Technical University of Delft

Gate Driving Mechanisms and Dynamics

CIE5050-09 Additional Thesis project

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Preface

Dear reader,

You are about to start reading my additional thesis that is part of the master programme Civil Engineering track Hydraulic Structures from the Delft Technical University. During my master programme I was already fascinated with dynamic related problems. At one day, in my job as a student assistant, I started a conversation with my supervisor Wilfred Molenaar about driving mechanisms and the corresponding dynamics. This is the day that we started drafting a project proposal and several months later the additional thesis was finished. The main reason why this particular subject seemed interesting is due to the problems that may arise as a result of improper dynamic design. Lots of situation are known where the dynamics play a negative role after installation of hydraulic gates. I was fascinated by that fact and wanted to help by giving insight in the behaviour of driving mechanisms and then hydraulic cylinders in particular.

Since this additional thesis focusses on already existing large scale hydraulic projects, it is intended for designers that participate in such projects. It can be used as insight in the behaviour of different hydraulic cylinder components with respect to the dynamic quantities of a system.

Furthermore I would like to thank several people that made this additional thesis happen. First of all I would like to thank Wilfred Molenaar for making it possible to start the project by performing necessary operations and finding extra supervisors and consultants from different companies. Besides, I would like to thank Bas Hofland and Orson Tieleman for making time to check my work and propose extra requirements and goals for the additional thesis. Last but not least, my consultants Gerard Bouwman (Rijkswaterstaat) and Pieter van Lierop (IV-Infra). They assisted during several meetings and provided me with necessary insight in the behaviour of driving mechanisms and the design process, thanks a lot.

Martijn Helsdingen

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Abstract

Vertical sliding valves that are part of a filling and emptying system of a lock are often subjected to an underflow of water during emptying and filling. The flow induces time varying forces on the structure which leads to dynamic behaviour of the valve. Whether the structure is in the range of resonance and what the amplitude of the vibrations will be depends on the mass, damping, stiffness and forcing quantities of the system. What influence the driving mechanism, a hydraulic cylinder, has on the dynamic characteristics of the system is not completely understood yet. This thesis focusses on gaining insight in the behaviour of such a cylinder and its various components in terms of stiffness and damping of the total system including the vertical sliding valve. The research questions focus on whether or not it is possible to influence the dynamic characteristics of a system (natural frequency and dynamic amplification) by adjusting the geometry of the hydraulic cylinder. Furthermore it is investigated which components are influencing the dynamic characteristics of the system and which damping and stiffness components are found to be subordinate to the dominant sources.

Results were based on a Python script that included all relevant sources of damping and stiffness of a hydraulic cylinder, as well as the fluid structure interaction components such as added mass, damping and stiffness. The added damping components included the self-excitation suction damping, which on beforehand was expected to be potentially dangerous in terms of excessive excitations. The sensitivity of different components to the natural frequency and dynamic amplification was explored. This was done for different cases, where in each case one variable varied while the others were kept constant. The results from the sensitivity analysis were used to find an optimal parameter that would lead to an optimal design in terms of natural frequency increase or decrease, reduction of the dynamic amplification and a minimal influence on the mass of the system. Besides these two studies, a third study was adopted to find the relative influence of different damping and stiffness components of the hydraulic cylinder for varying boundary conditions such as water level difference and gate opening.

The study showed four components of a hydraulic cylinder that influenced the dynamic characteristics the most when varying their dimensions in a realistic range. These where the diameter of the hydraulic cylinder, the cylinder length, the thickness of the rod and the length of the tube that transport fluid into the cylinder. From these, the tube length and the cylinder diameters turned out to be the most effective design variables for tuning the stiffness, damping and correspondingly the natural frequency and the dynamic amplification of the system. Furthermore it was found that under all conditions (varying water level, gate opening height, pressure and stiffness), the stiffness was mostly determined by the axial stiffness of the rod and piston as well as the stiffness due to compaction of the cylinder fluid. For damping, it was found that the cylinder only had limited influence and that most damping resulted from friction between the valve and the guiding rails.

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1 Introduction

Dynamic response in gates due to fluid structure interaction (FSI) is a common process in all kinds of hydraulic structures. The structures has to be designed according to the guidelines of safety. When these guidelines are not fulfilled and the safety of the gates is not guaranteed, it may have serious consequences for the purpose for which the gate is implemented. Gate vibrations can cause failure, leading to flooding of the hinterland as well as stagnation of transport over water.

The driving mechanisms used for hydraulic gates are part of the dynamic characteristics of a system, and therefore influence the occurrence and magnitude of gate vibrations. In case of navigation locks the gate or valve driving mechanisms enable the chamber to fill and empty which causes flow of water inducing dynamic forces on valves. In case of weirs or flood defence systems opening and closing of the structure may cause currents over or under the structures which induce on their turn dynamic forces as well. Besides these flow induced dynamic forces one may think of waves, ship collision, debris and density currents as well. The driving mechanism can be incorporated as an element of the dynamic model having a certain stiffness and/or damping that may worsen or damp the vibrations of the gate. Therefore the driving mechanism of gates and their influence on the dynamic characteristics of a system are important knowledge in the design of these structures. The structure nowadays is designed according to some basic principles that form the basis of the dynamic design. In a later stage detailed tests are performed to obtain a realistic vision on the gate/valve dynamics and the appropriate solutions.

1.1 Problem statement

Often the detailed dynamic design of gates and valves is executed in a later stage of the design cycle, after the first designs are made according to the static forces. Gates and valves cannot be tested on full scale earlier than the installation date of the gate to examine the final dynamics. Due to the complex character of dynamic vibrations it is difficult to predict the behaviour of a gate or valve in an absolute manner. Besides that, the details of a gate are often important regarding the dynamic behaviour of the gate. This leads to a problem that the gate cannot be dynamically assessed in detail before the full static design of the gate is finished and all detailed elements are known. The possibility exists that the design turns out to be insufficient during late stage testing operations, which often leads to a delay in the project and subsequently to an economical setback of the project. Figure 1 gives a visual representation of the dynamic design stage in the design cycle.

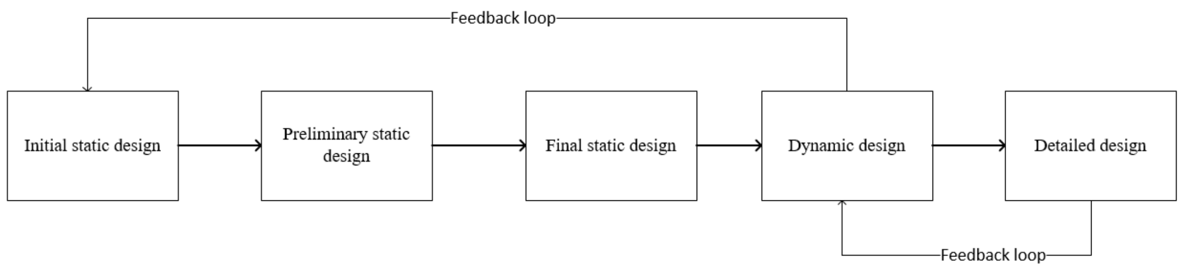


Figure 1 Design process flow chart. The dynamic design is incorporated after completion of the static design and may induce a feedback loop to the initial static design stage.

The above problem is often tackled by means of basic rules that follow from research and literature. These basic dynamic design rules may give insight in behaviour of the gate or valve in terms of excitation frequency and dynamic stability at the preliminary design stage. This is powerful knowledge due to the fact that it may predict whether or not the design is close to resonance. Nevertheless, more insight in the different components of driving mechanisms may lead to identification unforeseen events and solve or exclude them on forehand. Basic design rules regarding driving mechanisms and their dynamic response are still lacking. Due to nonlinear relations it is often difficult to quantify the effect of scaling up a certain component of a driving mechanism or gate in terms of stiffness and damping and the corresponding behaviour pattern. Quantifying the geometry of the drivers and gates may lead to hands on fixes for complex situations.

1.2 Scope of the project

A large variety of gate compositions is known in the work field of hydraulic structures. This study focusses on vertical vibrations of vertical sliding valves equipped with a hydraulic cylinder as a driving mechanism subjected to underflow. Several compositions of this type of gates and driving mechanisms exist in practice and are described further in the methodology (Ch. 5).

1.2.1 Goal of the project

The main goal of the project is to give insight in the behaviour of the hydraulic cylinder on one hand, the fluid structure interaction (FSI) on the other hand and the relative magnitude of the two. Brief overviews of different components that play a role in gate vibrations and relative magnitudes of these components may give crucial insights for gate designers to optimize designs. Especially for situations where vibrations occurred in the past that were not expected on forehand, such an overview may give extra options to find an optimal solution.

This thesis will focus on an overview of the relative magnitudes of dynamic components, to be known as:

- Mass components
- Damping components
- Stiffness components
- Dynamic characteristics of a system such as damping ratio and natural frequency
- Different excitation sources and their magnitudes

Qualitative description of these components has been included in chapter 5. In chapter 6 the relative importance of the components has been described.

1.2.2 Main deliverable

The main deliverable is a description and quantification of the absolute importance of the different dynamic components of a submerged vertical lifting valve as described in §5.2.1. Quantification is done in terms of relative magnitudes in which one may distinguish components that may become important for different situations. Solutions or designs may focus on these important components in order to find a possible efficient solution.

Coupling of the external forcing and forces coming from a hydraulic cylinder is an important aspect in optimizing the design. This aspect is expressed in the natural frequency of the system, which should be outside a specified range from the excitation frequency. Whether the dimensions of variables of a hydraulic cylinder may solve the dynamic problems that are expected is key knowledge for designers to tackle different issues. Furthermore the influence of the gate geometry in combination with external boundary conditions must be kept in mind all times.

1.2.3 Research questions

The discussed project scope leads to the following main research question:

- What role does a hydraulic cylinder play in combination with a vertical sliding valve in terms of stiffness, damping and the corresponding dynamic characteristics?

The sub questions are defined as follows:

1. Which components of an hydraulic cylinder are most sensitive for the stiffness, damping and natural frequency of the vertical sliding valve?
2. Which damping and stiffness components play a role in a lock gate with an vertical sliding valve?
3. Can the hydraulic cylinder influence the natural frequency of the system significantly?
4. If the hydraulic cylinder can influence the natural frequency, what is the most optimal solution in terms of minimal mass?
5. Can the hydraulic cylinder influence the damping ratio and amplification factor significantly?

1.3 Brief introduction in the research method

To get insight in the different components of a hydraulic cylinder, which leads to answers for the research questions, a model is adopted. The model is written in Python and contains the theoretical derivations of the different damping, stiffness and masses of a vertical sliding valve subjected to an underflow. A sensitivity analysis is executed by varying the dimensions of the different components, where after a broad set of data is collected that leads to conclusions which elements is most sensitive in terms of damping, stiffness, mass and the corresponding dynamic quantities of the system. Figure 2 gives an schematic overview of the situation that is considered. A more detailed overview is treated in §5.1.2 Figure 25.

