude in pressure oscillations as a function of the pipe length, as seen in figure 2.1 and figure 2.2.

Chapter 6 Experimental Results

6.1 System parameters

The system parameters have been subject to both analytic and experimental analysis.

6.1.1 Calculated system parameters

Wave speed

The wave speed in the different pipes is calculated using formula 2.4. For water at ordinary temperatures, the wave speed is a = 1440m/s, but this is expected to drop when the flow is confined within a pipe.

The correction coefficient c_1 is calculated using formula 2.5. This yields the following wave speeds for each of the pipes:

- Wavespeed for 7" pipe, $a_{7"} = 1371 m/s$
- Wavespeed for 5.5" pipe, $a_{5.5"} = 1354 m/s$

Pipeline period

The determined wave speed can be used to calculate the total period of the pipeline, which is given by formula 2.6.

- $T_N = 4.363 s$

This also yields the limit for valve closure time T_R, which is half the natural period of the system.

6.1.2 Experimentally obtained system parameters

Wave speeds

The wave speed can also be found experimentally. The approach for this is shown in figure 6.1 and 6.2.

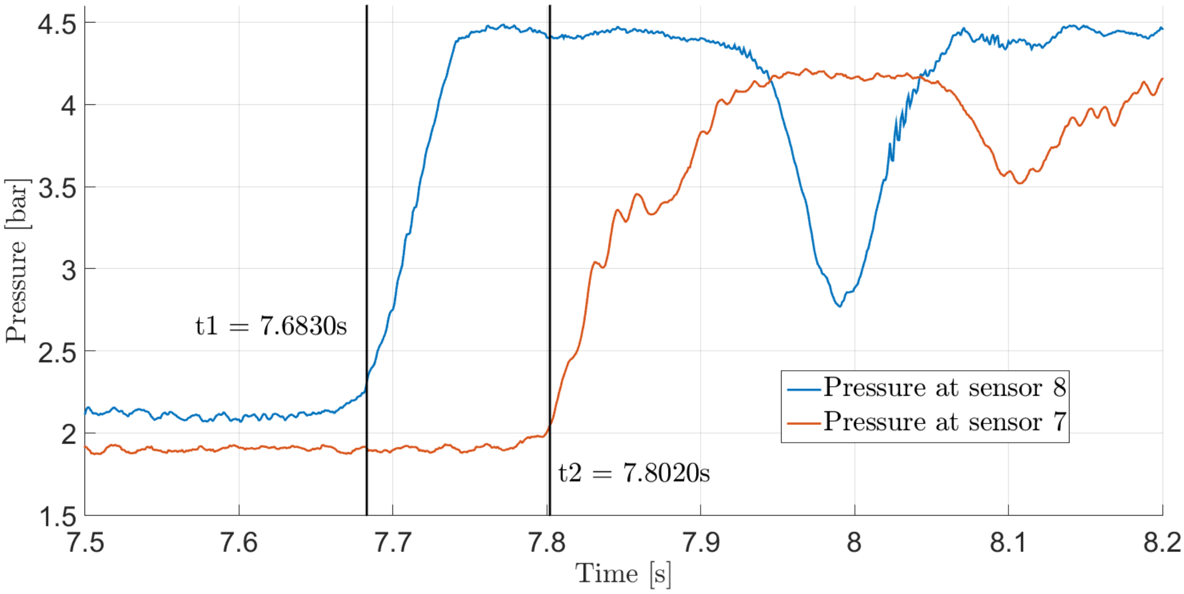


Figure 6.1: Comparison of the time of arrival for the elastic pressure wave at two different sensors in 7" pipe

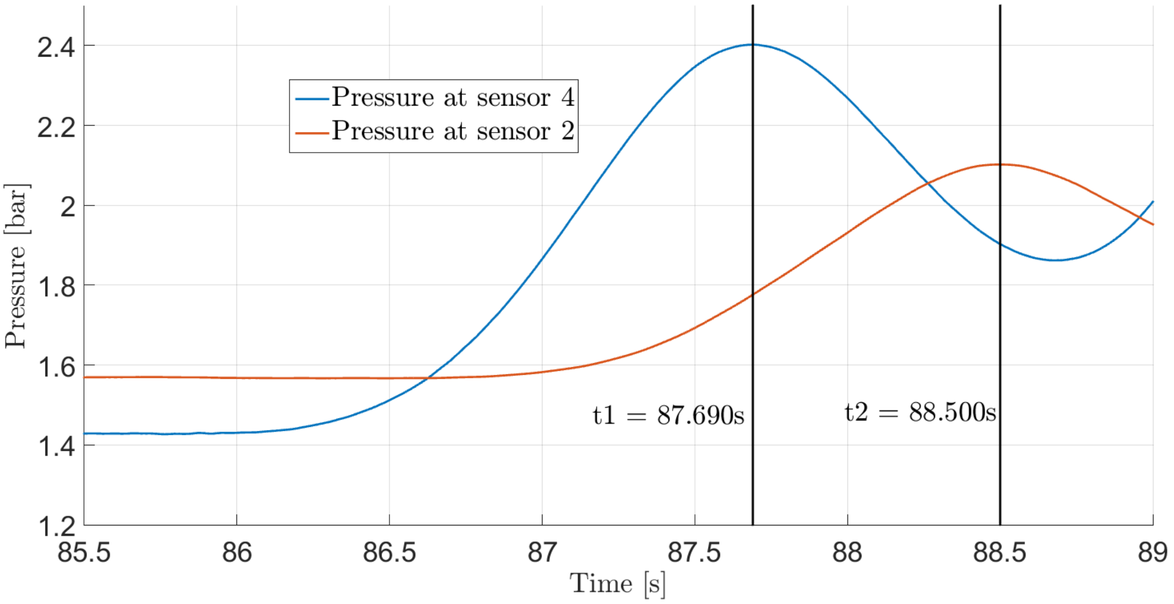


Figure 6.2: Comparison of the time of arrival for the elastic pressure wave at two different sensors in 5" pipe

For the 7" pipe, the time difference is found to be 0.119s, and the distance between the sensors is 158.3m. This yields a speed of sound, a = 1330.3m/s. This is not too far from the calculated speed of sound for this pipe.

For the 5.5" pipe, the time difference is 0.810s, and the distance between the sensors is 537.1m. This yields a wave speed of 663.1m/s, much lower than the calculated wave speed.

The typical time development of the pressure at the valve in the 7" pipe, following the valve closure is shown in figure 6.3. It is also compared to the Joukowsky head for validation of the determined speed of sound.

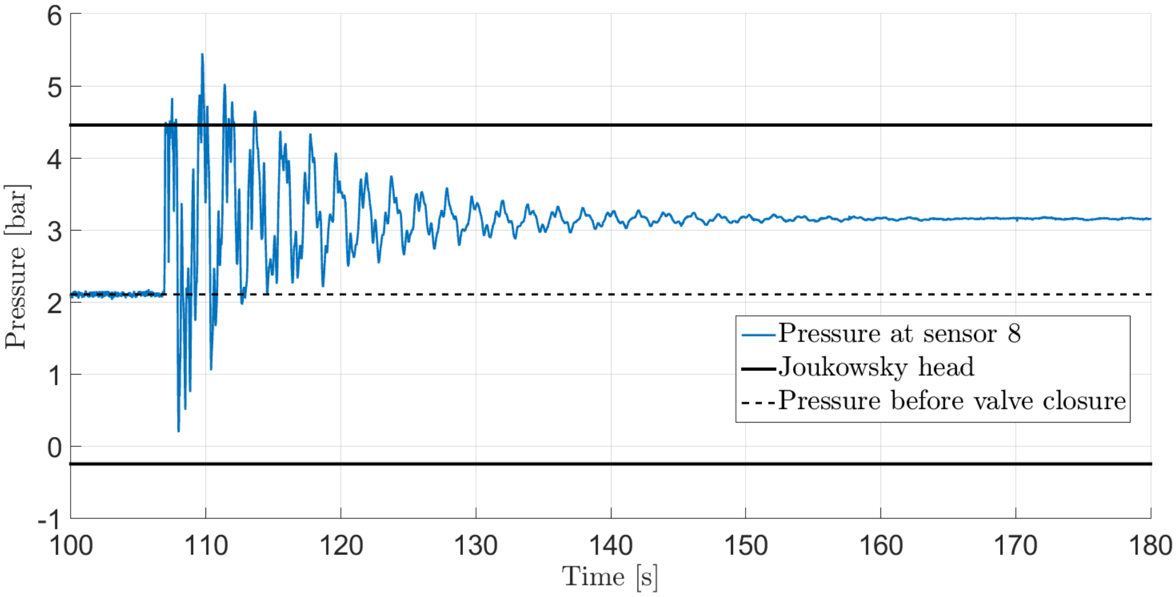


Figure 6.3: Water hammer at 200 l/min compared with calculated Joukowsky pressure

The pressure builds up to 4.482bar immediately after valve closure. The calculated Joukowsky pressure is determined to be 4.456bar if a wave speed of 1330.3m/s is assumed. If the wave speed is calculated from the obtained pressure amplitude, a wave speed of 1345.3m/s is found. The second peak in the pressure is observed to be higher than the first peak. The water hammer transient is completely dampened out after 60s.

The typical time development of the pressure at sensor 3 following the rapid valve closure is shown in figure 6.4. The pressure should build up to 4.753 bar if a wave speed of 663.1 m/s is assumed, but this does not occur. The first pressure amplitude only builds up to 3.65 bar. The following build up in pressure is caused by the pump.

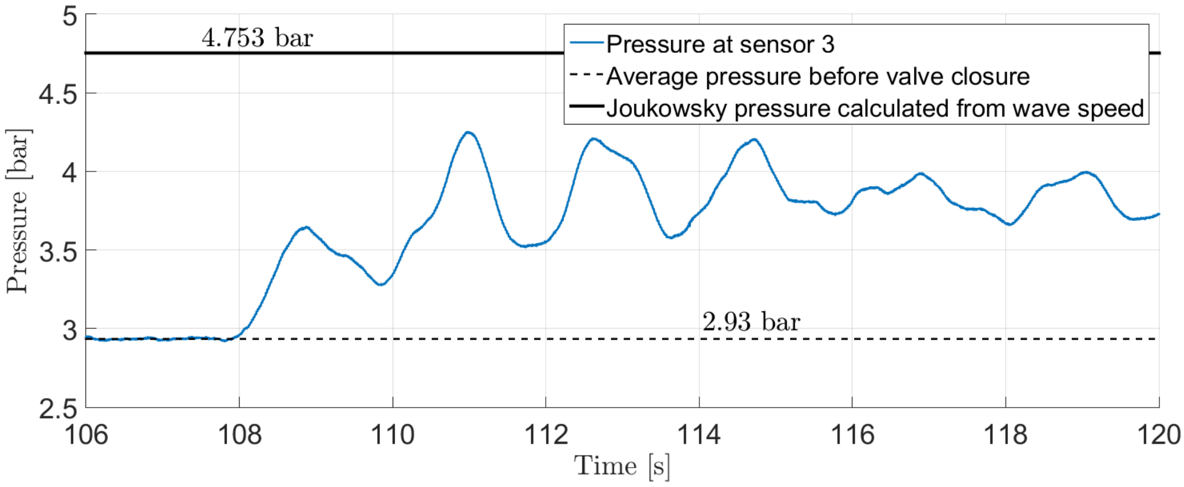


Figure 6.4: Water hammer at 200 l/min compared with calulated Joukowsky pressure

Pipeline period and natural frequency

The pipeline period can also be found experimentally. Following a rapid valve closure, the frequency of the water hammer transient will be the same as the natural frequency of the system. The result from such an experiment can be seen in figure 6.5.

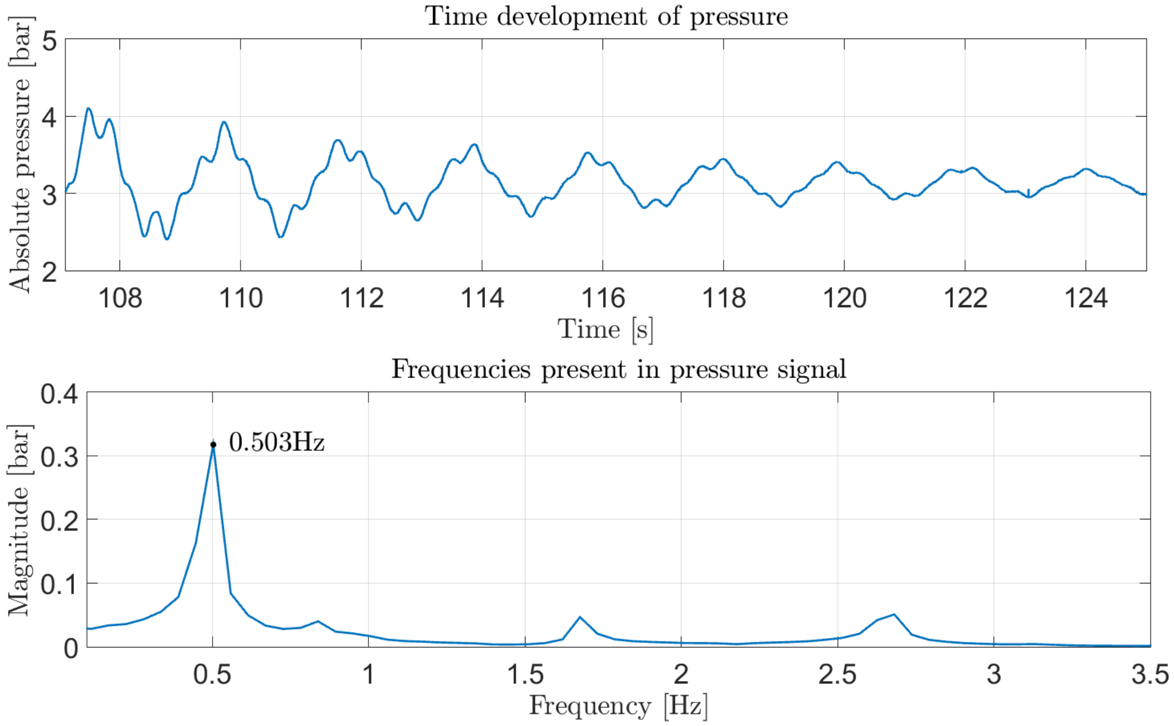


Figure 6.5: Water hammer transient and its frequency domain transformation

A natural frequency of 0.503Hz yields a natural period $T = 1.99s$. This does not correspond to the calculated natural period.

6.1.3 Head loss

The head losses through the system were found experimentally. The total head loss in the experiment was 7.65m. The distribution of the head loss through the system is shown in table 6.1

Stretch of flow loop	Head loss [m]	Portion of total head loss [%]
P1-P2	2.01m	26.27%
P2-P3	0.30m	3.89%
P3-P4	2.56m	33.51%
P4-P5	2.43m	31.75%
P5-P6	0.11m	1.42%
P6-P7	0.13m	1.74%
P7-P8	0.11m	1.43%

Table 6.1: Where in the flow loop head loss occurs

From the same experiment, the roughness of the pipes was obtained.

Pipe	Roughness ϵ
5.5"	0.0874mm
7"	0.0470mm

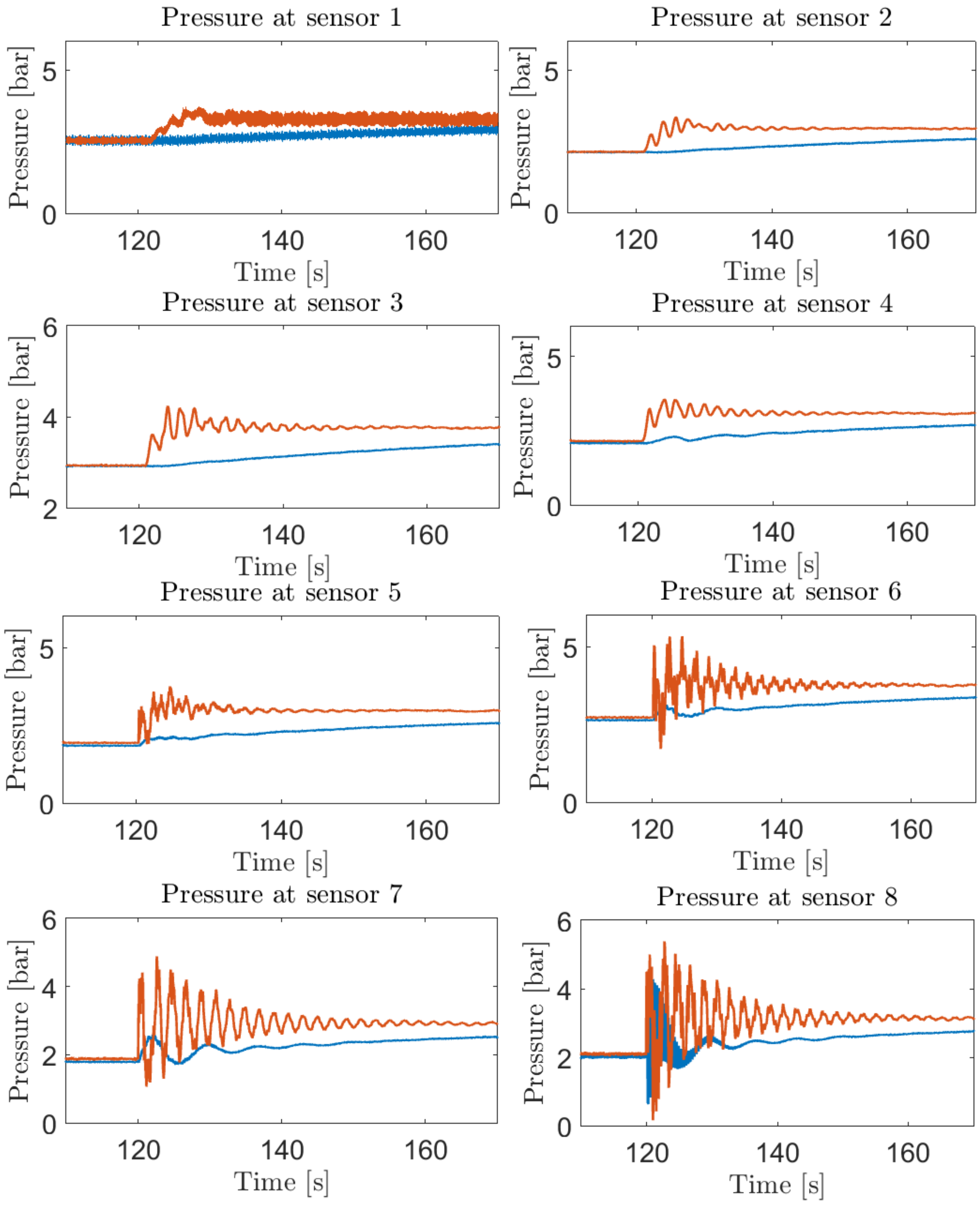
Table 6.2: Absolute roughness for each of the pipes

6.2 Water hammer

6.2.1 Effect of free gas on the water hammer transient

The water hammer experiments were performed with two widely different internal conditions inside the pipe. Some of the experiments were done before the final flushing while some were done before the final flushing. A comparison between these two flow conditions is illustrated in figure 6.6.

The most evident trend is that the pressure waves are dampened much sooner when a greater amount of air is present in the system. From the blue plots(before flushing), one can observe that the water hammer is almost completely dampened before it reaches sensor 6. In the red plots(after flushing) the water hammer is preserved throughout the system. It can also be seen that the pressure buildup in the system caused by the pumps after the valve closure is much slower for the blue plots. Another difference becomes obvious when investigating the difference in frequency in the pressure signal at the valve in the time following the sudden valve closure. The frequency of the signal before flushing is 0.136Hz while the frequency of the signal after flushing is 0.498Hz.


Pressure at sensor 1

Pressure at sensor 2

Pressure at sensor 3

Pressure at sensor 4

Pressure at sensor 5

Pressure at sensor 6

Pressure at sensor 7

Pressure at sensor 8

Figure 6.6: Water hammer at 200 l/min. Blue plots are before flushing, red plots are after

6.3 Verification of LVTrans model

In figure 6.7, the simulated head at the valve is compared to the experimental results.

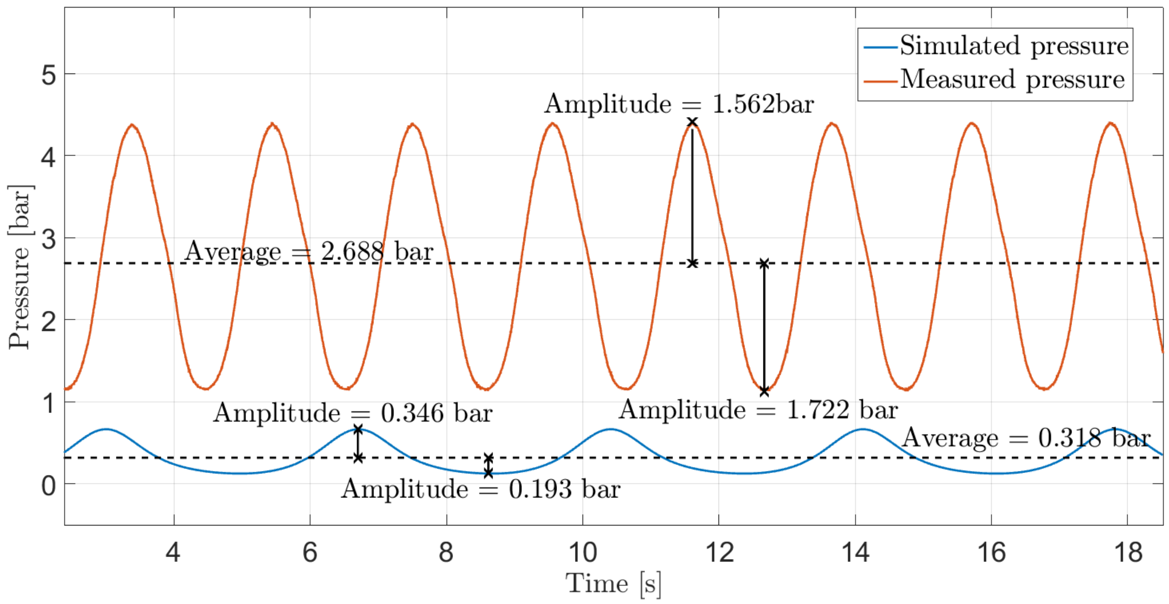


Figure 6.7: Comparison of simulated and measured head conditions at valve

The result shows a huge deviation between the simulated and the experimental results. Possible reasons for this are discussed in the next chapter.

6.4 Oscillatory flow

An example of a result from the experiments on oscillatory flow is shown in figure 6.8.

Some things are worth noting when studying these plots. When it comes to frequency, it is evident that the excitation frequency is present in all the plots. There is also another frequency of 0.722Hz present in all the signals, but this is only obvious in the plot for sensor five. Here it is the dominant frequency. The presence of other frequencies distorts the sine wave to a varying degree throughout the pipes. For sensor 1, a set of higher frequencies is also present. These are frequencies from the pumps. The amplitudes of these frequencies are effectively dampened before reaching sensor 2. The amplitudes also vary a lot through the system, indicating the presence of a standing wave. Note that the positive and negative amplitudes vary for all the plots.

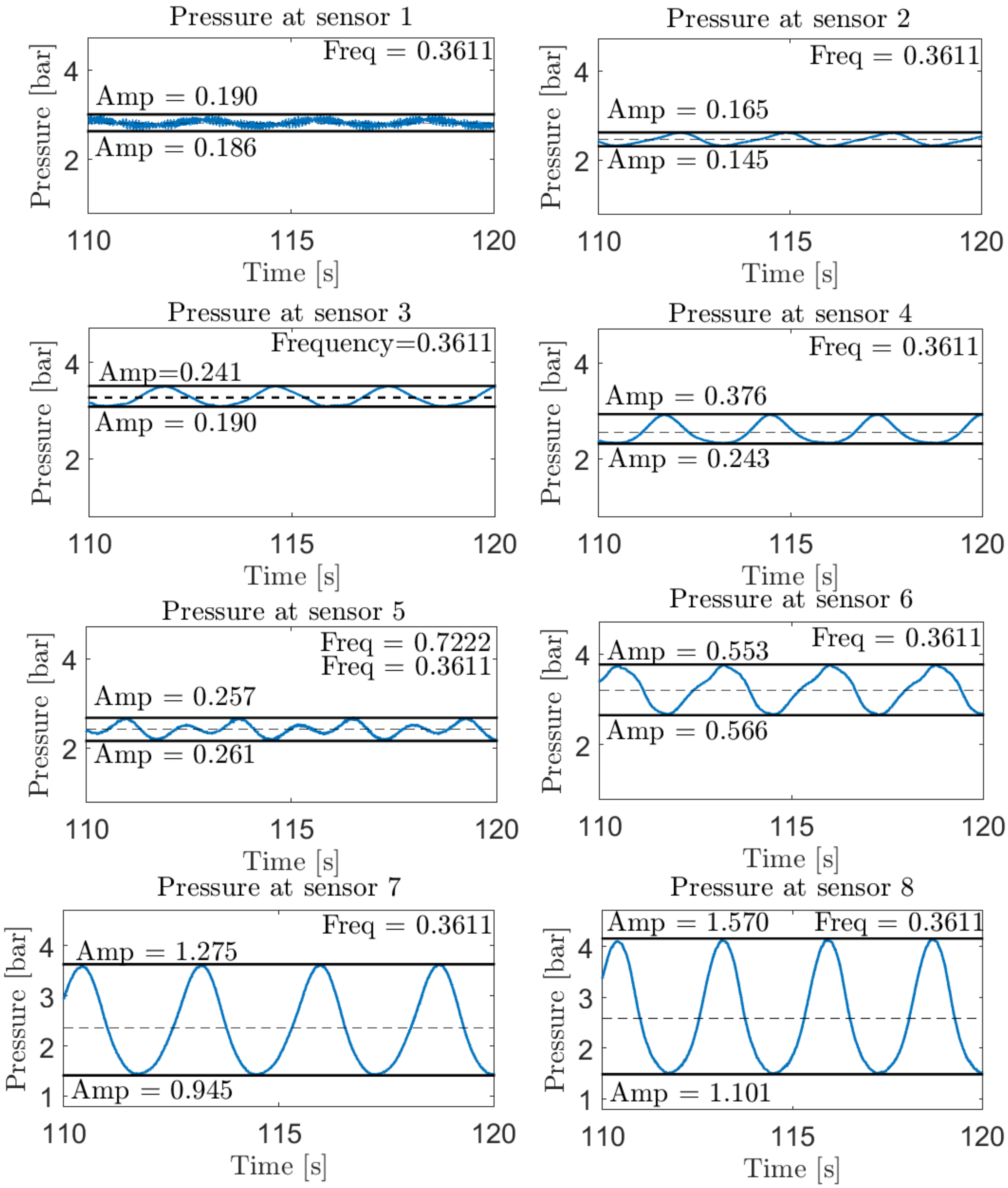


Figure 6.8: Overview of the results from the oscillatory flow experiments. The excitation frequency is here 0.3611Hz

6.5 Standing waves

The presence of standing waves was expected in the system, and these were expected to change in a certain way when the system was excited at various frequencies. The development of the pressure amplitudes directly upstream of the outlet as the frequency is varied around the natural frequency of the system can be seen in figure 6.9.