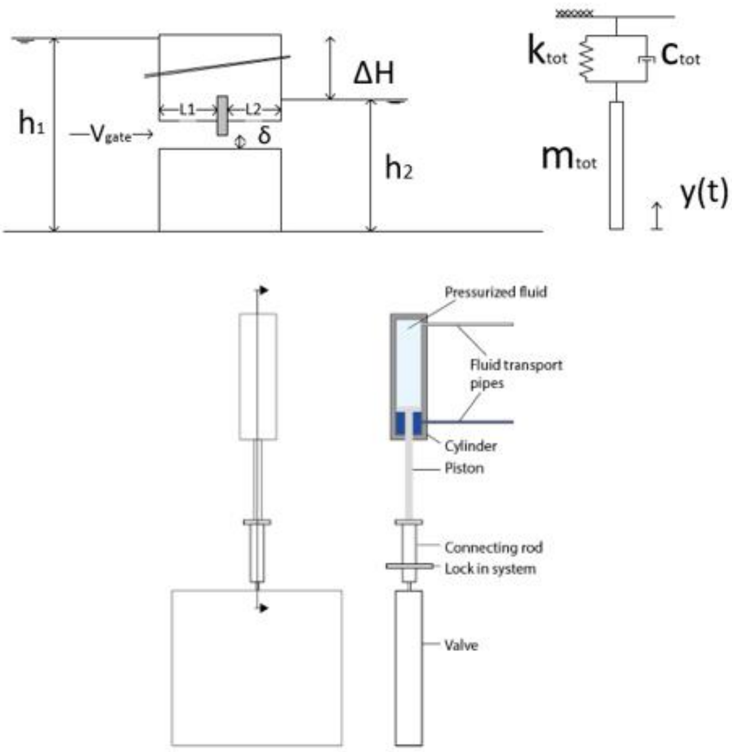


Figure 2 Vertical sliding valve in culvert through gate and dynamic mass-dashpot-spring system (top) and a lay-out of the valve including hydraulic cylinder (bottom).

Besides a sensitivity analysis for the different cylinder components, also the influence of the boundary conditions of the environment is tested. As a conclusion of the first two tests the final test focuses on solving the potential dynamic problems in an optimal design that is in line with the different basic design rules that are applied in practice. This leads to a conclusion on which components is best to design when dynamic problems are expected during the lifetime of a valve.

The forcing acting on the gate may be represented by a pressure variations over time for research regarding the excitation amplitudes and especially the coulomb damping components.

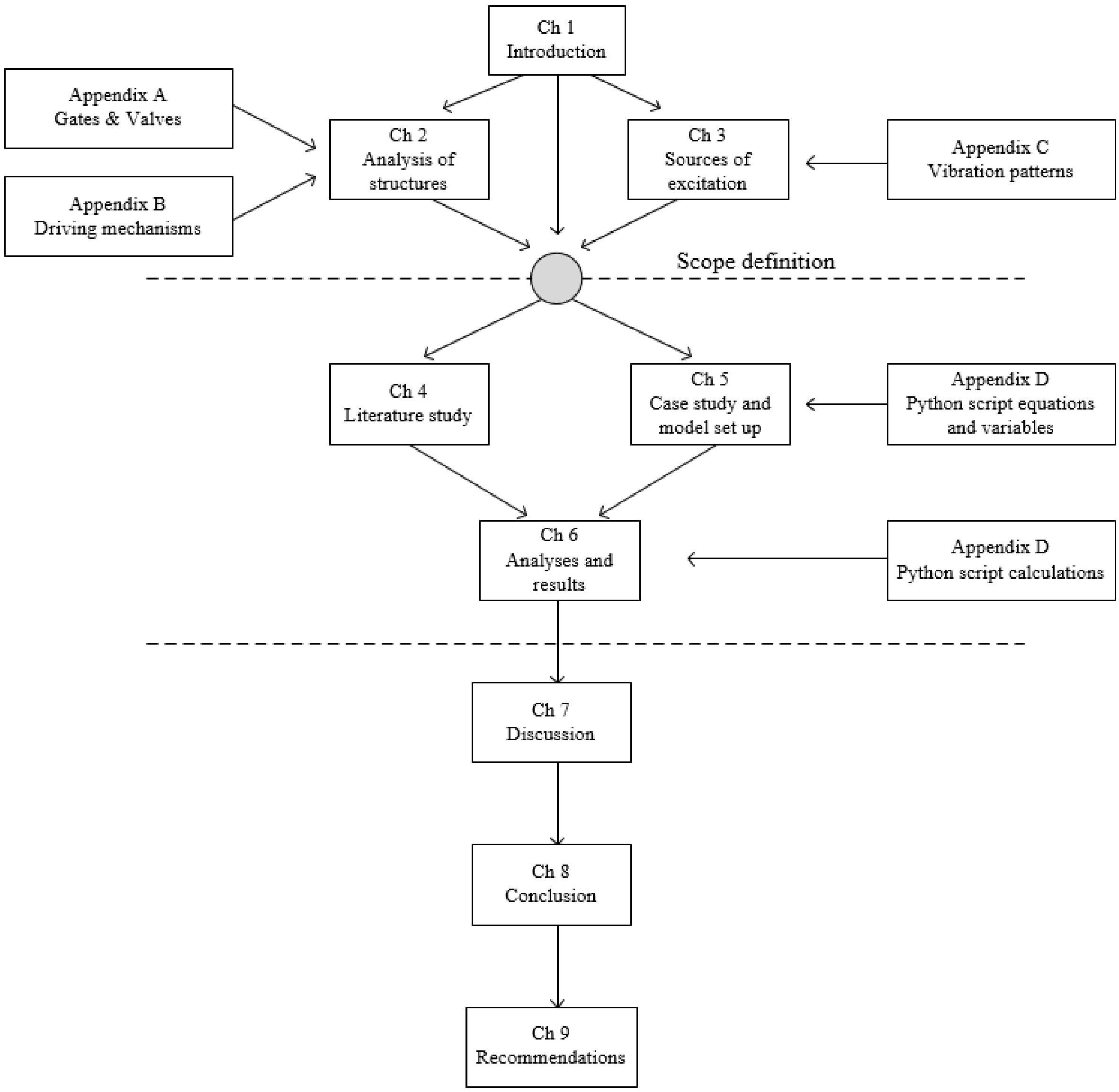
1.4 Thesis outline

In chapter 2 an overview of different valve and gate types is discussed together with the different driving mechanisms that are applied widely. This chapter serves as an overview of the different ways to implement and design a hydraulic gate. Narrowing down a wide range of possibilities, the vertical sliding valve was selected for further treatment in this report.

Chapter 3 will globally discuss the different sources of excitation that can be distinguished. Focus is on sources that match the scope of the project, the flow induced vertical vibration sources. Nevertheless, for the complete picture also the wave induced vibration sources are taken into account.

Chapter 4 dives into the literature that is available and relevant for the defined scope of the thesis. Literature on vertical vibrations due to an horizontal underflow heavily relies on empirical relations which are elaborated in detail in this chapter. Furthermore derivations regarding physical laws for stiffness and damping are treated in this chapter.

Chapter 5 discusses methodology of the research study. Where after chapter 6 contains the results of the study which is performed to obtain answers to the research questions that were drafted. Chapter 6 is followed by chapter 7 and 8 that respectively draft conclusions and highlight points of discussion.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2 Analysis of existing structures

This chapter gives a general overview of the different gates and driving mechanism that exist as part of hydraulic structures within the Netherlands. The last paragraphs serves as an overview of gates and their translation direction and which driving mechanism they are equipped with. Appendix A presents more detailed information regarding the structures, while the upcoming paragraphs will serve as a global overview.

2.1 Hydraulic gates and valves

Since this thesis focusses on vertical sliding valves, only this family of gates are treated in this chapter. In the Netherlands we apply several types of gates in our navigation locks. The valves that are used are limited to vertical sliding gates which are easily operated. A number of gates and valves is briefly discussed in the following paragraphs.

2.1.1 Vertical Lifting Gate

The name of this gate type already reveals its operating mechanisms. The gate opens and closes by means of vertical translation. The gate is embedded in between two towers where it may undergo a vertical translation over a rails. The gate is usually constructed using a skin plate which is supported by girders that are either vertically or horizontally installed. In some cases where the gate has to span large widths, the gate may also be equipped with a truss structure to increase the strength and stiffness.

Figure 3 Vertical lifting gates. Left: Eastern Scheldt storm surge barrier(Rijkswaterstaat & Cormont, 1991). Right: Hartel Barrier (Afdeling Multimedia Rijkswaterstaat & Rijkswaterstaat, 2000)

2.1.2 Vertical sliding valve

This gate is often applied in filling and emptying systems and is depicted as a valve in navigation locks. As the name suggests, it slides in a guiding rails in a vertical direction. Furthermore one can distinguish a gate and a valve by means of their dimensions. Gates tend to be larger in general and often act in free surface. Valves on their hand have smaller dimensions and are often installed in closed conduits. Opening the gate creates a path for water to flow along which fills or empties the navigation lock. They are either installed in the lock gates themselves or are part of culverts that are installed around the lock heads and provide a path for water to flow into the lock chamber. Depending on the dimensions of the lock, these vertical sliding valves can be up to several squared meters. In the rest of the report these vertical sliding valves will be addressed as valves.

Figure 4 Fill and empty valves in the sluice Empel. The smaller gates are opened and closed in vertical direction by hydraulic cylinders installed on top of the gates. (van Reeken & Rijkswaterstaat, 2014)

2.1.3 Other gate types

Besides the vertical lifting gate or valve multiple other important gate types can be distinguished. Appendix A gives more detailed information about these gate types. The types of gates that are often used are:

- Rolling gate
- Mitre gate
- Segment gate

2.2 Driving mechanisms

Driving mechanisms ensure that gates are able to open and close and have a mechanical character. Besides this function they may also act in terms of stiffness and damping in gate or valve vibrations. In this section a brief overview is given of the known driving mechanisms that are widely applied in the Dutch hydraulic structures. In §4.2 & 4.3 the stiffness and damping for different driving mechanisms is addressed. This chapter will focus on the hydraulic cylinder since this type of driving mechanism is most applied in hydraulic structures. Furthermore different other drivers are described in detail in appendix B.

2.2.1 Hydraulic cylinder

Hydraulic cylinders are widely applied in the newer gates and valves as a driving mechanisms. They come in all kind of shapes and the most applied in hydraulic structures will be discussed in this section. For older gates different types of driving mechanisms are often found. The reader is referred to appendix A for more information.

Workflow

The most common cylinder exists of a piston in a steel tube. From one side oil or another fluid can be pumped in under pressure, which forces the piston to translate horizontally. The injection of fluid is controlled by a control element which is installed between the piston and the pressurized fluid compartment. Figure 5 gives an overview of the workflow of a hydraulic cylinder. The small black arrows depict the flow of the fluid which is pushed into or is released for the cylinder.

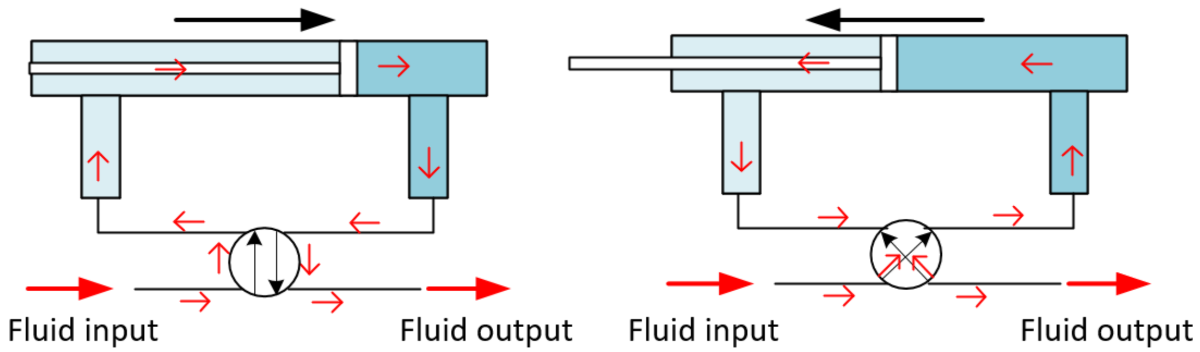


Figure 5 Workflow of hydraulic cylinder. Left: opening position and right: closing position. Small red arrows represent the flow of the fluid

From one side fluid is pushed into the control element, which leads the pressurized fluid to either the closing or opening side of the hydraulic cylinder. The fluid on the other side of the piston seal is drained off. This workflow makes it possible to either open and close a gate or valve by means of a pressurized fluid. The force can be adjusted by increasing or decreasing the pressure inside the piston. The pressure distribution in the piston may also affect the dynamic properties of the driving mechanisms, §4.3.1 goes into more detail on this matter. Mind that reservoirs for storage of the fluid are present at the other end of the fluid transportation tubes. These are not depicted in this figure and will not be taken into consideration in further course of this report.