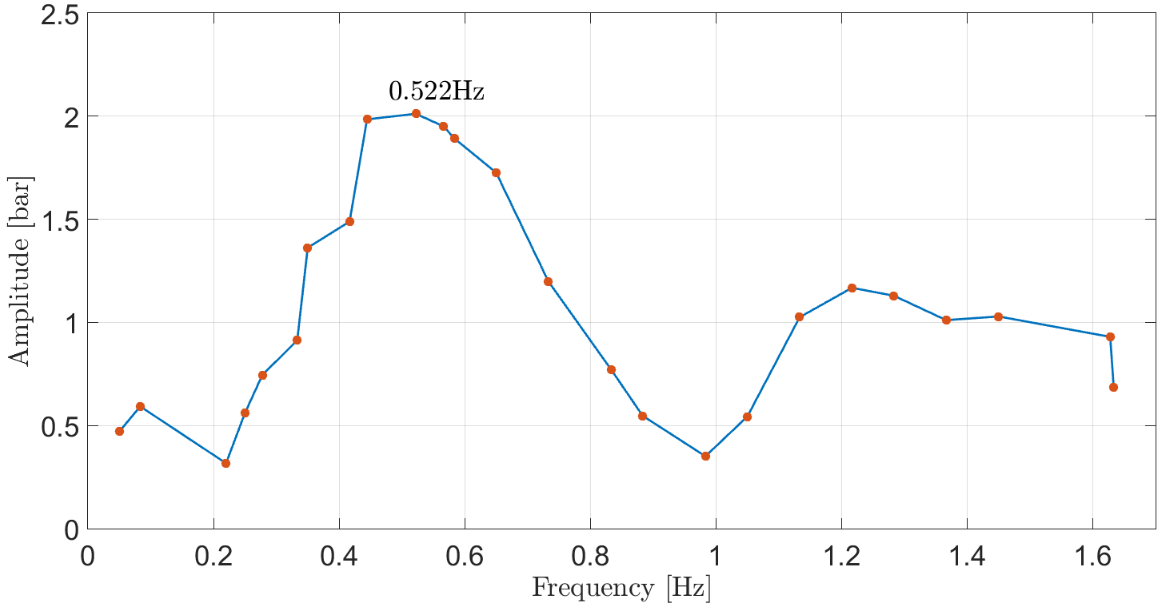


Figure 6.9: Effect of standing wave close to valve

Figure 6.10 illustrates how these standing waves develop for the lower frequency range throughout the pipe, not only at one sensor.

There are standing waves present in the 5.5-inch pipe too but of much smaller amplitude. The natural frequency of this pipe seems to be different from the natural frequency of the 7" pipe, The pump also has a much greater influence here, yielding a very different shape to the different harmonics. Due to the smaller amplitude of the standing waves, the harmonics are much less distinct in this pipe. A complete picture of the standing waves at the investigated frequencies are shown in figure B.10 in appendix B.

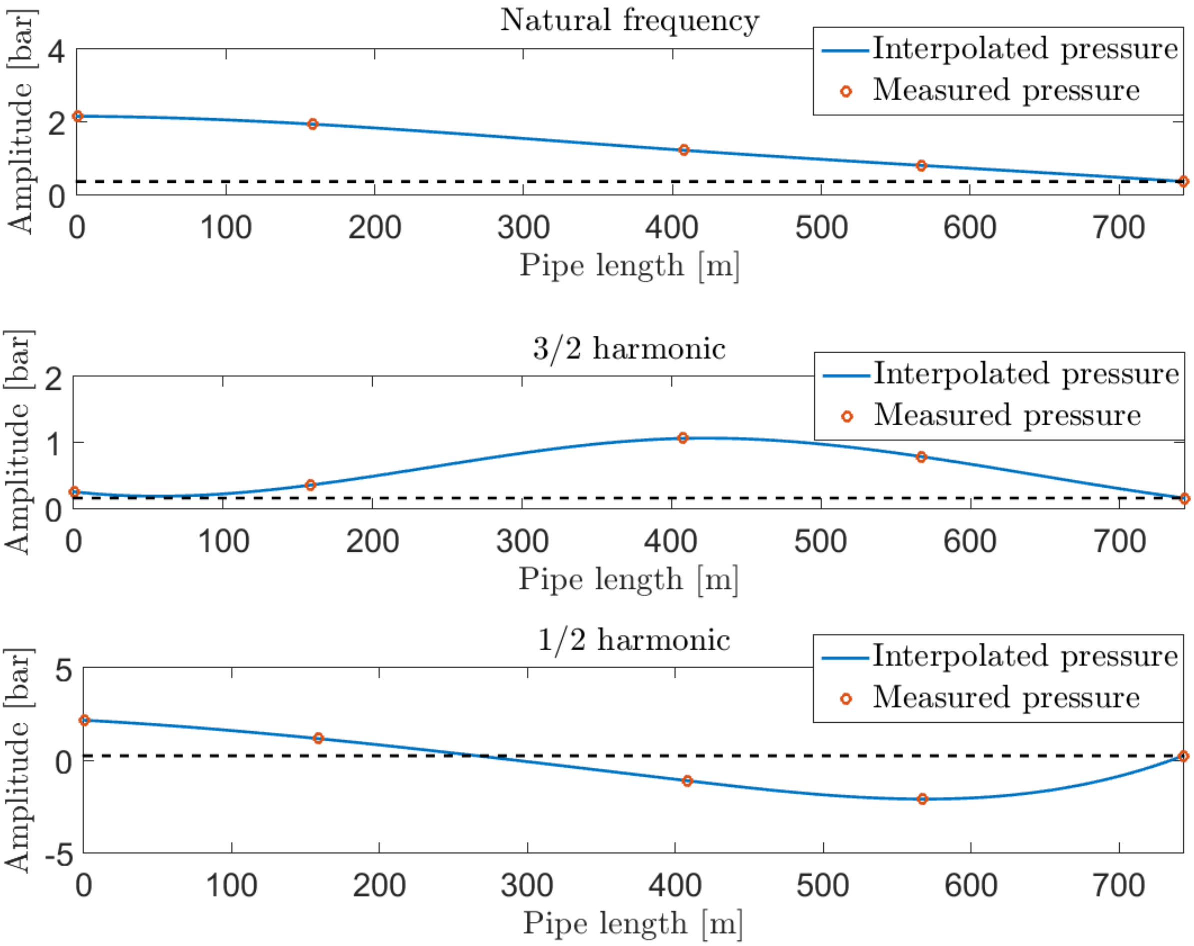


Figure 6.10: Development of frequencies at first harmonics for 7" pipe

Chapter 7 Discussion

7.1 LVTrans model

The experimental setup up was partly based on numerical simulations in both LVTrans and FloMASTER. The results from the LVTrans simulations and the measured data deviates by a lot. As seen in figure 6.7, the measured head oscillations are much bigger for the experimental data than for the simulations. Since the results provided by LVTrans are based on the method of characteristics (which is assumed to be exact), the deviation cannot be explained by numerical error. The deviation must, therefore, be explained by wrong parameters, or that something is not included in the model. The large deviation is believed to be caused by an unexpected reflection point, that will be discussed later. This causes the system to change fundamentally, causing entirely different dynamic conditions in the system. This makes the simulations, and the experiments incomparable as the results in practice represent two different systems. It is hard to model such behaviour accurately in LVTrans (or FloMASTER) as it is very far from an ideal and general case. One suggestion for improving the model is to leave out the 5.5" pipe and only focusing on the 7" pipe.

7.2 FloMASTER model

The results from the FloMASTER simulations could unfortunately not be verified, due to excessive damping of the transients through the system. However, the FloMASTER model has been very useful for identifying the need for an accumulator, asserting the pipe connection as a reflection point and for validation of experimental methods. In general, it has been useful for investigating the general dynamic behaviour of the system. The FloMASTER model can also be used for this purpose in the future.

7.3 Experimental setup

It is necessary to evaluate how well the experimental setup performed. This is important to establish a foundation for further improvements of the experimental setup.

Valve

Two different modes of operations were required by the valve.

The ball valve for generating the water hammer transient worked very well. It was easy to operate, and any problems related to the results were not caused by the valve itself.

The rotating valve was a bit more challenging, both during rig up and operation. The biggest problem during rig up is the leakage through the outlet slit. When the PE1000 disc is in place, the valve is not leakage proof. There is no way to shut off the water at the valve, except draining off the entire flow loop.

The valve itself worked very well during operation. However, some minor problems also arose that should be fixed before future experimental work in the flow loop. These problems did not interfere with the completion of the experiments but caused some undesired breaks and inaccuracies.

The governing of the rotational speed of the disc was a bit problematic. This was due to some bad adjustments of the rotary potentiometer governing the frequency converter. The frequency could be varied from 0.5Hz to 100Hz by this rotary potentiometer. A step of about 0.25Hz was desired to obtain a sufficient resolution in frequency around the natural frequency and some of the harmonics, but this resolution proved to be very hard to obtain in practice with the rotary potentiometer. Changing the frequency by 0.25Hz involved rotating the potentiometer a very very small increment, which was almost impossible to do by hand. A different solution should have been chosen for this.

Adjustment of the slit outlet area also proved difficult. The plan was to physically measure the dimensions of this area after inserting the adjustment pin. This proved difficult due to all the water spilling out through the slit and a different method for adjusting the slit outlet area should be considered. The best solution would be to measure before there is water in the system.

An error message concerning overheating of the motor was displayed on the frequency converter when the motor was operated in the lowest range of speed for a prolonged period. The motor itself showed no sign of overheating, meaning that it did not feel hot at all. This error message was suspected to be caused by some software that displays this warning when the motor is running at low speeds for a prolonged period, instead of physical temperature measurements. The motor should feel hot if it is at risk of overheating.

The performance of the valves based on the system effects they generate are discussed in a later section.

Pump

The main job of the pumps were to supply a flow rate that yielded the necessary magnitude of excitation. This was achieved in full. In general, the pumps worked great for the initial tests. If more flow or pressure is desired, more pumps of the same type can be added in different configurations. This can be relevant if it is desired to dissolve more air in the water by increasing the pressure. This, of course, has some practical upper limit, imposed by e.g. the mechanical strength of the pump housing. The pumps imposed some pressure oscillations on the system, but these high-frequency signals were efficiently dampened before having any observed effect on the experimental results.

It might be favourable to have more control over the pumps when future scientific experiments are performed. At the moment, there is no way to know anything about the pump speed. This is valuable information that should be monitored during the experiments. The flow was sometimes inexplicably changed during the experiments, effectively corrupting some of the results. The reason for this was that one of the pumps suddenly shut down, or changed its speed. This was not obvious by just listening to the pumps, as they both were very loud, but would be obvious if the speed was monitored.

Accumulator

There was always some air present in the system, effectively dampening out a lot of the pressure waves before they could reach the accumulator. This means that its real performance was never put to the test. However, the pumps did indeed operate at a relatively steady speed. An indication of the effect of the accumulator is shown in figure 4.8. Here, it is obvious that the amplitude in the discharge from the pumps is much smaller than the flow measured by the flow meter. This means that the amplitudes are indeed dampened by the accumulator, ensuring more stable operation of the pump.

Pipe connection

The pipe connection was based on existing components already available at the IRIS facility. This component's effect on the dynamic behaviour of the system was underestimated during the design phase but proved to be critical during the testing at the flow loop. Later analysis will reveal that it effectively served as a reflection point for the elastic pressure waves propagating through the system. This was expected, but the magnitudes of the reflected waves greatly exceeded the expected value. This means that further action is necessary to get rid of the unwanted effects from this component.

Measurement system

The measurement system was composed of the pressure sensors and the flow meter.

The system for measuring the pressure worked very well. No room for improvement was identified during the tests.

The flow meter was shown to have some latency, but this can be corrected by comparison with pressure measurements in the same part of the flow loop. It might have been better to position the flow meter between the accumulator and the pumps, to measure the actual discharge from the pumps.

Water recycling system

The water recycling system worked well for the operation during the water hammer experiments.