Installation in structures

The way of installation depends on, among other things, on the translation direction of the gate. The cylinders are often applied in line with the translation direction of the gates, which means:

- Vertical lifting gate : vertical suspension
- Rolling gate : horizontal suspension
- Mitre gate : horizontal suspension
- Segment gate : horizontal and vertical suspension

Especially for a segment gate the type of installation of the hydraulic cylinder may deviate from the regular vertical or horizontal suspension. All types of directions can be applied over here since the gate has to rotate around a certain pivot. In Figure 6 an example is given for the Haringvliet sluices which depicts the complex installation of the driving mechanism compared to an ordinary vertical lifting gate.

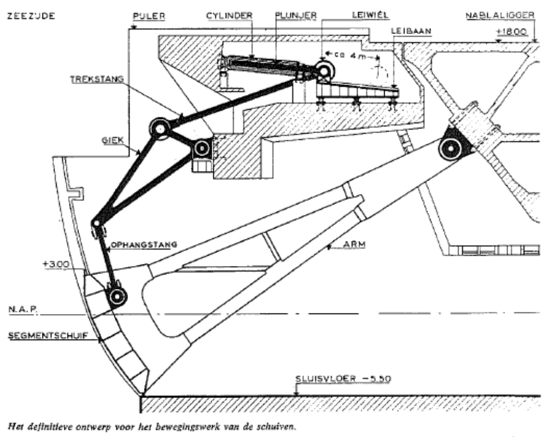


Figure 6 Driving mechanism of the segment gate of the Haringvliet sluices (Ferguson, 1971)

Figure 7 Hydraulic cylinder on the vertical lifting gate of the Kromme Nol. Left an overview of the total gate indicating detail A which is presented in the right figure. It shows how the piston is connected to the gate.

2.2.2 Other types of driving mechanisms

Besides the hydraulic cylinder several other types of drivers can be distinguished. These types are not often applied nowadays.

- Towing cables/chain
- Gear rack
- Panama wheel

2.3 Recollection – overview of gates and corresponding drivers

This paragraph will give a visualized overview of all different gates and valves that are discussed in §2.1 and appendix A, together with their:

- Translation direction
- Driving mechanism

Table 1 Overview of often used hydraulic gates with their translation direction and potential driving mechanism

Gate type	Translation	Hydraulic cylinder	Towing cable	Gear rack
Vertical lifting gate	Vertical	Y	Y	Y
Vertical sliding valve	Vertical	Y	Y	Y
Segment gate	Radial	Y	N	N
Rolling gate	Horizontal	Y	Y	Y
Mitre gate	Horizontal	Y	N	Y

3 Sources of excitation and vertical vibration patterns

This chapter focusses on a brief introduction of different types of sources that may lead to vertical valve vibrations. The knowledge is mainly based on the book of Kolkman & Jongeling (1996), in which they present an overview of all dynamic related subjects for hydraulic structures. Firstly a general overview of different vibration sources is presented, after which an overview of vibration patterns is given. In the last paragraph a wrap up is given which includes a summary of the different sources and patterns of vibrations.

3.1 General overview of sources of vibration

Excitations of gates and valves of different kind of structures can originate from several causes. The most common and experienced sources for dynamic behaviour are:

- Flow induced vibrations
- Wave induced vibrations
- Self-excitation
- Vibration patterns of water bodies around gates
- Ship collision

From this list the most important sources, where gates are often subjected to, are flow induced and wave induced vibrations and their corresponding self-excitation of the gate.

3.1.1 Flow induced

Flow induced vibrations are commonly found in elements that are subjected to a flow of water during either filling or emptying a lock or in case of a weir in a river. The flow underneath a structure can either induce vertical or horizontal vibrations, which depends on how the gates or valves are installed in their surroundings. We can distinguish the following excitations(Kolkman & Jongeling, 1996) that lead to vertical vibrations:

- Sudden force due to flow
- Turbulent flow
- Flow instability
- Self-excitation
- Amplification due to fluid resonance

Sudden flow force (Drag flow)

A stepwise increase of the flow velocity around an object will lead to a sudden load that acts on the structure for as long as the water flows underneath, over or around the structure. This might be schematized as a harmonic load with a frequency of 0 Hz.

The corresponding force of the flow of water is given by means of a drag force behind the structure. This widely applied equation is known as:

$$F = \frac{1}{2} C_D \rho u^2 A \qquad (1)$$

In which:

C_D	= Drag coefficient of object
ρ	= Density of fluid
u	= Flow velocity of fluid
A	= Cross sectional area of the object perpendicular to flow direction

An sudden excitation of the structure may enhance self-excitation, which may be very dangerous and should be avoided at any costs. This mechanism will be treated in one of the upcoming paragraphs. Mind that the drag force acts in the horizontal plane. Due to pressure increase locally underneath the gate because of this force, a vertical component will originate as well.

Turbulent flow – vortex shedding

A turbulent flow causes pressure variations below or downstream of the gate. These pressure fluctuations on their turn lead to a fluctuating force in time. Using the Strouhal number the frequency of the vortex can be obtained (Kolkman et al, 1996):

$$S = \frac{fL}{V} \tag{2}$$

In which:

S = Strouhal number
f = Excitation frequency of surrounding liquid
L = Dimension of the gate or element that experiences vibration
V = Flow velocity

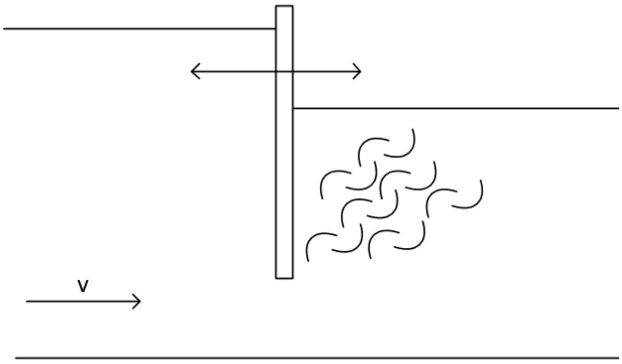


Figure 8 Turbulence induced horizontal vibrations

The Strouhal number describes the oscillation patterns around a gate and therefore can be taken as a measure for either the dominant excitation frequency, in case of a range of excitation frequencies, or the only excitation frequencies in case of a uniform flow condition which is constant in time.

The Strouhal number is more or less a constant for several hydraulic engineering structures for a given geometry. They can be obtained by means of scale tests or literature. In general it is advised to compare the Strouhal number with the natural frequency of a gate. The guidelines prescribe a sufficient safety buffer between the natural frequency and the excitation frequency, in that case(TNO, 2017):

$$S_n \approx (2-3)S \rightarrow \; f_n > 2 - 3\frac{SV}{L} \tag{3}$$

With:

S = Strouhal number using (dominant) excitation frequency

S_n = Strouhal number using natural frequency of the system

According to Kolkman et al (1996), it is not feasible to stiffen the structure such that the above requirement for the Strouhal number holds for every vibration mode of a gate or valve. Only the one that are excited due to turbulent flow should be designed such that the Strouhal requirement holds. The latter holds for practically all excitation sources that induce a range of excitation frequencies.

Flow instability

Kolkman et al (1996) state that flow instability occurs at structures where the detachment point is not clearly defined such as rounded surface of gate areas where water flows. Instable attachment points are found in structures that have a well-defined detachment point, but have a large width so that flow can attach again.

Kolkman et al (1996) mention that the length between the detachment and reattachment point is a measure for the forcing frequencies and therefore are a good measure of the excitation frequencies that a gate may experience. A stable design considers a stable detachment point together with avoiding the flow to reattach. Design C in Figure 9 is an example of a well-defined design(TNO, 2017).

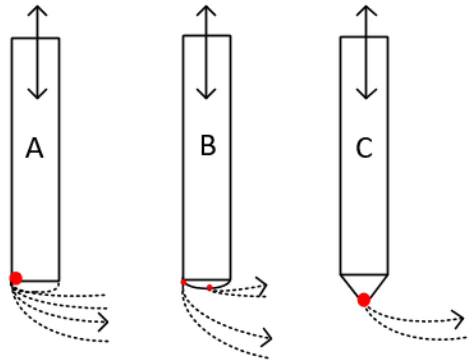


Figure 9 Flow instability. A: Reattachment point downstream, B: unstable detachment point, C: Stable detachment point

Self-excitation

Due to the FSI an extra damping term is added to the system. These are commonly known as water damping. This extra damping is not always positive, but might be negative as well(Kolkman et al, 1996). In case of a total damping term, so damping of the system or driving mechanism combined with added water damping, is negative, self-excitation may occur. The vibrations that are induced by a sudden load or pulse will then exponentially grow in time and are not damped out anymore, which is visualized in Figure 10.

Self-excitation may also be quantified by means of the Strouhal number. Kolkman and Vrijer(1977) found that critical Strouhal numbers are in the range of 0 - 0.1. They also proposed methods for solving the suction damping of vertical lifting gates who vibrate in the vertical plane. This will be elaborated in §4.6.6.

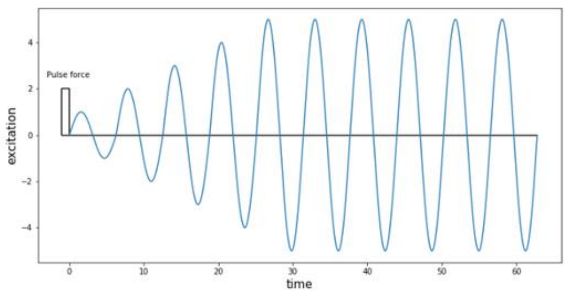


Figure 10 Response of gate experiencing self-excitation due to pulse force

Mind that the initiation of self-excitation is visualized by means of a pulse force. The pulse force on the other hand is not the only perturbation in an equilibrium that induces self-excitation. Pulse force equivalents may also be found in sudden flow forcing.

Amplification due to resonance of fluid

Fluids in a confined lock chamber may be excited by a sudden induced flow during emptying and filling or by dynamic excitations of gates in waterways that vibrate with the same frequency as the natural frequency of the waterbody. The latter is only the case for large sized floating structures. Due to resonance of the fluid standing waves might occur that interact with the gates in a transverse or longitudinal direction. Resonance of fluids are mainly induced in finite water bodies such as sluices or a closed part of a river for flood protection.