The operation of the rotating valve involved a lot more spillage. The collection bucket worked to some degree but was not able to collect all the water. This meant that refilling of the tanks was necessary during these experiments.

The connection between the tanks and the pumps could have been solved better. It was expected that the water level would even out between the tanks, but this was not the case. This was inconvenient as it was only possible to fill one tank at a time. A new configuration for this connection is suggested in the next chapter.

Pipes

The pipes worked well for these experiments. The roughness of the pipes is relatively low, keeping the head loss low even for long distances. The mechanical integrity of the pipes is never even close to being challenged, even when resonance frequencies are imposed on the system.

On the downside, it is very challenging to remove the air entrapped in the system completely. Due to the length and the curvature of the pipes, it is hard to exactly locate where the air is trapped. Procedures for coping with these challenges have been suggested.

7.4 Experimental procedures

It has been very important to establish experimental procedures to make sure that good results can be achieved.

7.4.1 Preparation of flow loop

Before the experiments could begin, some preparations were necessary.

Removing air

Venting of air at designated locations in the flow loop was one of the possible measures for removing entrapped air from the system. There were two drawbacks with this technique. All of the air was not successfully vented at once, meaning that many iterations are required. With the required waiting to let air move to the highest peaks in the pipe, this process becomes very tedious and is not viable as the only measure to remove air. There was also no suitable venting valve on the smallest pipe at the second peak away from the pump/valve end of the pipes. This means that all the air cannot possibly be vented out by this technique alone and it is also believed to be the cause of the higher air content in the smaller pipe.

The prolonged flushing of the pipes at high flow rates proved to be an efficient measure for removing the air from the pipes. This was observed by the behaviour of the water/air jet exciting the system during flushing, as well as in the results from the subsequent experiments. However, the flushing measures taken during this initial testing was not sufficient to remove all the air from the system. This means that this method must be improved before future experiments in this flow loop.

7.4.2 Performing experiments

Procedures during the experiments are also important to obtain good results efficiently.

Water hammer

The procedure for the water hammer experiments is relatively straightforward. The established procedure worked great for the initial tests, so no amendments are required.

Oscillatory flow

In general, the procedure for the experiments on oscillatory flow worked well. A relatively extensive amount of good data was acquired in a short amount of time. However, the procedure itself can be made more efficient. This can be achieved by making a preliminary plan for the step in frequency and duration of data logging for each excitation frequency. The operation of this valve imposed a small safety hazard for people in the vicinity of the valve. Measures to minimise this hazard are suggested in the next chapter.

7.5 System parameters

Obtaining the correct system parameters is important for the analysis of the experimental results, for the evaluation of the current experimental setup and the future work in this flow loop. Some deviating results concerning these parameters are obtained, hence the need for evaluation of both the method linked to the acquisition of the system parameters and the results.

7.5.1 Wave speed

There is some uncertainty linked to the calculated wave speeds used in the simulations. As mentioned, the presence of free gas has a massive impact on the wave speed, shown in figure 2.5. This is hard to compensate for with the provided formulas, without knowing the actual air content. This means that an experimental approach is necessary to accurately determine the system parameters.

The experimental results obtained by the method proposed by Lari Kela and Pekka Vähäoja [8] are considered to be valid only to some extent. For the 7" pipe, this method is considered to be quite accurate. It is easy to locate the time of arrival of the elastic pressure wave. There are however some limitations in the accuracy. The measured distance between the sensors is not exact and the time of arrival for the elastic pressure wave is not determined exactly. The same uncertainties are also valid for the wave speed in the 5.5" pipe, but some additional uncertainties are also present here. Since the pressure waves appear so dampened, the more distinguishable local maximums are used for comparison instead. However, simulations in FloMASTER cannot confirm that this approach is viable. The basis for this is shown in figure B.1 in appendix B. This simulation indicates that the peaks represent the same moment in time, but this is not the case in the experimental results. This leads to the conclusion that this method is highly questionable when using the peaks for comparison. Verification by investigation of the Joukowsky pressure is necessary for both of the pipes, but especially for the 5.5" pipe. This verification will be discussed later in this chapter.

The obtained wave speed for the 7" pipe matches the calculated wave speed quite well. This wave speed is only valid for a section of the pipe, and further analysis will reveal changes in the wave speed throughout the 7" pipe. The wave speed obtained for the 5.5" pipe deviates by a lot from the calculated wave speed. Although the method behind this result is questionable, it is clear that a greater quantity of air is present in the 5.5" pipe.

7.5.2 Natural period

The natural period of the system has also been subject to both analytic and experimental analysis. The results show that the natural period found by formula 2.6 and the natural period found experimentally does not coincide. This is expected, as the actual wave speed of the system is different from the one calculated beforehand. The following should be noted about what the natural period obtained for the analytic and experimental approach means for the average wave speed through the system.

- The average wave speed from formula 2.4 is 1362.5 m/s
- The average wave speed from experimental analysis is 2989.8 m/s

A wave speed of 2989.8 m/s is simply not possible for a pipe filled with water. The wave speed in pure water is about 1440 m/s, and when the flow is confined within a pipe, the wave speed will always be lowered. It is, however, possible that the pressure wave reflects almost entirely off something different than the inlet, yielding a completely different natural period. This might be the case if a pocket of air is present somewhere in the system, serving as a point of reflection. An air pocket will greatly affect the wave propagation, as it greatly affects the impedance of the fluid. Since it is known that air is present in the system, it is believed to be the cause of the unexpected natural period. Due to the location and the geometry of the pipe connection (figure 4.10), it is suspected that air may be entrapped in e.g. one of its bends. Some verification of this can be done by a quick simulation in FloMASTER, by inserting an accumulator in the pipe connection to simulate an air pocket entrapped here. The reduced period is observed in the simulations, confirming the possibility of an air pocket in the pipe connection. The setup for this simulation is shown in figure B.2, and the resulting water hammer transient is shown in figure B.3 However, the period does not match exactly, but is off by a few percent. This is assumed to be caused by uncertainties in the actual wave speed of the pipes.

Another result that confirms the natural period found experimentally is the period given by the maximum in pressure amplitude observed in figure 6.9. Here, one can observe the amplification in amplitude as the excitation frequency is approaching the natural frequency. This is due to the buildup of energy in the system, known as resonance. This resonance is expected to give a maximum in pressure amplitudes at the valve for the natural frequency, and this also seems to be the case. The natural frequency was found to be 0.503Hz by investigating the frequency of the water hammer transient, and the maximum amplitude of the frequency response plot is at 0.522Hz. The small deviation between these results can be explained by the fact that the frequency was not adjusted continuously, but with steps of about 0.08Hz.

7.5.3 Head Loss

The head losses appear to be the biggest in the small pipes between the pump and the larger diameter steel pipe, and in the connection between the two pipes. These are minor diameter sections of the flow loop, so this is expected. However, that the portion of the total head loss was, in fact, this big was a bit surprising. The found roughness of these pipes is in the range of what is expected for new pipes of this kind. This means that the condition of the pipes is better than expected. Ideally, several experiments should have been done with a bigger variation in flow rate to verify this result. If the same roughness of the pipes had been found for various flow rates, it would confirm the calculated roughness of the pipes.

7.6 Water hammer

In general, the water hammer transient is quite far away from the clean signal shown in figure 2.4. This is due to a lot of reflected pressure waves from zones of different impedance throughout the pipe. The air content is established as the cause of these impedance variations. The many oscillations in the signal also indicates that the air content is changing throughout the pipe.

7.6.1 Effect of flushing

Investigation of the water hammer transient can be used to say something about the air content in the pipes. The difference between the results of the experiments on the water hammer transient before and after flushing is evident. It should be established that it is, in fact, the air present in the system that is the cause of this change. There are mainly two different arguments that can be used to verify this. First, a change in natural frequency is observed. This can be caused by a change in the wave speed, that can be explained by a change in the air content within the pipes. The second argument is the observed delay in the pressure buildup after valve closure. It makes intuitive sense that when a greater amount of air is present in the system, it takes more time to build up the pressure. This is because air is much more compressible than water, and it takes some time for the compression to complete. This indicates that it was indeed air that was the cause of this behaviour, and that the flushing of the system had much of the desired effect.

7.6.2 Verification of speed of sound

The experimentally obtained wave speed was found to be 1330.3m/s for the 7"-pipe. The theoretical Joukowsky pressure matched the actual amplitude for the initial pressure wave quite well. If the speed of sound is calculated from the obtained pressure amplitude, a wave speed of 1345.3m/s is found. This is quite close to the obtained wave speed from the other experiment, indicating that the method by Lari Kela and Pekka Vähäoja [8] works to some degree. However, the deviation is still substantial. In one way, the Joukowsky pressure approach seems more practical when it comes to determining the actual wave speed in the system. It is much easier to identify the amplitude in the first peak than to determine the instantaneous time of arrival of the elastic pressure wave. On the downside, this method only accounts for what happens in between the sensors to some extent. It is not given that the wave speed remains constant, and a change in wave speed will cause a reflection. This is seen in figure 6.3, as a rapidly changing pressure signal with many peaks. In theory, it is possible to read the wave speeds throughout the pipes from the magnitudes of these peaks. However, this signal simply contains too much noise to extract the necessary information about the wave speed throughout the pipe. This means that a combination of the two methods is necessary to obtain an accurate result.

When the Joukowsky pressure approach is used for sensor 3, placed in the middle of the 5.5" pipe, the wave speed is determined to be 260.6m/s. This does not correspond very well with the other approach. This is probably due to variations in air content throughout the pipe. With the limited amount of pressure sensors. it is not possible to obtain sufficient data for accurate evaluation of the wave speed in this pipe. Another observation that is made is the fact that the second pressure peak is higher than the first. This is because the static pressure in the system is increased due to the change in flow conditions at the pump. The zero flow will now cause the pump to increase the head according to the pump characteristic.

7.7 Oscillatory flow

Good results are obtained from the experiments on oscillatory flow. The excitation frequency is present throughout the system, and the amplitudes of the system response greatly exceed the required magnitude, even with low flow rates. The presence of a signal with exactly double the frequency is assumed to be caused by the introduced reflection point in the pipe connection. The good results also confirm that the rotating valve worked as intended and that the pumps provided the required flow rates.

7.8 Standing waves

Standing waves are shown to be present in the system. The results show that they behave as expected in the 7" pipe, while being heavily dampened in the 5.5" pipe. This also indicates a greater air content in the 5.5" pipe. The development of the standing waves confirms an important behaviour in the system. It was expected that there would be a node in the pump end of the system and that the standing waves would develop from there. The results indicate that this node is located in the connection between the pipe instead. This means that the pressure waves are reflected here instead of the at the pump. This confirms the suspicion raised in the section concerning the natural frequency and speed of sound, concerning the wave reflection in the pipe connection.

Chapter 8 Recommendations for future work

8.1 Experimental setup

Valve

A few measures should be taken to ensure smooth and continuous operation of the rotating valve. The potentiometer should be replaced with one suitable for the required increment in frequency. To avoid leakage, a system should be developed that makes it easier to seal off the slit when the valve is not operating. This can be achieved by inserting another valve (e.g. a ball valve) directly upstream of the rotating valve. The manufacturer of the motor should be contacted to verify that it is safe to neglect the error message that is warning about the motor overheating.

A small concern for the safety of people moving in the vicinity of the rotating valve when it is rotating at high speed has been discussed. A measure to increase the safety is to build a box around the valve, ensuring that splinters of a broken disc are contained.

Pumps

To create a complete pump characteristic and to obtain more control of the pump operations, it would be interesting to measure the pump speed in some way. This would also make it possible to monitor the pumps during operation, effectively detecting undesired instabilities. One possible solution is to measure the speed of the pump shaft connected to the impeller by a tachometer. This requires some modifications to make the shaft visible. Another solution could be to record the sound of the pump motor. Some frequency analysis of the resulting signal can reveal the speed of the motor. This method needs further investigation before implementation

Accumulator

There is no reason to believe that the accumulator is not working as designed, although it was not possible to confirm this during the initial tests. No further improvements are

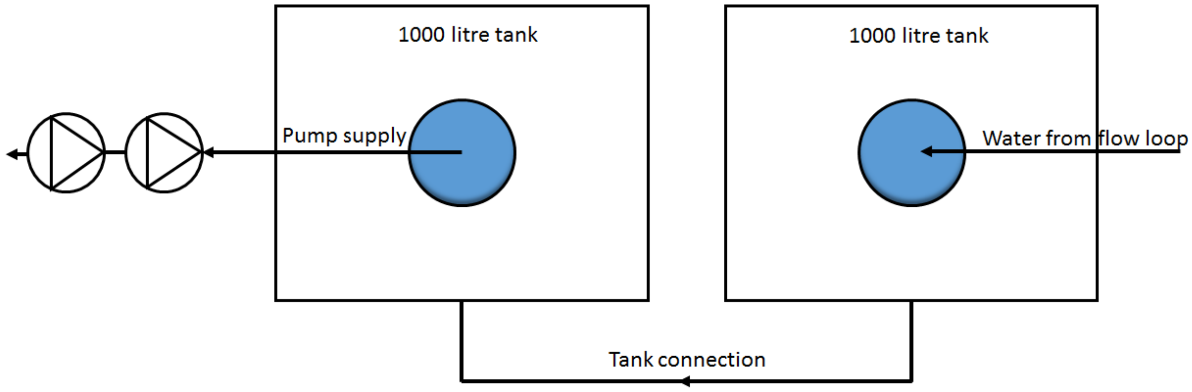


Figure 8.1: Improved configuration for water recycling/supply system

suggested.