The oscillations of the fluid in the compartment that is connected to the gate or valve induce a varying head difference over the structure. This varying head difference on its turn will lead to a varying discharge and thus varying flow velocity through the gate. The latter may induce both vibrations parallel or perpendicular to the flow. Besides the fluctuations in flow, one might also distinguish a pressure fluctuation pattern due to the varying water level differences.

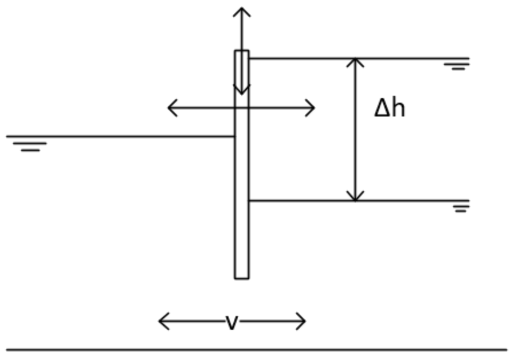


Figure 11 Fluid resonance and corresponding flow reversal and vibration directions

3.1.2 Wave induced

Wave induced sources for excitation may be limited to a number of cases. Well known are extreme conditions and the corresponding waves for structures that are located near the sea, ocean or large lakes. Large wave heights slamming periodically against a door lead to a time varying force which may be expressed as an harmonic excitation. In case of navigation locks translator waves can occur which are an order smaller than wind waves, but can induce vibration patterns close to the natural frequency of smaller elements.

As far as wave induced vibrations are concerned, they can both influence the horizontal and vertical vibrations of structures, depending on the type of gate that is analysed. Vibrations in the horizontal plane are mainly the cause of harmonic slamming of waves against the gate. In case over vibrations perpendicular to a wave direction, pressure difference play a role. Periodically varying of wave heights can induce a periodic varying pressure in case of an open gate or valve. This will lead to a time varying force. Both situations are schematized in Figure 12.

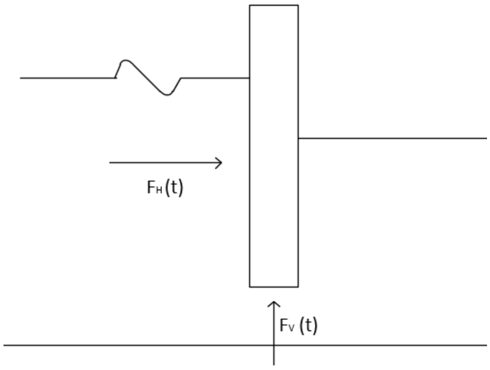


Figure 12 Pressure and corresponding force variation in time on gate or valve subjected to waves

3.2 Vibration patterns of gates and valves

Different gate types generate different vibration patterns. The direction of excitation is mainly determined by the way a gate or valve is installed in the structure and their corresponding degree of freedom. Besides that, the filling and emptying system is also responsible for the way a gate or valve behaves in a dynamical sense and which sources are present in the gates geometry.

This paragraph will globally highlight the potential vibration pattern of a vertical sliding valve. In the final section a general overview will be given in the form of a table which includes the gates and their excitation sources with corresponding vibration patterns or directions.

3.2.1 Vertical lifting valve

Vertical sliding valves are often subject to strong flow patterns through the gate in case of filling and emptying locks. Vertical sliding valves may be designed for extreme storm conditions and corresponding flow velocities.

Excitation source	Excitation direction
Sudden force due to flow	Horizontal plane
Turbulent flow	Vertical plane
Flow instability	
Amplification due to fluid resonance	
Self-excitation	

Mind that for especially the flow induced vibrations, the geometry of the edge of the gate is of importance for the resulting vibration pattern. Having said this, the potential solutions of dynamic vibration patterns can be endless.

3.3 Wrap-up excitation sources and vibration patterns

Combination of gate types and sources of excitation results in a wide variety of FSI behaviour. The dynamic behaviour of these gates and valves depend on a lot of variables for each specific case, from which the following may play a role:

- Dimensions of the gate
- Location of the structure and the corresponding forces
- Vibration direction of the gate and the way of installation in the structure
- Driving mechanism of the gate
- In case of a hydraulic cylinder the length of the pressure pipes plays a role in the stiffness of the total mechanism

- The geometry of the flow area across the gate, is their open surface water or does the flow concentrate in a pipe in case of valves in gates?

This is a brief selection of the huge amount of variables that may play a role in the dynamic behaviour of the structure. In order to obtain a manageable project the focus for this additional thesis will be on flow induced vertical vibrations of structures driven by hydraulic cylinders.

To be more specific, the following will be treated in this thesis:

1. Valves that undergo an excitation in the vertical plane and which are driven by a hydraulic cylinder.
2. Importance of different dynamic components of a vertical lifting gate
    a. Damping and stiffness of the hydraulic cylinder
    b. Damping and stiffness of the valve itself
    c. FSI components: added mass, damping and stiffness

4 Literature – Dynamic quantities revisited

This chapter will focus on the literature about dynamics that is already studied in detail. It will serve as a basis for the development of a script and analysis of the data later in this additional thesis. The following subjects are touched:

1. Dynamic quantities of structures that interact with water
2. Eigen frequency and resonance
3. Dynamic stability of structures
4. Frequency response
5. Situations that drive dynamic response
6. Basic design principles

In this chapter a more detailed view will be given on the relevant dynamic components, according to the available literature. The valve configuration given in chapter 4 will be investigated. The literature study is based on valves and their corresponding vertical translation direction.

4.1 Mass

The mass of the system can be deconstructed in three components:

1. Mass of the gate or valve
2. Mass of the piston in the hydraulic cylinder
3. Added water mass

The most straight forward masses that can be determined are the mass of the gate or valve and that of the piston in the hydraulic cylinder. Vibrating in the vertical plane, the gate also excites the piston which therefore should not be neglected in the equation of motion. This hold for both vertical or horizontal installed hydraulic cylinders.

4.1.1 Added water mass

In the situation as described in chapter 1.2, a vertical sliding valve excited by a horizontal flow and vibrating in the vertical direction, Kolkman et al(1996) propose a formulation for the added water mass. Figure 13 gives a representation of this extra mass.

$$m_w = \frac{2}{3}\frac{\rho a^3 L}{d} \tag{4}$$

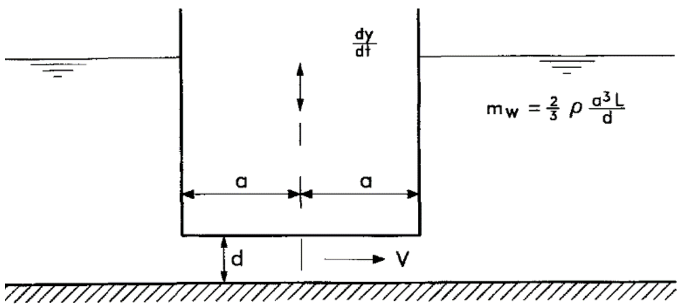


Figure 13 Added water mass for vertical lifting gate with open water surface (Kolkman et al, 1996)

4.2 Damping

Damping in gates and valves that are embedded in a structure and are driven by a hydraulic cylinder can be classified into the following classes:

1. Coulomb damping due to friction
2. Damping exerted by the hydraulic cylinder
3. Added water damping

An schematic overview of where these damping components act can be found in figure 22. The damping of the system itself, without any interaction with water, solely depends on the friction of the gate with the structure and the piston with the hydraulic cylinder. Damping that is determined by means of friction is the so called Coulomb damping. Furthermore we can also define a viscous damping term in case of a piston moving in a hydraulic cylinder. Due to the presence of a viscous fluid one might observe damping by movement of the piston through this fluid volume.

The total damping force that is exerted on a system is proportional to the velocity of the gate or valve:

$$F_{damping} = \left(c_{system} + c_{mechanical} + c_w\right)\frac{dy}{dt} \tag{5}$$

4.2.1 Coulomb damping

Coulomb damping is often observed in vertical sliding gates where the gate meets its surrounding structure. The damping is exerted on the structure and often represented by a certain percentage of the reaction force of the gate's friction force. The friction force may be expressed in the following formulation:

$$F_w = \mu N \tag{6}$$

According to Gerard Bouwman (personal communication, may 25th, 2020) the friction coefficient for sliding gates is in the order of 20%. This value is substantiated by data delivered by IV-infra (personal communication, june 15th, 2020). Variables from equation 6 are known to be:

N	= Normal force on interface valve - rails	
μ	= friction coefficient	≈ 20%

Spijkers, Vrouwenvelder, & Klaver (2005, pp. 170–173) describe a method for deriving the Coulomb friction in a mathematical way. Without the derivation, the result is given in equation 7.

$$c_{coulomb} = \frac{4F_w}{\pi\hat{y}\omega} \tag{7}$$

In which:

F_w	= friction force originating from the water pressure on the valve
$\hat{y}$	= amplitude of vertical vibration
ω	= excitation frequency

The friction force originates from the water pressure acting on the gate, which generates a pressure force on the guiding rails of the valve. This pressure force is solely a function of the water level difference over the lock gate and is approximated by equation 8.

$$F_w = \mu\rho g\Delta HWh_{gate} \tag{8}$$

In which:

F_w = friction force originating from the water pressure on the valve
ΔH = water level difference
W = width of the valve
h_{gate} = height of the valve

Equation 7 further shows that the Coulomb damping is frequency dependent and should therefore be treated that way.

Note that coulomb damping may be active in both the interface between the valve and the gate as well as the piston and the hydraulic cylinder.

4.2.2 **Viscous damping**

Viscous damping is solely found in the hydraulic cylinder and due to the relative size of the piston compared to the gate or valve, this term is often neglected in dynamic computations (Gerard Bouwman, personal communication, may 25th, 2020). Nevertheless, a viscous damper may be incorporated in case of negative damping values of the design.

4.2.3 **Material damping**

Energy dissipation by strain of a material can be incorporated as a damping component. Spijkers et al(2005, p180) give a range of dimensionless damping parameters for different kind of materials. Assumptions for this derivation is that the material is not loaded up to 70% of its fatigue stress threshold. Relevant dimensionless damping coefficients are(Spijkers et al, 2005, p180):

- ζ_{steel} = 0.004
- $\zeta_{aluminium}$ = 0.018

Material damping is found in the piston of the hydraulic cylinder and might also be present in the hydraulic cylinder itself. Expansion of the cylinder due to an increase in pressure will lead to strain in the material and consequently to energy dissipation.

In case of a piston it is important to understand that the critical damping is the one of the piston itself, rather than the complete system. The same holds for the hydraulic cylinder.

4.2.4 **Added water damping**

Damping due to fluid structure interaction may cause severe problems in case of negative added water damping.

Kolkman et al (1996) describe several cases in which additional water damping may be active in a structure. These are known to be:

- Radiation of energy into surface waves
- Additional damping for structures that vibrate perpendicular to the flow direction

Radiation in to surface waves

The damping that is linked to radiation of energy into surface waves may be described by the following relation(Kolkman et al, 1996)

$$c_{w-waves} = \frac{2\rho g^2}{\omega^3} \tag{9}$$

In which ω is denoted as the excitation frequency of the gate or valve. In most cases of problematic FSI the frequencies are high and can therefore be **neglected**.

Vertical flow damping – suction

Negative damping may be experienced in case of a varying opening of the gate in combination with a stable discharge through the gap (Kolkman et al, 1996). In the case of negative damping, the structure will become unstable and the excitations will grow in time. This source of excitations is described in detail in §3.1.1.