Pipe connection

Two different improvements are proposed for this component.

The reflection point in this components can be minimised by various measures. The unnecessary bends should be removed, to reduce the risk of entrapped air at these locations. The diameter should be increased to 5.5" to avoid unnecessary reflections from a change in the area.

Another solution may be to embrace this component as a reflection point but making it a lot better defined. This can be achieved by inserting an accumulator at this location. This will effectively enforce a well-defined reflection point. The accumulator can also be used to vent out air entrapped in the system at this location. The 5.5" pipe can then be used as a supply line, while all measurements are done in the 7" pipe.

Measurements system

The measurement system worked very well. The only thing to be considered is the placement of the flow meter.

Water recycling system

The way the pumps are connected to the supply tanks should be changed to the configuration shown in figure 8.1. This will make sure the water level in the tanks levels out automatically.

Chapter 9 Conclusion

The literature study shows that transient flows in pipes are still a field of study that needs further research. By investigating previous work done on transient friction, it becomes clear that this is an important subject that still requires more research to come up with models that are both simple and accurate. Such models are important in both the design and operation of hydropower plants, hence the shown interest by the hydropower research community. In a field of research in such a state, it is important to provide useful experiments for validation of different models. As already mentioned, the scale of this experiment is presumably unrepresented in previous research, implying that it will serve as a significant contribution to the further development of this subject. The literature study also shows that frequency response measurements are a powerful tool in such experimental work.

The models created in both LVTrans and FloMASTER have been compared to the experimental results. The results from the LVTrans model did not match the experiment, but for obvious reasons. It is believed that only small changes in the model will make the model more representative of the psychical system. The FloMASTER model was used for identifying the need for an accumulator, and for its design. Unfortunately, it was not possible to verify that it was working as intended. This was due to excessive damping at other locations in the flow loop, making the accumulator less important. Despite the inaccuracies in the numerical models, simulations have been crucial in the design of this flow loop. The resulting pump solution consisting of the pumps, water recycling system and the accumulator, worked well for the initial tests. Possible improvements have been proposed.

The pipe connection was not properly designed on beforehand, but only based on existing components at the IRIS facility. Its significance was underestimated as it was considered a passive component in the flow loop. This was a mistake, as the analysis has revealed that this component is more important than first assumed. Several measures for improving this solution has been suggested.

The ball valve for generating the water hammer transient worked very well. No problems have been identified with this component. Hence no improvements are suggested. The rotating valve was much more complex than the ball valve but still worked very well. Some practical problems have been detected, but all of these can be fixed with relatively simple measures.

The described measurement system worked very well. More pressure sensors would be preferable, as it would make more detailed analysis possible. This is not feasible because

IRIS does not allow for more holes to be drilled in their pipes. The flow meter showed some deficiencies, but these can be compensated for by combining the flow measurements with pressure measurements. Relocating the flow meter is advisable to get accurate information about the pump discharge.

Several techniques for analysis of the flow loop have been developed. The analysis has successfully revealed the dynamic behaviour of the system. The analysis has been important for the evaluation of the flow loop performance and should be done in combination with future experimental work in the flow loop.

The project has been managed well. All deadlines have been kept, no unforeseen incidents have occurred, and the project has been completed according to the initial plan. All the involved parties seem satisfied with the status of the project at this time.

In general, the initial tests went well. Much required experience from the operation of the flow loop was acquired, and necessary experimental procedures for further work has been developed.

With the improvements suggested in this thesis, the flow loop is considered to be suitable for further experimental work.

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Appendices

Chapter A Nepal Paper

Proceedings of the International Symposium on Current Research in Hydraulic Turbines

CRHT – VII

April 04, 2016, Turbine Testing Lab, Kathmandu University, Dhulikhel, Nepal

Paper no. _ _ _

Planning, start-up and testing on a pipe flow loop for the investigation of transient characteristics

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Abstract

The purpose of this project is to aid in the design of a large scale test facility for experiments on transient flows, more precisely transient friction and the effect of free gas. Two long pipes have been devoted to this purpose, but more components are needed in order to initiate the experiments. This project has investigated the design of several of these components. In order to do experiments on transient in pipe flows, some mechanism is needed to excite the flow. Two different designs have been developed. One is well tested in similar experiments, and only minor adaptations are made for this project. The performance of this mechanism is believed to be sufficient for future experiments. The performance of the second excitation mechanism is not known from previous experiments, so in this project it is subject to further analysis by means of CFD. The transient characteristics are first simulated in the 3D CFD software ANSYS, and then implemented in the pipe transient software Flowmaster. This yielded the system response. This analysis showed that the proposed design needs further work and adaptation in order to be implemented. Contributions are also made for the choice of pump, water recycling system and measurement system. Some system characteristics, such as the first harmonic, are also calculated.

Keywords: transient flows, experiment, free gas, transient friction

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1. Introduction

A good understanding of transient flows in pipes and ducts is essential to ensure safe operation of hydropower plants. Forced transients can be used actively to improve this understanding by means of frequency response measurements. Flow Design Bureau AS will facilitate experiments involving such transients in a flow loop located outside of Stavanger, Norway. This flow loop is about 1400m long, yielding ideal conditions for big scale experiments. Many good models exist for transient effects such as transient friction in pipes, but good experimental data on this scale is scarce. The purpose of this project is to design an experimental setup that can be used in other studies on different aspects of transient flows in pipes. The experimental setup should be suitable for experiments on transient friction losses and the effect of free gas on transients. To achieve this, a few more components must be designed or selected. A model of the flow loop will also be created in a 1D-simulation software for transient pipe flows (such as Flowmaster or LVTrans).

2. Theory

2.1 Background

IRIS, the International Research Institute in Stavanger, are in the possession of the mentioned pipes. These pipes are perfect for large scale experiments on transients in pipes, but need some additional components in order to complete meaningful experiments. A schematic drawing can be seen in figure 1. This is a principle sketch, and not necessarily how the final design will appear. The pipes are 700m long, with a pipe diameter of 7 and 5.5 inches. Pressure measurements will be performed at several locations throughout the flow loop. The fluid flowing inside the pipes will be water that may contain some impurities.

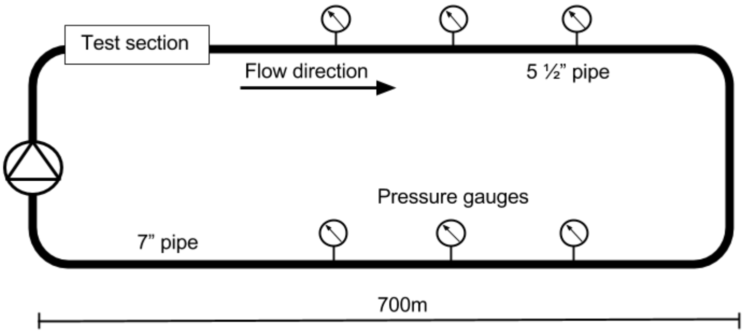


Figure 1: Schematic drawing of the test facility

2.2 Theoretical background

2.2.1 Transient friction

The Moody chart does not predict the damping for oscillations of varying frequency. This effect is indeed present, meaning that some kind of friction dependence must be developed. A lot of research has been done on this subject. [p. 43, 1]

2.2.2 Effect of free gas

The presence of free gas in the fluid will affect the propagation of pressure waves throughout the system. This effect is determined by how the gas is distributed throughout the system. It can either be evenly distributed as gas bubbles, or as pockets of air. This will either cause a change in the speed of sound through the medium, or serve as a reflection point for the pressure waves.[p. 9, 4] Effectively, the amplitude of the dynamic pressure waves will be reduced due to these effects. This is because of two different effects, called attenuation and scattering. Attenuation is damping due to the compressibility of the gas bubbles, while scattering is damping due to the change in impedance as the pressure wave moves through regions of different wave speeds. [p. 12, 2]

2.2.3 System period and natural frequency

The period of a pipeline system is the time it takes for a pressure wave to propagate back and forth through the pipeline. This is also known as the reflection time.

$$T = \frac{2L}{a} \quad (1)$$

The critical period for the first harmonic of a pipeline system is twice the period of the system. The systems will have several harmonics that will be apparent when exposed to increasing frequency. [p. 15, 2]

2.2.4 Courant Number

The Courant number is a measure of how many cells the fluid passes through in one time step in transient simulations. In order to choose an appropriate time step, the Courant number must be examined. The Courant number is calculated as: [3]

$$C = \frac{u\Delta t}{\Delta x} \leq C_{max} \quad (2)$$

3. Method

The following section will explain the process behind the choice and analysis of the different components, as well the calculation of system characteristics.

3.1 Determination of excitation mechanism

It is necessary to excite the flow in two fundamentally different ways. These two different ways include excitation in the form of a rapid valve closure to generate water hammer and excitation in the form of a sine wave oscillation of flow and pressure. Since these two flow excitations are so fundamentally different, it was decided to consider two separate excitation mechanisms.

3.1.1 Water hammer excitation

The water hammer excitation mechanism does not need to be very complicated. The only thing required is that it can shut down the flow in a time shorter than the reflection time of the system, $T_c < T_R$. The reflection time of the system is quite long, because of the extensive length of the pipes. No further design was needed here, as standard components will satisfy the requirements.

3.1.2 Oscillatory flow excitation

The oscillatory flow excitation mechanism is more complex. This mechanism must be able to excite the flow as a sine wave over a wide range of frequencies, and at the same time be practical to implement in the given system. Two different concepts are considered here. One is implemented within the pipe, while the other is to be implemented at the outlet of the flow loop. Two different approaches for determining the design seemed reasonable here. First, other scientific work was evaluated to see if any other successful excitation mechanisms had been developed by others previously. This turned out to be the case for the excitation mechanism at the outlet of the flow loop. This excitation valve can presumably be implemented with no fundamental changes to the design. Some previous work had also been done on the internal excitation mechanism, but this work needed more adaptation and analysis in order to be implemented in the flow loop in question. This was done by modelling the design and performing a CFD analysis in ANSYS CFX. The transient characteristics obtained from this work was implemented in Flowmaster, and the system response was evaluated. The response was then evaluated against certain demands regarding the magnitude of the amplitude of the excitation.

3.2 Determination of pump

Some considerations must be taken into account when choosing the pump. It must be able to deliver the necessary flow rates for the experiments, withstand pressure waves, especially those generated by the water hammer, while also fulfilling some restrictions in the power capacity enforced by the test facility. It is also important that the pump does not generate any unwanted frequencies in the system, that might interfere with the experiments in some way. Cost is also a limitation in this process. A simulation of the head loss for different flow rates was done in order to determine the necessary pump capacities.

3.3 Measurement system

It is necessary to measure two different variables during the experiments. These include pressure at various locations as well as the flow rate throughout the system. The flow meter can be more or less be arbitrarily placed, but some restrictions are enforced on the locations of the pressure sensors. There is only a limited amount of pressure taps available throughout the system, and no more can be installed in the existing pipe. This is to avoid problems with the pipe integrity in later experiments with much higher pressures.

3.4 Water recycling system

There will be used a lot of water during these experiments, all of which might be discharged to the atmosphere, depending on whether an open or closed loop system is chosen. However, it seems both unpractical and unnecessary to spill all that water. Hence, a water recycling system necessary. Briefly explained, such a system will consist of a storage tank, as well as a mechanism to capture the water discharged to the atmosphere, and redirecting it to the storage tank. At the time of writing, the design of this system is not yet completed.

3.5 Determination of system characteristics

Some system characteristics were also necessary to ensure desired operation of the experiments. This includes the natural frequency of the system, the already mentioned pressure loss, and a complete frequency response of the system.

3.5.1 Natural frequency

One way of determining the actual natural frequency/period in the model of the system, is to simulate a valve closing at the end of the flow loop. In practice, this is done by inserting a valve in the Flowmaster network, and closing it instantly at a specific time using a tabular controller. The natural period of the system can be found from the period of the pressure waves. This procedure also serves as a verification measure of the theoretical expression

3.5.2 Frequency Response

This can be achieved by the use of LVTrans. The flow loop must be modeled including all the components, and a suitable range of frequencies are enforced on the system. The response is recorded, and illustrated using a Bode plot. At the time of writing this procedure is not completed, hence the results are not included.

4. Numerical Setup

In the following, it is explained how the numerical model for the performance of the internal excitation mechanism is developed.

The procedure for setting up a problem is very different in ANSYS and Flowmaster. This is mainly due to the fact that ANSYS is a 3D CFD software, while Flowmaster is a 1D pipe dynamics software. The governing equations are different as well.

4.2 ANSYS

Shortly described, the process of solving a CFD problem in ANSYS consists of the following steps. Modelling fluid domain to implement geometry, meshing of domain, setting up boundary conditions and solver settings, solving, and finally post processing the results. All of these aspects cannot be explained in detail here, but the following section will give a quick overview over the setup for the simulation.