Flow will fluctuate simultaneously with the fluctuations of the gate opening. This induces pressure fluctuations underneath the gate or valve which have a time dependent character. Kolkman and Vrijer(1977) indicate this added damping as suction in the flow gap and propose a relation that may describe the negative damping of a gate that is excited in the vertical. They proved with tests that the negative damping is proportional to the Strouhal number:

$$C_{SE} = \frac{-c_w}{C_F \rho B \sqrt{2gH_0}} = \frac{\mu C_i}{(1 + 2\pi S \mu C_i)^2} \qquad (10)$$

In which:

C_i	= Coefficient of inertia	= $(C_{L,upstream} + C_{L,downstream})\ \delta$
δ	= Gate opening	
C_L	= Coefficient for added water mass	= obtained from Figure 14
d	= Water depth	
C_{SE}	= Coefficient of self-excitation	
b	= Width of gate or valve	
B	= Area of the bottom part of the gate	
H_0	= Water head difference over the gate	
C_F	=	= $C_s + C_L/Ci$
C_S	= Suction coefficient	$\approx \delta$ if $\delta < 0.65b$ = 0 if $\delta > 0.65b$
μ	= Contraction coefficient	
S	= Strouhal number	$= \frac{\omega\delta}{2\pi\sqrt{2gH_0}}$

The results from Kolkman & Vrijer(1977), depicted in Figure 14 & Figure 15, show a range of critical Strouhal numbers in which negative damping may occur that is rather low. The critical range is somewhere around 0 up to 0.2.

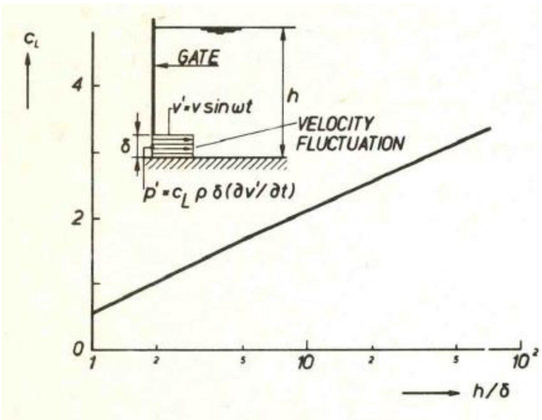


Figure 14 Definition for C_L coefficient as a function of water depth and gate opening (Kolkman and Vrijer, 1977)

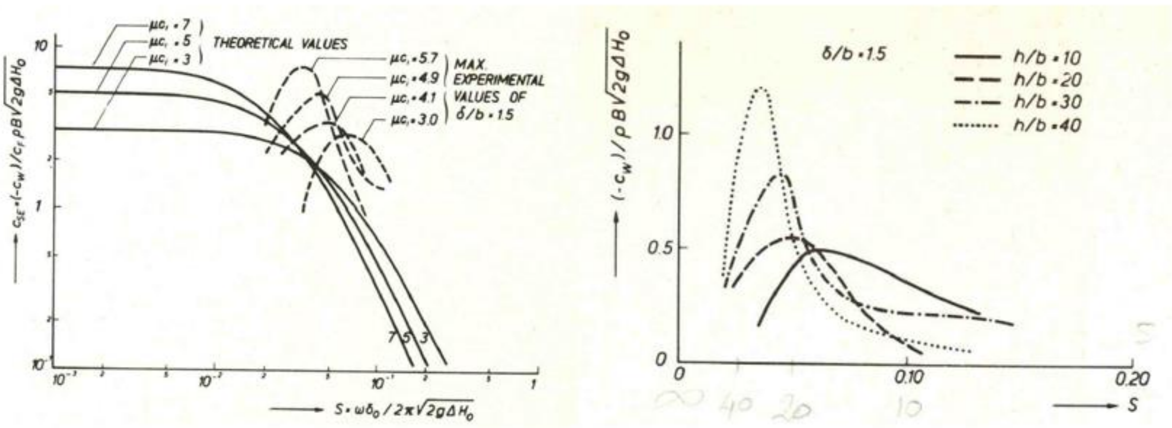


Figure 15 Experimental results of Kolkman and Vrijer(1977) for negative suction damping coefficient. Left: coefficient of self-excitation vs Strouhal number, right: negative suction damping vs Strouhal number

Derivations of the negative added water damping from Kolkman & Vrijer (1977) are based on the momentum balance downstream of the gate. It is assumed that the local head difference in this region results in an additional force in the gate opening. This force can be expressed in a pressure increase in the vertical direction, which fluctuates in time.

4.3 Stiffness

Damping and stiffness of a structure are somewhat related to each other and find themselves to be originated from the same sources often. Stiffness of a system is only working opposite to the displacement rather than the velocity, and therefore the two are more or less 90 degrees out of phase with each other.

The stiffness of a system that interacts with water can be decomposed into the system stiffness and the added water stiffness. Both will be treated in this paragraph.

4.3.1 Stiffness of the hydraulic cylinder

The bulk modulus of the fluid is the key characteristic that is used in the derivation of the stiffness induced by the hydraulic cylinder.

$$B = -\frac{V}{\Delta V}\Delta p \tag{11}$$

In which:

V = Volume of water in the cylinder

ΔV	= Change of volume of water in the cylinder	
Δp	= Pressure exerted on the water body	
B	= Bulk modulus of water	

Consulting Figure 5 and Figure 26, we might express the volume of water in the cylinder by the relation in equation 12 and the difference in volume as given in equation 13.

$$V = \frac{1}{4}\pi D^2 * L \tag{12}$$

$$\Delta V = \frac{1}{4}\pi D^2 * y \tag{13}$$

The pressure that is exerted on the water body is induced by the force in the piston, which is known as:

$$F_p = A_{piston}\Delta p = \frac{1}{4}\pi D^2 \Delta p \tag{14}$$

Using equations 13– 16 and applying the definition of a spring force we obtain an expression for the spring stiffness like in equation 15.

$$B = -\frac{\frac{1}{4}\pi D^2 * L}{\frac{1}{4}\pi D^2 y} * \frac{F_p}{\frac{1}{4}\pi D^2} \rightarrow F_p = \frac{\frac{1}{4}B\pi D^2}{L} y \rightarrow k_{cyl} = \frac{B\pi D^2}{4L} \tag{15}$$

With:

B	= Bulk modulus of fluid	= 1.5 GPa
L	= Length of cylinder where force acts on	
D	= Diameter of the cylinder	

4.3.2 **Stiffness due to deformation of steel components**

When focussing on the driving mechanism of the gate and vertical vibrations, we may also define a stiffness that follows from the deformation of steel components. The relation between axial deformation and stress is given by Hooke’s law (eq. 16).

$$N = EA(x)\varepsilon = EA(x)\frac{\Delta l}{l} \tag{16}$$

The stiffness can now be obtained by taking the right hand side of equation 16 except for the Δl, which denotes the displacement. Most cylinders are equipped with a piston that has a constant surface area. In that case the area doesn’t depend on the vertical coordinate anymore.

$$k_{axial} = \frac{EA}{l} \tag{17}$$

In case of bending resistance of steel components, we should adopt another formulation for deformations. Bending deformations occur in cases where the hydraulic cylinder is place perpendicular to the gate or valve(Figure 16).

Mind that axial stiffness can also be found in the hydraulic cylinder itself. Radial expansion ca be expressed in terms of length by means of the perimeter of the cylinder.

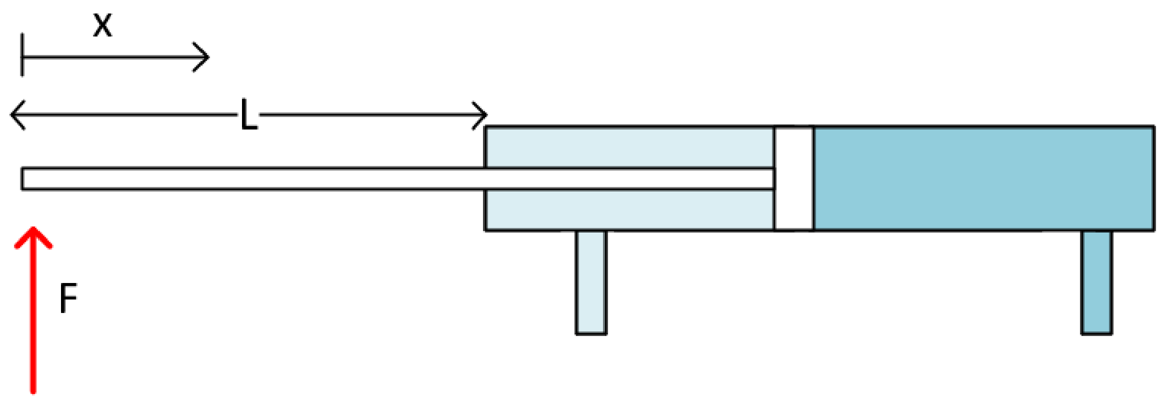


Figure 16 Horizontal installed hydraulic cylinder subject to a vertical force due to vertical gate excitation

Bending deformations follow from bending moments induced in the piston. Hartsuijker & Welleman (2007, pp 543-544) describe a solution method that expresses the deformation following from the moment distribution(eq. 18). This method can be simplified in simple mechanical models, where as it can be widely applied to more difficult compositions as well.

$$-\frac{d^2w}{dx^2} = \frac{M(x)}{EI} \tag{18}$$

Using an expression for the moment that is related to the external forcing, equation 18transforms to equation 19 (Hartsuijker et al, 2007, p557).

$$-EI\frac{d^4w}{dx^4} + q = 0 \tag{19}$$

For simple cases this differential equation is already elaborated and expressed in so called ‘forget-me-nots’. In case of a cylinder as depicted in Figure 16the bending may be expressed as given in equation 20.

$$F = \frac{3EI}{l^3}w \rightarrow k_{bend,fixed\ end} = \frac{3EI}{l^3} \tag{20}$$

4.3.3 **Stiffness due to deformation cylindrical components/tubes**

The hydraulic cylinder is equipped with tubes that transport the fluid from a source towards the hydraulic cylinder to enable it to move and build up pressures inside the cylinder. These tubes have a circular shape and increasing pressures will lead to deformation of these tubes. Like the case for axial deformation, the pressure can be expressed as a normal force leading to strain in the tube. The relation between these two is the stiffness of the component. In the case the tubes do not deform under pressure exerted by water, the water has to deform itself. This can be expressed by using the bulk modulus approach as given in §4.3.1.

$$B = -\frac{\frac{1}{4}\pi D^2 * L}{\frac{1}{4}\pi D^2 y} * \frac{F_p}{\frac{1}{4}\pi D^2} \rightarrow F_p = \frac{\frac{1}{4}B\pi D^2}{L}y_{tube} \tag{21}$$

In which y_{tube} is the displacement of the water column inside the tube due to a deformation inside the hydraulic cylinder. The total volume displaced should be equal in case of a very stiff cylinder and tube. This leads to the following expression.

$$y_{tube} = \frac{R^2_{cylinder}}{R^2_{tube}} y \tag{22}$$

Substituting this value leads to an expression for the stiffness of the fluid inside the tubes.

$$k_{tube} = \frac{B_{fluid}\pi}{L_{tube}} R^2_{cylinder} \tag{23}$$

In may well be true that expansion of the tubes themselves will occur. Nevertheless for computational reasons this method of modelling is chosen. It should be further investigated which stiffness component is dominant, fluid compression or radial expansion of the material.