The domain is specified as shown in figure 2.

Some general remarks worth noting are:

- The domain was meshed using the standard ANSYS meshing tool
- The time step was chosen based on the Courant number (FORMELREF)
- The inlet was specified with a constant velocity of 2.3m/s while the outlet was specified as a pressure outlet with zero relative pressure. Turbulence intensity was set to 5% at the inlet.
- SST was chosen as the turbulence model
- Higher order schemes were used for both time and space

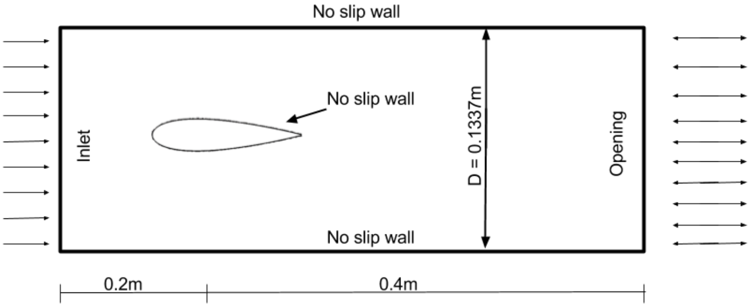


Figure 2: Computational domain

In the simulations, the excitation is achieved by the change in angle of attack of the hydrofoil. This movement was governed by the following equation:

$$\alpha = \alpha_0 + \alpha_{amp} \sin(2\pi f t)$$

where f is the excitation frequency, α is the angle of attack, α_0 is the default angle of attack and α_{amp} is the amplitude in the angle of attack.

It was desired to run several simulation varying both the default angle of attack and the amplitude in the angle of attack. However, only one such simulation was successfully completed. This was a simulation with $\alpha_0 = 20^o$, $\alpha_{amp} = 20^o$ and $f =$ natural frequency.

4.3 Flowmaster

Setting up a problem in Flowmaster is different from ANSYS. In Flowmaster, it is necessary to create a so called network, based on predefined components such as pipes and valves. The components are then dragged and dropped into a workspace and linked together. Necessary data is specified for each component. Solving is done by the method of characteristics.

The Flowmaster model was developed using actual dimensions and components. Drawings and other details were provided by IRIS. The characteristics are not obtained directly from ANSYS, they have to be calculated from other variables. The drag force working on the excitation mechanism is suitable for this, and can be calculated with an expression written in CEL, the CFX Expression Language. The pressure drop can then be calculated using a control volume analysis on the obtained drag force. Once the characteristics of the excitation mechanism from ANSYS are obtained, they should be implemented in the Flowmaster model to investigate the system response.

5. Results

4.1 Excitation Mechanism

4.1.1 Internal oscillatory flow excitation

The proposed internal excitation mechanism is shaped as a NACA0021 hydrofoil, fitted inside the pipe. A shaft runs through the center of the foil and out through the pipe wall. To ensure sine motion of the excitation mechanism, a circular disk is fastened to the shaft outside the pipe wall. Another shaft is hinged to this disk and a crank shaft. The crank shaft is driven by an electric motor. This concept of creating oscillatory motion is illustrated in figure 3.

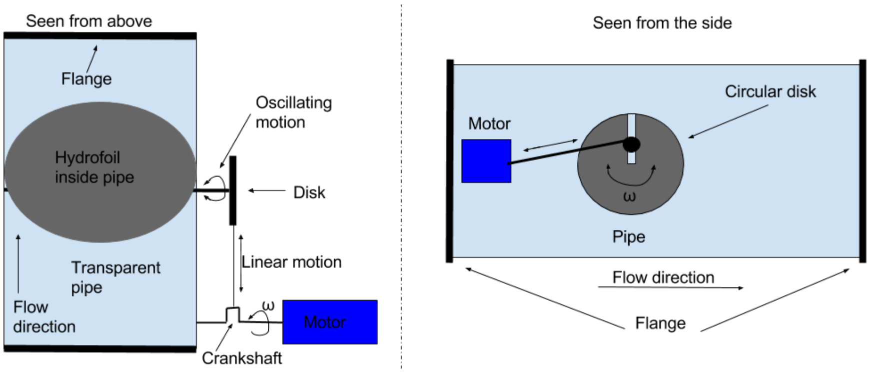
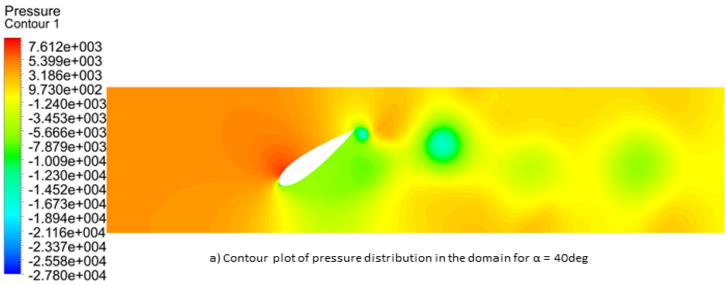
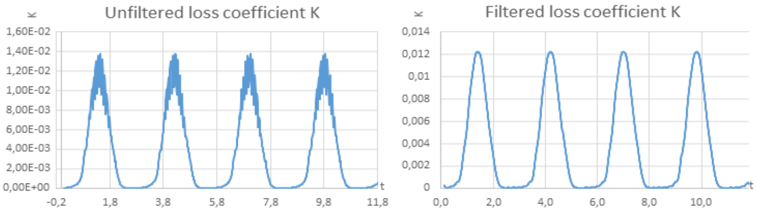


Figure 3: Concept drawing of internal excitation mechanism

This excitation mechanism had not been previously tested, hence it was desired to do some CFD to investigate its performance. The CFD analysis with the described numerical setup yielded the results shown in figure 4.

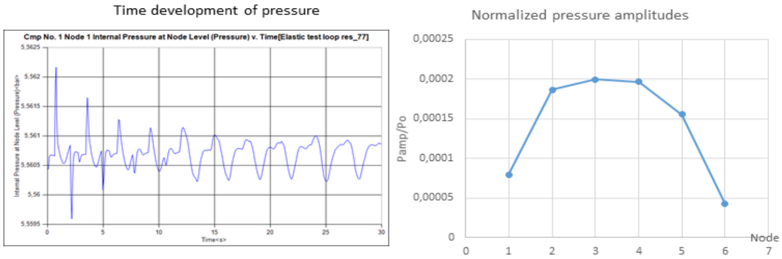


a) Contour plot of pressure distribution in the domain for α = 40deg



b) Time development of the unfiltered loss coefficient

c) Time development of filtered loss coefficent



d) Typical time development of the pressure at a given location in the flow loop. Here directly downstream of the excitation mechanism

e) Normalized pressure distribution through the flow loop

Figure 4: Results from ANSYS simulation

In figure 4a, one can see a qualitative illustration of the pressure distribution in the domain, for the default angle of attack. The excitation mechanism now starts in this position, and oscillates up and down. High frequency oscillations in the characteristics had to be filtered away in order to successfully implement these characteristics in Flowmaster. The derived loss coefficient is shown in figure 4c. In figure 4d, a typical time development of the pressure downstream of the excitation mechanism is shown. Here, one can see that after some initial transient behaviour, the solution develops in an oscillatory manner. However, not as a perfect sine wave. The normalized pressure amplitude reaches a maximum at about the half-length of the flow loop, but only reaches a value of about 0.005% of the pump pressure. This is far below the desired magnitude of excitation.

4.1.2 External oscillatory excitation mechanism

The proposed excitation mechanism placed at the outlet is more or less exactly the same as Svingen [p. 83, 1] has used for flow excitation in his experiments on transient friction. This mechanism forces the flow out through a slit in the cross section of the pipe outlet. A rotating disc is placed so that it partly closes the slit. The disc is shaped as a circle with a superimposed sine curve. When the disc passes over the slit, it ensures that also the outlet area varies as a sine function. This also applies to the flow throughout that very same area. Several discs will have to be made, with different numbers of sine periods with various amplitudes.

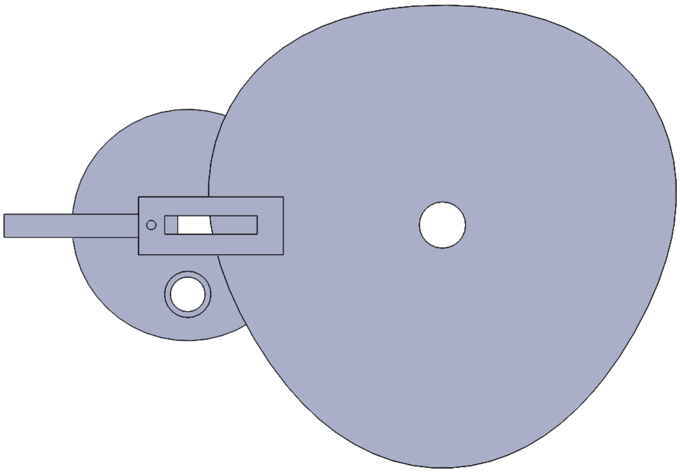


Figure 5: Svingen’s adapted valve design, seen from front

No further analysis is done on this design, but its performance will be made clear during the experiments. Due to the uncertainties concerning the oscillating hydrofoil as an excitation mechanism, Svingen’s design was chosen instead.

4.1.3 Water Hammer excitation

The proposed water hammer excitation mechanism is simple. It only consists of a ball valve placed at the outlet of the flow loop. The valve needs to be closed in a shorter time than the reflection time, which is about 2.8s. This is not very fast, hence the valve can be manually operated. A large diameter valve is costly, hence it is suggested to shrink the cross section in the final portion of the pipe leading up to the ball valve.

4.2 System characteristics

The water hammer simulation yielded the following results, illustrated in figure 6.

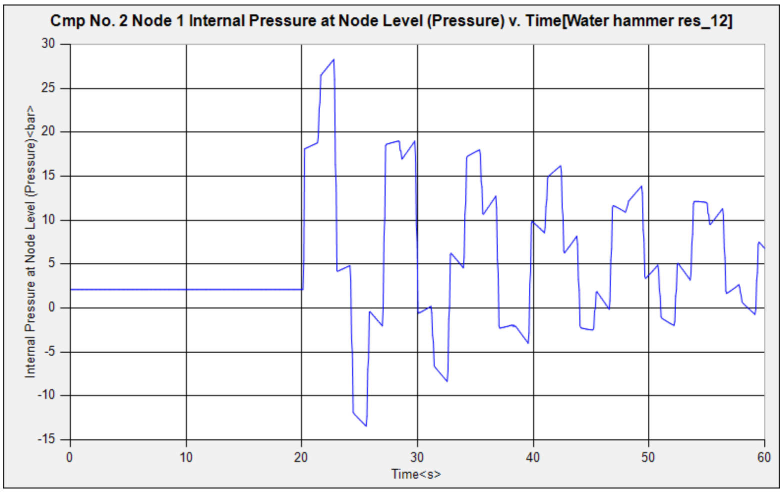


Figure 6: Time development of the pressure at an arbitrary node in the time following instant valve closure

The shape of the pressure waves is somewhat unexpected, but the period of these signals can be seen to be 5.6seconds. This yields the natural period, which will be half of the period found here.

$$f = \frac{1}{5.6s} = 0.179Hz$$

The theoretical period of the system is determined by formula 1, and also calculated to be 5.6s

6. Discussion

6.1 Chosen components

The simulations show that the internal excitation mechanism does not provide the desired system response. The validity of the simulations will be discussed later, but it is certainly not possible to say that it will work properly for the experiments. The external excitation mechanism however, is well tested in other similar experiments, hence it seems wise to continue with this design.

6.2 Simulated results

6.2.1 ANSYS

The contour plot in figure 4a, is mostly meant to give a qualitative impression of the pressure distribution throughout the domain.

The derived loss coefficient is in principle as expected. It is more or less sine shaped, with a steady periodicity and amplitude. However, the magnitude of the loss coefficient appears to be far too small. Investigation of the system response of a system subjected to this loss coefficient confirms this.

As always when doing CFD, some error is introduced. In this project, the magnitude of the error has not been thoroughly investigated, as it can be a time consuming process for transient simulations. However, one can make some general remarks regarding this. The main sources to error are assumed to be the following:

- Poor mesh quality
- Insufficient mesh resolution
- Insufficient size of domain
- Model implementation

- Unrealistic boundary conditions
- Unjustified two dimensional approximation

6.2.2 Flowmaster

It is seen in plot of the time development of pressure 4d that the system occupies an oscillatory state after some initial transient behavior. It can be seen that the characteristics does not yield a sine shaped pressure excitation, which was desired.

Since Flowmaster is solving the governing pipe flow equations by means of the method of characteristics, little numerical error is introduced here. The main source of numerical error is therefore in ANSYS. However, the model setup in Flowmaster is not exactly equal to the test facility. Despite the simplifications that are made in the Flowmaster network, the model is assumed to be sufficient for the purpose of this project. Namely to give an indication of the system dynamics when to system is subjected to the simulated characteristics of the excitation mechanism.