4.3.4 Added water stiffness

Kolkman and Jongeling (1996) define three types added water stiffness:

1. Partly submerged body in water – Archimedes
2. Flow stiffness
3. Sudden stiffness

The most common types that are experienced in vertically vibrating gates are flow stiffness and sudden stiffness. Nevertheless, Archimedes might also play a role in the gate dynamics and will therefore be treated as well.

Partly submerged body in water

Stiffness is directly obtained by oscillations of the gate in free surface water. The force exerted on a structure in water can be directly derived from the law of Archimedes.

$$F_{Archmedes} = \rho_w g V_{displaced} \tag{24}$$

The displaced water volume is equal to the volume of the gate below the water surface.

$$V_{displaced} = L_{gate} b_{gate} d_{gate} \tag{25}$$

In which d_{gate} is taken as the equilibrium depth of the gate for which it floats without any oscillations. Taking the gate out of equilibrium by a excitation y, we obtain the following formula for Archimedes.

$$F_{Archmedes} = \rho_w g L_{gate} b_{gate} y \tag{26}$$

The added stiffness can now be retrieved directly and reads:

$$k_{w-Arch} = \rho_w g L_{gate} b_{gate} \tag{27}$$

Flow stiffness(drag)

Kolkman & Jongeling (1996) state that the stiffness due to flow around a gate is proportional to the hydraulic head, and therefore proportional to the flow velocity squared. They propose a formula for a stiffness that is derived by using a flow force exerted in the vertical(eq. 28). This formula already shows that the stiffness only holds for gates in which a force in the vertical is induced.

$$k_{w,flow} = \frac{\partial F}{\partial y} \tag{28}$$

The term ‘F’ in equation 28 is the flow force that is exerted on the gate opening. This flow force depends on the gate opening itself, since this determines the flow velocity underneath the gate.

$$F_{flow} = \frac{1}{2} C_f \rho v_{gate}^2 A_{gap} \tag{29}$$

In which the flow velocity v may be expressed in terms of known discharge and gate opening.

$$v_{gate} = \frac{Q}{W\delta(t)} = \frac{Q}{W * \left(\delta_0 + y(t)\right)} \tag{30}$$

In which:

δ_0 = Initial gate opening
W = Width of the culvert or channel
Q = Discharge
C_f = Drag coefficient for submerged rectangular structure in flowing water

Sudden stiffness (inertia)

Flow inertia is determining this type of stiffness. In case of flow that cannot follow the gate opening variations, one may experience sudden stiffness. Kolkman & Jongeling (1996) describe the following relation for the sudden stiffness.

$$k_{w,sudden} = \frac{2F_{flow,static}}{\delta_0} \tag{31}$$

In which $F_{flow,static}$ is the force that acts on the gate due to the flow velocity. The sudden stiffness can either be negative or positive, depending on the direction of the flow force. This added stiffness will mainly act in the horizontal.

4.4 Analytical methods for dynamic problems

Several methods for describing the excitations of systems that encounter dynamic forcing are derived. A single degree of freedom(SDOF) system, as given in Figure 28, is best solved using the time domain or frequency domain. In case of multiple dynamic forcing frequencies one might obtain the dominant frequencies from a Fourier transform, which represents data into the frequency domain solely.

The solution procedures will focus on a SDOF system only. This paragraph will elaborate the equation of motion briefly where after more detailed explanations about the two relevant solution methods is given.

4.4.1 Equation of motion

The equation of motion(EOM), based on Newton’s 2nd law, is yet given in §4.4.1 and reads:

$$m_{tot}\ddot{y} + c_{tot}\dot{y} + k_{tot}y = F(t) \tag{32}$$

In which:

m_{tot} = mass of the system and the added water mass
c_{tot} = damping of the system and added water damping
k_{tot} = stiffness of the system and added water stiffness

The EOM can be recognized as a second order differential equation. The general solution for y depends on the solution method that one uses. These are described in the upcoming paragraphs.

4.4.2 Solution in the time domain

Rewriting the EOM into its canonical form, we derive at an expression given in equation 33.

$$\ddot{y} + 2\zeta\omega_n\dot{y} + \omega_n^2 y = \frac{F(t)}{m_{tot}} \tag{33}$$

In which:

ζ = Damping ratio (paragraph 5.5.2)
ω_n = Natural frequency (paragraph 5.5.1)

Metrikine (2006) describes a method which requires to first find a solution for the homogeneous solution, where after a particular solution is found. The homogeneous solution is given in its basic form in equation 34 (Metrikine, 2006). It is found by assuming a general solution that fits the differential equation. For the time domain this solution has the form of an exponential function (eq. 35)

$$y_h(t) = \sum Y\, e^{st} \tag{34}$$

$$y_h(t) = e^{-\zeta\omega_n t}\left(Y_1 e^{\omega_n t\sqrt{\zeta^2-1}} + Y_2 e^{-\omega_n t\sqrt{\zeta^2-1}}\right) \tag{35}$$

Depending on the forcing type, a particular solution can be assumed which is then substituted into the EOM. The final formulation for the displacement of the system is obtained by adding the two solutions and find the unknown coefficients using initial conditions. Perhaps the most general way to solve the total vibration pattern is to make use of the Duhamel integral. This integral represents all kinds of forcing methods and can be widely applied. Equation 36 gives the solution of the vibrations when using the Duhamel integral (Metrikine, 2006).

$$y(t) = e^{-\zeta\omega_n t}\left(Y_1 e^{\omega_n t\sqrt{\zeta^2-1}} + Y_2 e^{-\omega_n t\sqrt{\zeta^2-1}}\right) + \frac{1}{m\omega_1}\int_0^t F(\tilde{t})e^{-\zeta\omega_n(t-\tilde{t})}\sin\left(\omega_1(t-\tilde{t})\right)d\tilde{t} \tag{36}$$

In which:
F(t) = Forcing on the system
ω_1 = $\omega_n\sqrt{1-\zeta^2}$

4.4.3 Solution in the frequency domain

Metrikine and Tsouvalas (2019)[1] describe a method for solving the problem in the frequency domain. Using a Fourier transform, the time dependent vibrations can be transformed to frequency dependent vibrations.

$$\begin{bmatrix} x(t) \\ F(t) \end{bmatrix} = \frac{1}{2\pi}\int_{-\infty}^{\infty}\begin{bmatrix} \tilde{x}(\omega) \\ \tilde{F}(\omega) \end{bmatrix} e^{i\omega t}d\omega \tag{37}$$

They propose the following EOM, which can be rewritten in terms of the frequency spectra, which is depicted in equation 38.

$$(-\omega^2 + i\omega c + k)\tilde{x}(\omega) = \tilde{F}(\omega) \tag{38}$$

[1] Lecture slides available on student communication platform Brightspace which requires a log in account available for students and employees from the TU Delft.

From this equation the time dependent vibration can be immediately obtained via an inverse Fourier transform.

4.5 Dynamic characteristic a system

Characteristics of the system that is examined may express the vibration patterns and excitations in a qualitative way. Without performing extensive calculations, one might already make solutions based on the characteristic of a system in combination with the forcing. The most important dynamic characteristics are defined to be:

- Natural frequency (ω_n)
    - Predicts whether or not the system is subjected to resonance.
- Dimensionless Damping (ζ)
    - Predicts whether the system is dynamically unstable and how vibrations are damped out.

4.5.1 Natural frequency

The characteristic of a system that determines whether a gate or valve enters the resonance regime is the natural frequency. It depends on the mass of the system and the stiffness. Both the added mass and stiffness should be included in case of a gate or valve that interacts with water. Equation 39 gives a relation of which the natural frequency can be determined.

$$\omega_n = \sqrt{\frac{k_{system} + k_w}{m_{system} + \sum m_w}} \tag{39}$$

In which:

m_{system} and k_{system} = mass and stiffness of the system in absence of water

m_w and k_w = added water values for mass and stiffness

4.5.2 Dimensionless damping

The rate of damping of the system, expressed in Figure 17, determines on the dimensionless damping. It is defined as the ratio of the damping present in the total system over the critical damping(Kolkman et al, 1996).

$$\zeta = \frac{c}{c_{crit}} = \frac{c_{system} + \sum c_w}{2\sqrt{(m_{system} + \sum m_w) * (k_{system} + \sum k_w)}} \tag{40}$$

Korevaar (2016) describes in his master thesis that the dimensionless damping can be divided into five categories.

- $\zeta < 0$: Negative damping, self-excitation vibration grow in time
- $\zeta = 0$: No damping is present in the system
- $0 < \zeta < 1$: Underdamped system, vibrations die out gradually
- $\zeta = 1$: Critically damped system, no vibration are observed and system return to equilibrium
- $\zeta > 1$: Overdamped, system return to equilibrium fast

4.5.3 Resonance

In case of a dynamic force that excites the structure with a forcing frequency close to the natural frequency, the structure experiences resonance vibrations. Depending on the damping ratio, these vibrations lead to extensive excitations compared to regular vibrations originating from the same forcing value.

Single forcing frequency

Considering a harmonic force, which is often seen in flow induced vibrations, we might express the ratio of the resonance amplification as a function of the natural frequency, forcing frequency and the damping ratio. Metrikine(2006, p. 39) proposes the following relation(eq. 41). This relation is known as the dynamic amplification factor.

$$\frac{|X|}{x_{static}} = \frac{1}{\sqrt{\left(1 - \frac{\omega^2}{\omega_n^2}\right)^2 + \zeta^2 \frac{\omega^2}{\omega_n^2}}} \tag{41}$$

In which:

x_{static}	= Static displacement	= F/k
X	= Steady state vibration amplitude	
ω	= Forcing frequency	

This relation may be expressed in a graph, which is given in Figure 17. The damping factor is included by means of different lines that represent different values for the damping factor.

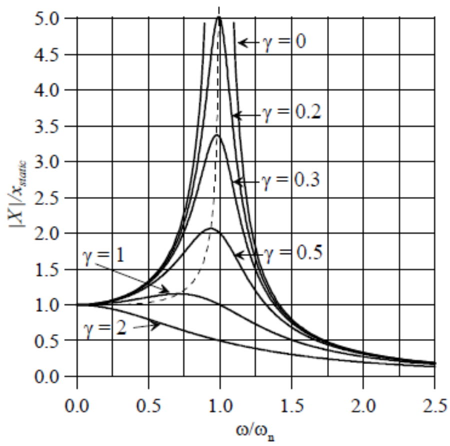


Figure 17 Dynamic magnification factor versus ration of natural frequency and forcing frequency (Metrikine, 2006, p. 39)

Range of forcing frequencies

In case of multiple forcing frequencies one may obtain a range of excitation frequencies of the system. Resonance may then occur in the case where the natural frequency is located in this band. When a system is triggered in multiple frequencies, a Fourier transform of the obtained data will give insight which frequency attracts the most energy.

4.6 Dynamic forcing on vertical vibrating valves

Flow induced vertical vibrations are the cause of five specific excitation sources (Kolkman et al, 1996):

- Drag force due to flow
- Turbulent flow
- Vortex shedding
- Flow instability due to instable separation point (§3.1.1)
- Self-excitation in form of negative suction damping (§4.2.4)

As for the other flow induced sources, for these it also holds that the main cause of flow forcing is the head difference over the gate or valve and the corresponding flow velocity. In this paragraph several mathematical approximations are made to link the hydraulic head difference to certain forcing patterns. These may be used in deriving basic design rules for future design.

4.6.1 Overview flow induce vibration sources

As already discussed in paragraph 3.1.1, five different sources are recognized for flow induced vibrations. three of which are discussed in detail in this section and a mathematical link between them and the hydraulic head is established.

The sources that initiate vertical vibrations are listed in Table 2. Most of them are related to the flow velocity underneath the gate. The flow velocity underneath a gate is on its turn directly coupled to the local energy head difference over a gate or valve. This will be elaborated per specific force.

Table 2 Excitation sources and governing variables

Excitation source	Governing variables
Flow drag	- Flow area - Flow velocity - Drag coefficient (geometry)
Turbulent flow	- Amplitude velocity fluctuation
Vortex shedding	- Flow velocity - Forcing frequency
Flow instability	- Width of gate - Relative gate opening - Radius of circular gate - Flow velocity
Self-excitation	- Flow velocity - Natural frequency - Relative gate opening

4.6.2 Flow drag

The force experienced by the structure is known as the flow resistance and can be obtained via (NASA, n.d.):

$$F_{R-flow} = \frac{1}{2}\rho A_{gate} v^2 * C_w \tag{42}$$

In which C_w is coefficient of drag and is usually in the order of 1.1 – 1.3 for flat plates in flowing water(NASA, n.d.).

In case of an inclined edge of the gate the horizontal drag force may lead to a vertical directed force. Decomposing the force to a perpendicular force on the edge, gives rise to the magnitude of the vertical flow force.

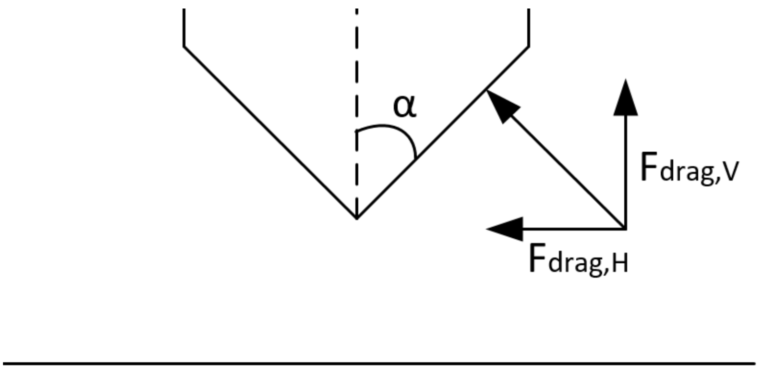


Figure 18 Decomposition of horizontal drag force

The vertical drag force may be calculated using the inclination angle of the lower edge of the gate.

$$F_{drag,V} = \tan(\alpha)\, F_{drag,H} = \frac{\tan(\alpha)}{2} \rho A_{gate} v^2 * C_w \tag{43}$$

4.6.3 Turbulence

Excitations based on turbulence has to do with pressure fluctuations underneath the gate itself. Turbulent flows cause fluctuations in velocities which on their turn cause fluctuations in the local energy head based on Bernoulli's principle. The pressure differences encountered are in the velocity fluctuation range of 10% (Belfroid, 2017). Belfroid (2017) also proposes a relation(eq. 44) between the pressure differences and the velocity fluctuations in time.

$$p' = 0.01 * \frac{1}{2} \rho U^2 \tag{44}$$

4.6.4 Vortex shedding

Vertical gate or valve excitation is directly linked to vortices via the Strouhal number. The Strouhal number indicates at which frequencies the vortex is generated(Kolkman et al, 1996). This can be understood as the forcing frequencies of the vortices on the gate.

The Strouhal number on its turn is a ratio between the forcing frequency and the flow velocity. In general the governing guideline is to keep a sufficient buffer between the dominant forcing frequency and the natural frequency. Expressed in Strouhal number this leads to the following requirement(TNO, 2017):

$$S_n \approx (2-3)S \tag{45}$$

With:

S = Strouhal number using (dominant) excitation frequency

S_n = Strouhal number using natural frequency of the system

Detailed information on how the Strouhal number is used and calculated is given in §4.7.1. Belfroid(2017) gives some key information about the range of Strouhal numbers: "For bluff bodies (2D structures) typical Strouhal numbers are St = 0.2 with the effective length the thickness of the body."(p. 14). Mind that the Strouhal number is not a given constant on forehand and should always be calculated using the specific geometry of the design.

4.6.5 Flow instability

Unstable separation points lead to fluctuations in pressure underneath the gate. In case of a stable separation point and a stable reattachment point, the gate may also experiences fluctuating pressures. The point of separation and reattachment are mainly governed by the geometry of the edge of the gate on which the flow acts.

Martin et al (in Kolkman et al, 1996) state that the geometry of a gate is crucial in case of a stable reattachment point. In case of a rectangular shaped edge, Kolkman et al(1996) propose the following requirement for the width of the gate in order to prevent pressure fluctuations.

$$b = \frac{\frac{1}{2}(n - 0.33)V}{f_{excitation}} \tag{46}$$

With:

n = integer = 1,2,3,…
V = Flow velocity
f = Excitation frequency

In case of a circular shape edge Naudascher (in Kolkman et al, 1996) propose a radius that is greater than the gate thickness and TNO (2017) adds a sloping edge with an angle greater than 45 degrees in case of an inclined edge to the design requirements.

4.6.6 Self-excitation

Negative damping allows a gate to increase the excitation under a constant dynamic force. The added water damping and flow velocity may contribute and eventually lead to a negative damping of the system. In case of a negative added water damping that exceeds the damping of the system itself, we speak of self-excitation. Self-excitation evolves from several principles which are known to be:

1. Negative damping due to suction
2. Galloping

Suction

The added water damping is directly related to the vibrations of the structure. Varying flow gaps lead to a suction force within the gap. Equation 10 gives a relation for the suction damping coefficient in terms of the negative added damping and the flow velocity in the gap. Figure 19 gives experimental results from Kolkman & Vrijer (1977) from which the negative water damping due to suction can be obtained. Mind that increasing the damping of the system itself is an effective measure for preventing self-excitation. Furthermore it can be conclude from Kolkman &Vrijer (1977) that the range of Strouhal numbers for negative water damping to occur is rather small. The range of critical Strouhal number varies between 0 and 0.2. For a detailed description of suction damping the reader is referred to §4.2.4. In this section a detailed description of figure 19 is given.

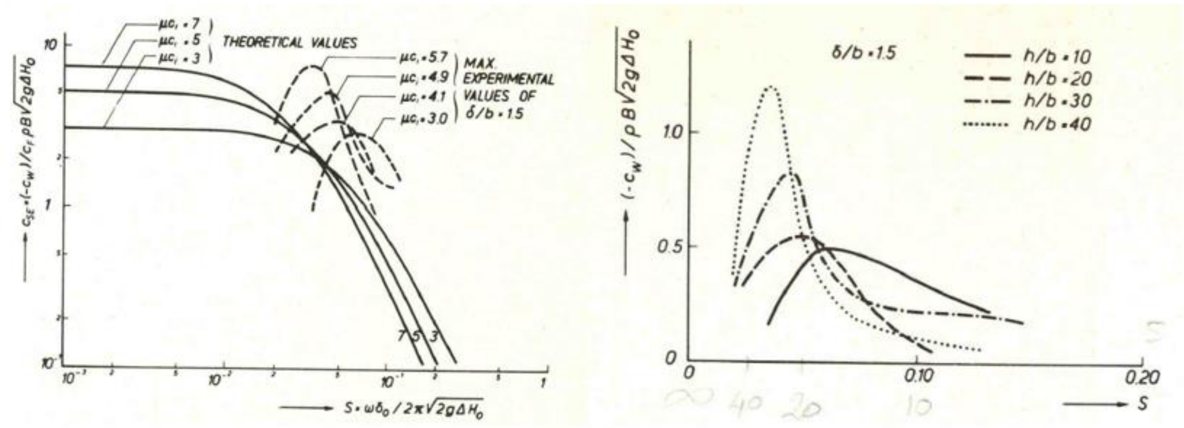


Figure 19 Experimental results of Kolkman and Vrijer(1977) for negative suction damping coefficient. Left: coefficient of self-excitation vs Strouhal number, right: negative suction damping vs Strouhal number

Galloping

Naudascher (as cited in Belfroid, 2017) gives several methods for solving the dynamic instability of structures subjected to galloping. In case of a submerged gate Naudascher proposes to have a look at the graph of the lift coefficient and the relative opening of the gate. The lift coefficient for such an submerged gate is given in Figure 20. The general method of dealing with galloping is to check whether the Strouhal number experienced by the gate is small enough compared to the Strouhal number of the system itself. This can be express in a reduced velocity threshold value (Naudascher, as cited in Belfroid, 2017), given in equation 47.

$$V_r = \frac{V}{f_n e} < \frac{4\pi}{\frac{dC_y}{d\left(\frac{S}{e}\right)}} S_c \tag{47}$$

in which:

e	= Gate width	
f_n	= Natural frequency of the gate	
S_c	= Scruton number	$= \frac{2m\zeta}{\rho d^2}$
m	= Mass of the system	
d	= Thickness of the gate	
ζ	= Damping ratio of the system	
$\frac{dC_y}{d\left(\frac{s}{e}\right)}$	= Slope of graph lift force vs relative gate opening (Figure 20)	

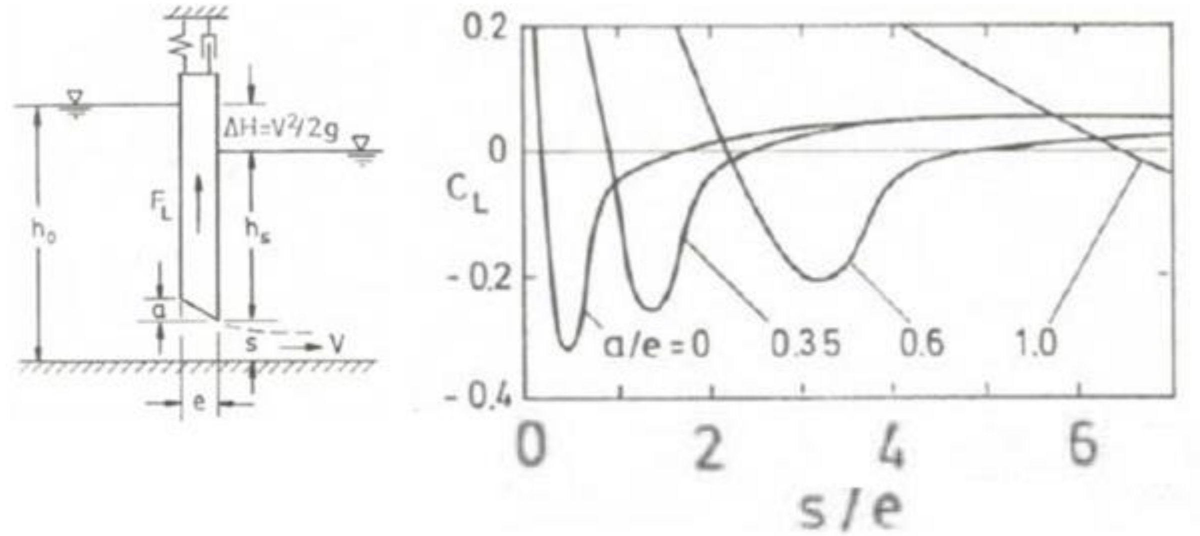


Figure 20 Symbols and schematization for a gate experiencing underflow (left) and lift coefficient as function of relative gate opening s/e (right) (Naudascher & Rockwell, 1994, pp. 209)

4.7 Relevant dynamic flow parameters

4.7.1 Strouhal number

The Strouhal number can be seen as an important parameter in defining vertical flow induced vibrations. It is defined as the dimensionless ratio of the frequency and the flow velocity(Kolkman et al, 1996).

$$S = \frac{fL}{V} \tag{48}$$

In which:

f = Excitation frequency
L = Length parameter: Gap height, gate thickness etc..
V = Flow velocity underneath the gate

Under some specific conditions the Strouhal number is a constant value:

1. In case of flow of a Reynolds number greater than $3x10^4$ (Korevaar et al, 2017, pp. 33)
2. In case of a valve or gate that is completely submerged under water (Kolkman et al, 1996, pp. 84)

When these requirements are not met, the Strouhal number cannot be taken as a constant. Naudascher (in Kolkman et al, 1996, pp. 86) give experimental results for the Strouhal number underneath a gate, subjected to underflow and vertical vibrations (Figure 21). The Strouhal number describes the periodic fluctuations of the flow underneath a gate. This results in periodic forcing in terms of turbulence and therefore relates to the phenomena of flow drag, turbulence and vortex shedding.