In general, the magnitude of the excitation seems to be way to low. This is mostly due to the fact that the excitation mechanism is too small compared to the cross sectional area of the pipe. As the design is now, the excitation mechanism takes up a maximum of 64%. In practice, this can easily be achieved, but it makes modeling trickier in ANSYS. The excitation element is also streamlined, but this is believed to have less of an impact of the magnitude of excitation since the flow is separated.

Filtering away the high frequency oscillations in the characteristics will also alter the simulated system response. Such oscillations in the characteristics are believed to be caused by vortex shedding, and will also be apparent in the measured pressure response downstream of the excitation element.

7. Further work

Since this master thesis is not completed, it is obvious that some work is yet to be done. At the time of writing, the design of the experimental setup is being finalized and the production of the designed components has begun. The experimental setup is to be tested in May 2017. The authors job here, is mainly to ensure that the system works properly, so that other research projects can make use of the test facility. The experimental setup must be considered successful if it can generate both oscillatory behavior with the desired frequencies and magnitude, as well as a water hammer effect with a sufficient magnitude. It is also important that the experiments do not break any of the components in the system. The transient effects are to be measured, but will not be subject to substantial analysis.

8. Conclusion

The proposed internal excitation mechanism is considered to be practically possible to implement in the closed flow loop. A more detailed analysis should be done in order to provide the necessary details for actually building this mechanism. The results indicate that the excitation mechanism should fill a larger portion of the cross sectional area of the pipe.

The external excitation mechanism has been proven to work, hence this design is chosen for the experiments.

The preliminary studies done on the closed flow loop appears to be correct. This is because the results from theory and simulations correspond, and that the simulated results exhibited the expected behaviour.

The internal excitation mechanism does not provide the required pressure amplitudes throughout the system. This is mostly due to the fact that the size of the excitation mechanism is insufficient. Some error is introduced in the simulations, but it is not possible to conclude on the magnitude of the error at this time. However, it is possible to claim that most of this error is introduced in the ANSYS simulations. Because of this uncertainty, it is not wise to make conclusions based on the completed simulations. Doing so might just as well lead to a wrong conclusion which is considered to be worse than no conclusion.

Acknowledgement

I would like to express my appreciation to all those who made the completion of this report possible. I would like to thank the staff on the Water Power Laboratory for facilitating such a pleasant working environment. A special gratitude to my supervisors, Bjørnar Svingen and Morten Kjeldsen. An extra appreciation is directed to Kjeldsen, for always being available, and for dedicating so much of his time to answer my questions. Without him, the completion of this project would never have been possible.

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[1] Svingen, Bjørnar "Fluid structure interactions in piping systems", 1996

[2] Thorley, David, "Fluid Transients in Pipeline Systems", Professional Engineering Publishing, 2004

[3] Courant, R.; Friedrichs, K.; Lewy, H. "On the partial difference equations of mathematical physics" September 1956

[4] Wylie, E. Benjamin; Streeter, Victor L., "Fluid Transients", McGraw-Hill 1978

Chapter B Additional Figures

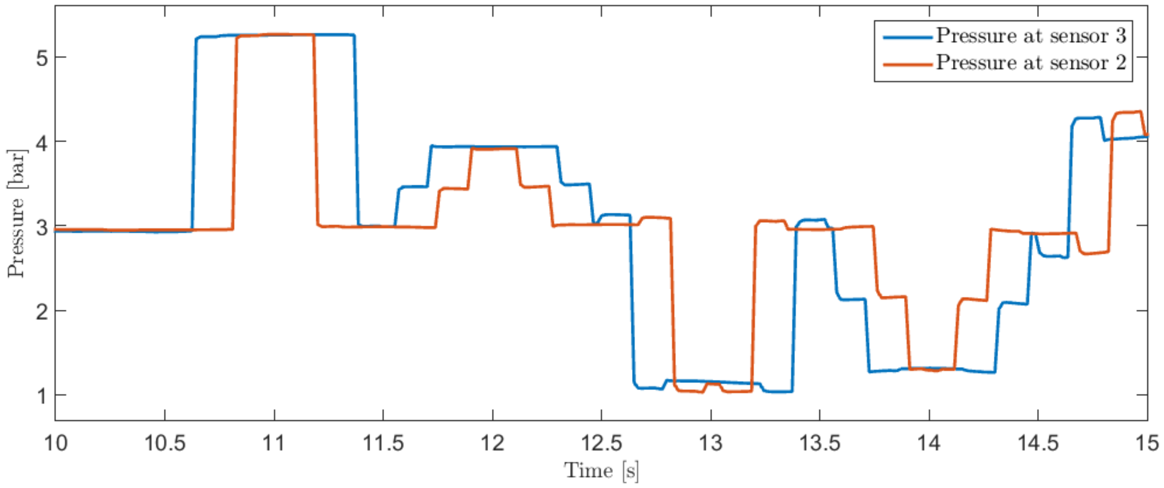

Figure B.1: Validity of using peak in pressure signal as reference point for wave speed calculations

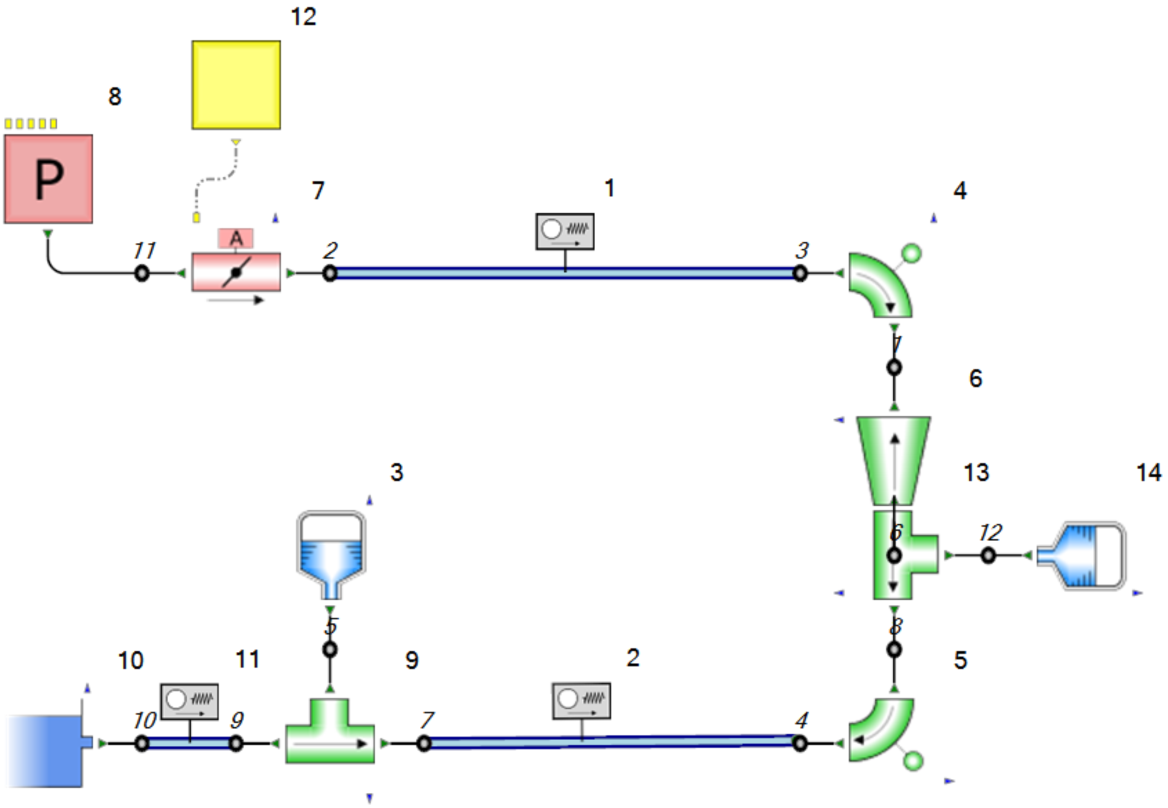

Figure B.2: FloMASTER network for investigation of reflection point

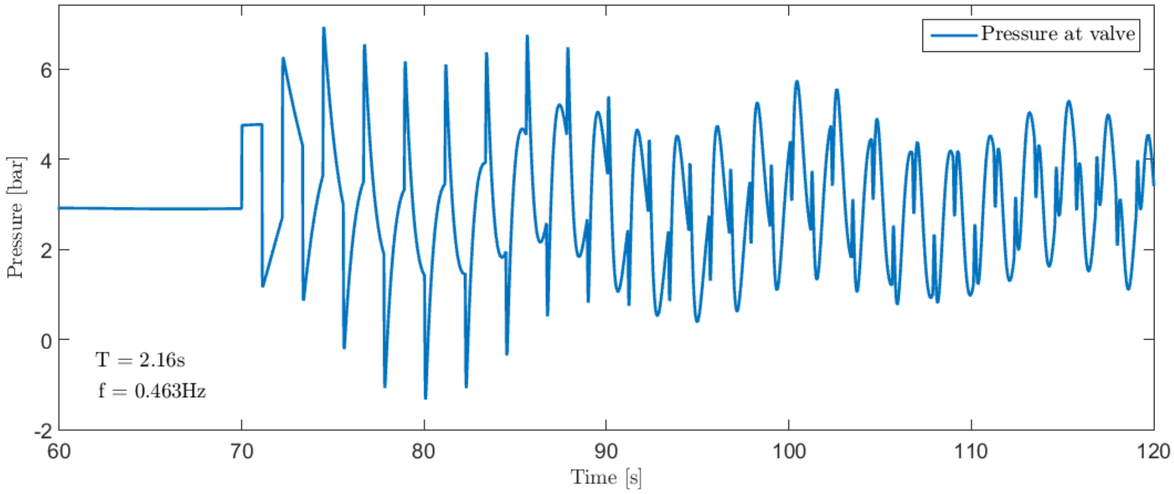

Figure B.3: Water hammer transient with air bubble located in pipe connection

Figure B.4: Photograph from test facility showing the length of the pipes

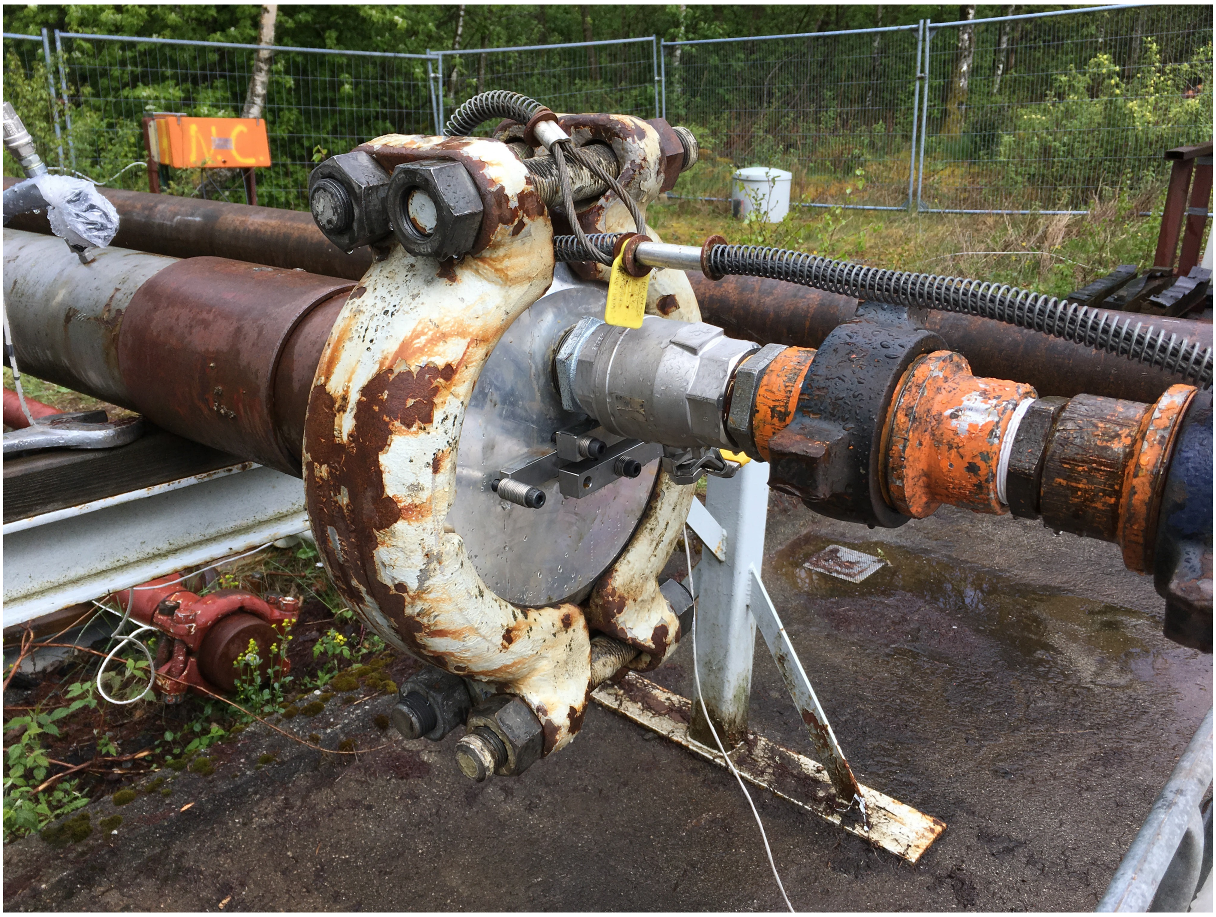

Figure B.5: Photograph showing the ball valve without the rotating disc mounted and the slit sealed off

Figure B.6: Photograph of the rotating disc during operation

Figure B.7: Photograph of the collection bucket, a part of the water recycling system

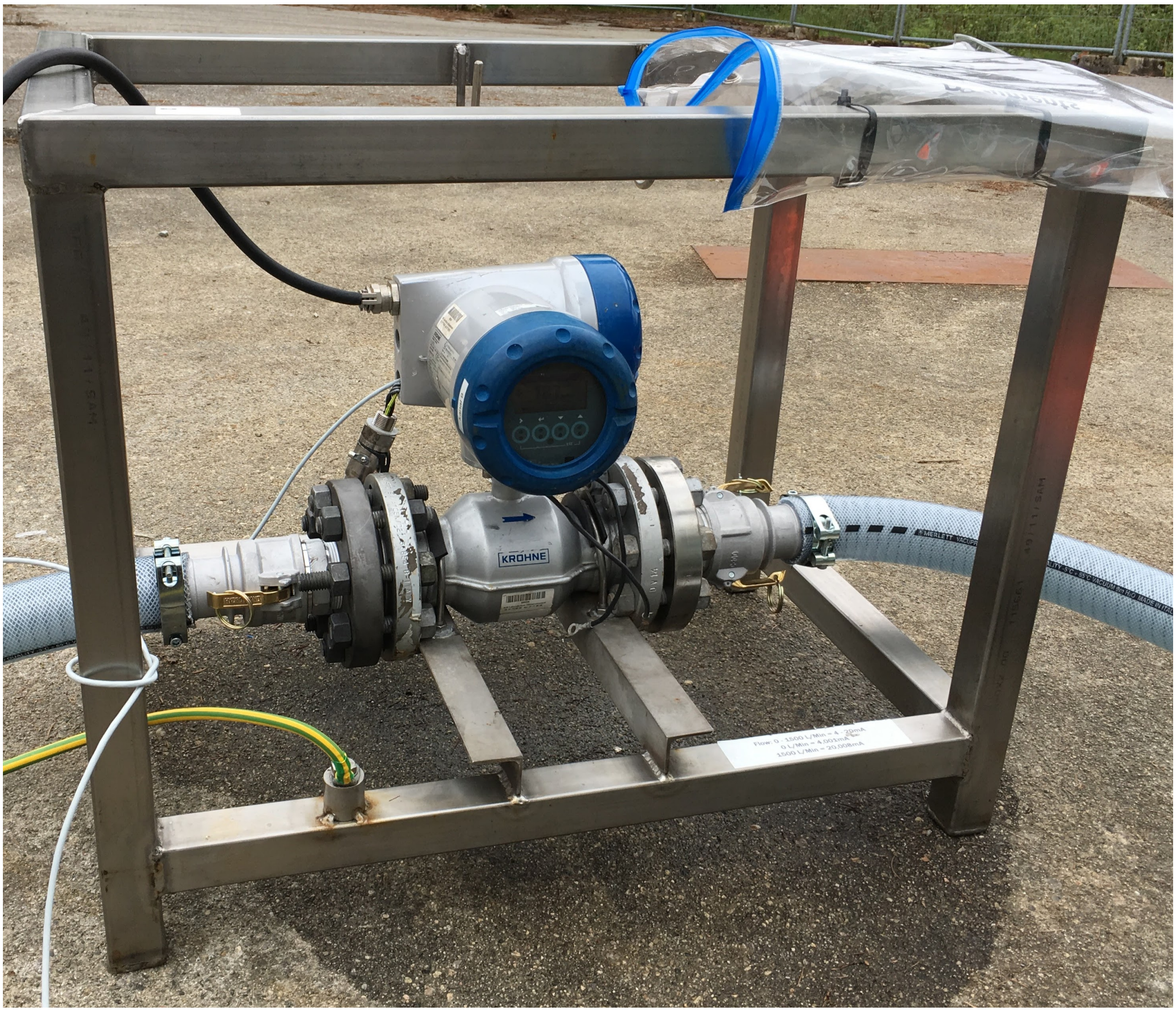

Figure B.8: Photograph of the flow meter

Figure B.9: Photograph of the frequency converter used for governing the motor and the potentiometer

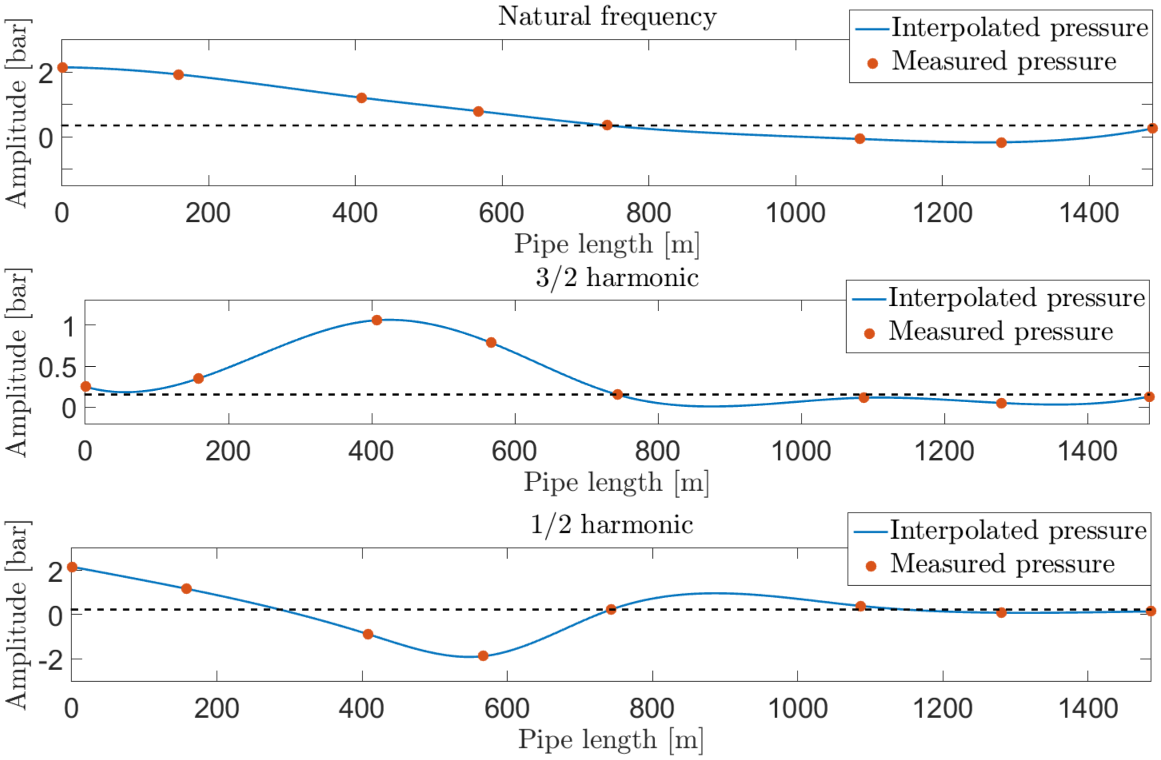

Figure B.10: The complete standing waves through the system

Chapter C Risk Assesment

NTNU	Risikovurdering	utarbeidet av	Nummer	Dato
		HMS-avd.	HMSRV2603	23.10.2013
		godkjent av	side	Erstatter
HMS/KS		Rektor	1 av 2	04.02.2011

Enhet: EPT/IVT **Dato: 08.05.2017**

Deltakere ved risikovurderingen (m/ funksjon): Eirik Myrvold Hansen (masterstudent NTNU), Carl W. Bergan (stipendiat NTNU), Ingrid Vilberg (stipendiat NTNU/ansatt i FDB).

ID nr	Aktivitet fra kartleggings-skjemaet	Mulig uønsket hendelse/ belastning	Vurdering av sannsyn-lighet	Vurdering av konsekvens:				Risiko-verdi	Kommentarer/status Forslag til tiltak
			(1-5)	Menneske (A-E)	Ytre miljø (A-E)	Øk/ materiell (A-E)	Om-dømme (A-E)		
1	Reise til og fra IRIS i bil	Trafikkulykke	2	D	A	D	A	8	Sikkerhetsbelter
2	Opphold på IRIS testanlegg i Stavanger	Påkjørsel av truck, skader fra annet maskineri eller andre forsøk	1	C	A	A	A	3	Verneutstyr i hht IRIS retningslinjer for HMS på arbeidsområdet
3	Gjennomføring av forsøk på IRIS-testrigg	Ødeleggelse av eksperiemntelt utstyr	3	A	A	B	A	3	Verneutstyr ihht IRIS retningslinjer for HMS på arbeidsområdet
4	Opprigg av forsøk	Klemskader, kuttskader	2	B	A	A	A	4	Verneutstyr ihht IRIS retningslinjer for HMS på arbeidsområdet

Sannsynlighet
1. Svært liten
2. Liten
3. Middels
4. Stor
5. Svært stor

Konsekvens
A. Svært liten
B. Liten
C. Moderat
D. Alvorlig
E. Svært alvorlig

Risikoverdi (beregnes hver for seg):
Menneske = Sannsynlighet x Konsekvens Menneske
Ytre miljø = Sannsynlighet x Konsekvens Ytre miljø
Økonomi/materiell = Sannsynlighet x Konsekvens Øk/matriell
Omdømme = Sannsynlighet x Konsekvens Omdømme

Sannsynlighet vurderes etter følgende kriterier:

Svært liten 1	Liten 2	Middels 3	Stor 4	Svært stor 5
1 gang pr 50 år eller sjeldnere	1 gang pr 10 år eller sjeldnere	1 gang pr år eller sjeldnere	1 gang pr måned eller sjeldnere	Skjer ukentlig

Konsekvens vurderes etter følgende kriterier:

Gradering	Menneske	Ytre miljø Vann, jord og luft	Øk/materiell	Omdømme
E Svært Alvorlig	Død	Svært langvarig og ikke reversibel skade	Drifts- eller aktivitetsstans >1 år.	Troverdighet og respekt betydelig og varig svekket
D Alvorlig	Alvorlig personskade. Mulig uførhet.	Langvarig skade. Lang restitusjonstid	Driftsstans > ½ år Aktivitetsstans i opp til 1 år	Troverdighet og respekt betydelig svekket
C Moderat	Alvorlig personskade.	Mindre skade og lang restitusjonstid	Drifts- eller aktivitetsstans < 1 mnd	Troverdighet og respekt svekket
B Liten	Skade som krever medisinsk behandling	Mindre skade og kort restitusjonstid	Drifts- eller aktivitetsstans < 1uke	Negativ påvirkning på troverdighet og respekt
A Svært liten	Skade som krever førstehjelp	Ubetydelig skade og kort restitusjonstid	Drifts- eller aktivitetsstans < 1dag	Liten påvirkning på troverdighet og respekt

Risikoverdi = Sannsynlighet x Konsekvens
Beregn risikoverdi for Menneske. Enheten vurderer selv om de i tillegg vil beregne risikoverdi for Ytre miljø, Økonomi/materiell og Omdømme. I så fall beregnes disse hver for seg.

Til kolonnen "Kommentarer/status, forslag til forebyggende og korrigerende tiltak":
Tiltak kan påvirke både sannsynlighet og konsekvens. Prioriter tiltak som kan forhindre at hendelsen inntreffer, dvs. sannsynlighetsreduserende tiltak foran skjerpet beredskap, dvs. konsekvensreduserende tiltak.

NTNU	Kartlegging av risikofylt aktivitet	Utarbeidet av	Nummer	Dato
		HMS-avd.	HMSRV2601	22.03.2011
		Godkjent av	Side	Erstatter
HMS		Rektor	1 av 1	01.12.2006

Enhet: **IVT/EPT** **Dato: 03.05-2017**

Deltakere ved kartleggingen (m/ funksjon): Eirik Myrvold Hansen (masterstudent NTNU), Carl W. Bergan (stipendiat NTNU), Ingrid Vilberg (stipendiat NTNU/ansatt i FDB).

Kort beskrivelse av hovedaktivitet/hovedprosess: Reise til IRIS testområde i Stavanger. Gjennomføre forsøk på IRIS testrigg i rørsløyfe

ID nr.	Aktivitet/prosess	Ansvarlig	Eksisterende dokumentasjon	Eksisterende sikringstiltak	Lov, forskrift o.l.	Kommentar
1	Reise til og fra IRIS i bil	IV,		Setebelter,	xx	xx
2	Opphold på IRIS testanlegg i Stavanger	IV, EMH		Hjelm, briller og vernesko	xx	I hht IRIS retningslinjer for HMS på arbeidsområdet
3	Gjennomføring av forsøk på IRIS-testrigg	IV, EMH	SJA sammen med IRIS + egen vurdering	Hjelm, briller og vernesko	xx	I hht IRIS retningslinjer for HMS på arbeidsområdet
4	Opprigg av eksperiment	IV, EMH		Hjelm, briller og vernesko	xx	I hht IRIS retningslinjer for HMS på arbeidsområdet