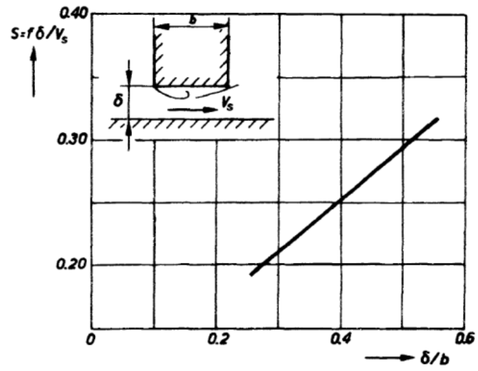


Figure 21 Strouhal number for vertical vibrating gates (Naudascher, as citied Kolkman et al, 1996, pp. 86)

The linear relation of the Strouhal number and the ratio of gate opening height and gate width can be expressed in a formula as given in equation 49.

$$S = 0.5 * \frac{\delta}{b} + 0.05 \quad ; \; 0.2 < \frac{\delta}{b} < 0.55 \tag{49}$$

4.7.2 **Flow velocity**

Given a discharge and a gap height, one might find relatively easy the velocity in the gap. More interesting is to express the velocity in terms of hydraulic heads, since this is a well-known quantity for hydraulic gates and valves.

Flow velocity is initiated by head difference between two reference points. In case of a pipe and sudden flow expansion head loss may occur, which reflect in the flow velocity as well. The flow velocity can be expressed using Torricelli's law (eq. 50)

$$V = \sqrt{2g\Delta H} \tag{50}$$

In which:

V = Flow velocity
ΔH = Head difference

The head difference exists out of the difference between the water levels over the gate and the head losses that the flow experiences.

$$\Delta H = H_{up} - H_{down} - H_{loss} \tag{51}$$

In the specific case of sluices the head loss is negligible, which can be retrieved from the fact that the water level on both sides of the gate should become equal. The flow velocity may therefore be described using equation 50.

4.8 Summary

Chapter 5 gives insight in the different dynamic components, forces and excitation sources that play a role in the dynamic behaviour of vertical vibrating valves subjected to underflow. It was shown that relevant dynamic quantities can be subdivided into the quantities belonging to the system itself, to the driving mechanisms and FSI induced quantities.

The FSI quantities are largely depending on the flow velocity of the fluid that travels underneath the gate. It is also found that FSI is largely based on empirical results and only show guidelines for a dynamic design in terms of excitation frequency and natural frequency, often expressed in terms of the Strouhal number.

The driving mechanism, hydraulic cylinder, generates additional stiffness to the system. The damping generated by the cylinder is often assumed to be negligible. If damping is generated it mostly belongs to the material damping of the cylinder and in a smaller demand to the friction induced between the piston and cylinder. Viscous damping is not found in the cylinder due to the sealing between the two compartments. Fluids are not allowed to transport themselves between different compartments, which excludes viscous damping.

4.8.1 Brief overview of dynamic components

Figure 22gives an overview of a vertical sliding valve that is excited by under flowing water. The damping and stiffness components are clearly sorted into mechanical, system and FSI components.

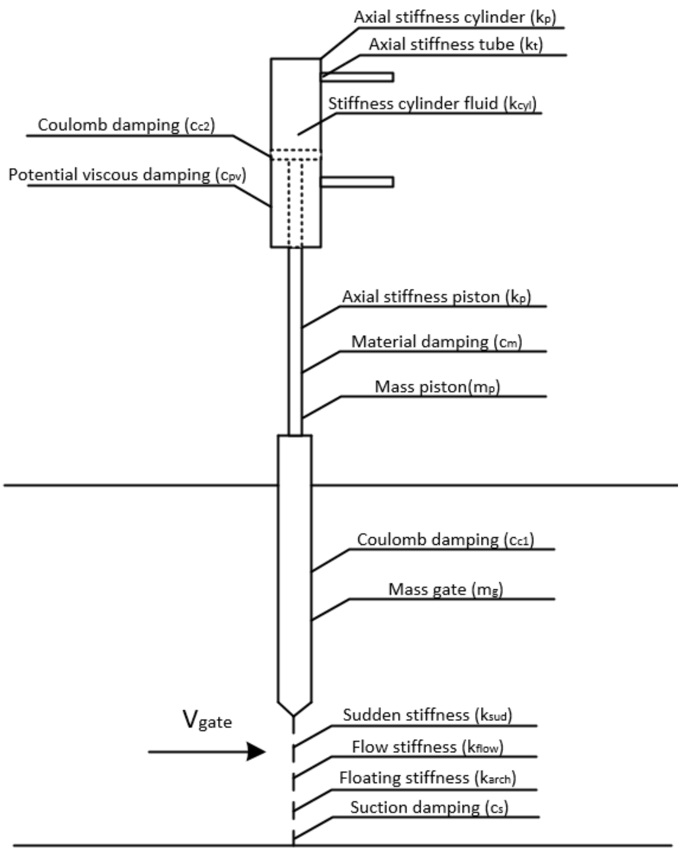


Figure 22 Dynamic components of a vertical sliding valve including a hydraulic cylinder

The values for the different components can be found in the previous sections of the report. For easy reference a brief overview is given in Table 3.The potential viscous damping that is mentioned in Figure 22 and Table 3 is not taken into account in the different analyses elaborated in chapter 5. This is solely depicted to give an overview that matches the reality as good as possible.

Table 3 Overview dynamic components and corresponding equations or paragraphs

Component	Paragraph/Equation number
System	
- Mass gate	Input known
- Coulomb damping	Equation 7
Driving mechanism	

-	Mass piston	Input known
-	Material damping piston	4.2.3
-	Axial stiffness piston	Equation 17
-	Coulomb damping cylinder	Equation 7
-	Axial stiffness cylinder	Equation 17
-	Fluid stiffness cylinder	Equation 15
-	Potential viscous damping cylinder	Absent in the system
Fluid Structure Interaction		
-	Added water mass	Equation 4
-	Suction damping	Equation 10
-	Floating stiffness	Equation 27
-	Flow stiffness	Equation 28
-	Sudden stiffness	Equation 31

4.8.2 **Brief overview of dynamic forces**

Dynamic forces from flowing water that are acting on the structure are often related to the flow velocity. The distinguished external forces listed and their dependence can be formulated as follows:

- $F_{flow} = F_{shedding}$ $\sim v^2$
- $F_{turbulence}$ $\sim v^2$

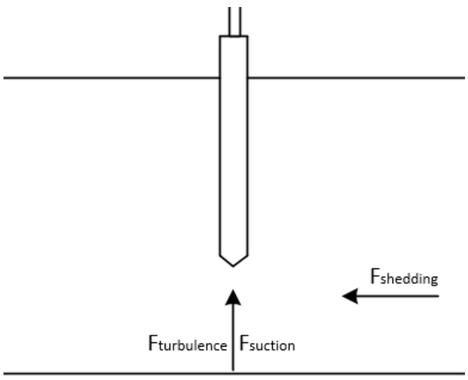


Figure 23 External dynamic forces

External forces may be modelled by a time dependent pressure dataset which can be either easily rewritten to an external force or to a flow velocity. Flow velocity is obtain by the assumption that the energy head is constant in time, which leads to the Bernoulli relation for conservation of energy. In case of a known head upstream, one might find a relation between the pressure and flow velocity by means of equation 52.

$$\frac{p}{\rho g} + \frac{v^2}{2g} = constant \tag{52}$$

5 Methodology

This chapter will elaborated the method of research that is applied in this additional thesis. A case study is examined based on a vertical sliding valve that is currently used in a hydraulic structure in the Netherlands which cannot be mentioned for privacy reasons. Default values for the dimensions are taken from this structure. The valves are analysed by means of a python script which contains the different relations given in previous part of this report. First a general overview will be given where after more details about the valve and boundary conditions are clarified. The last paragraph will dive into the modelling approach and will give relevant formulas that are used in the script.

5.1 Hydraulic situation(s) considered – Case study

In this additional thesis the main focus will be on the vertical sliding gate that experiences a horizontal flow. Figure 24 - Figure 27 give a representation of the global design that is considered. The starting point of the considered systems is that they:

- Experience a horizontal flow
- Show vibrations in the vertical directions
- Experience excitation sources as described in §3.1.

Relevant parameters on which derivations of several dynamic quantities are made in the continuations of this report are given in the table below.

V_{gate}	Flow velocity under gate	L_{piston}	Length of piston
δ	Gate opening	L_{oil}	Length of oil
b	Gate thickness	$L_{cylinder}$	Length of hydraulic cylinder
W	Gate width	$D_{cylinder}$	Diameter of hydraulic cylinder
h_{gate}	Gate height	$V_{cylinder}$	Volume of hydraulic cylinder
ΔH	Energy head difference over gate		
h_1	Water depth upstream		
h_2	Water depth downstream		
E_{steel}	Young's modulus steel		

5.1.1 Structure under consideration

Vertical sliding valve

A lock gate with vertical sliding valves as a filling and emptying mechanism. The lower edge of the valve has an pointed shape that has an angle of more than 45 degrees to reduce the instable flow vibrations. Furthermore the gate is equipped with girders in the vertical direction to manage vibrations in the horizontal direction. This is not relevant in this additional thesis and will therefore not be incorporated. Figure 24 gives a schematic overview of the valve. The dimensions are assumed to be:

- Width of the valve W = 2000 mm
- Height of the valve h_{valve} = 1500 mm
- Thickness of the valve b = 250 mm

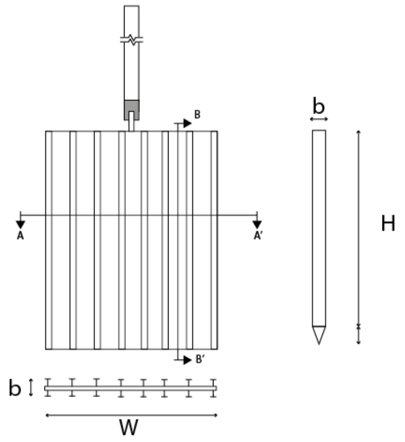


Figure 24 Geometry of the vertical sliding valve

The lower edge of the valve is chosen to have an inclined angle of 45 degrees to prevent flow instability to occur.

5.1.2 Driving mechanism

Lay out of hydraulic cylinder

The main driving mechanisms behind the vertical sliding valve is a hydraulic cylinder. Figure 25 and Figure 26 gives an impression of such an cylinder including important variables. The variables of the cylinder are parametrized in the final equations to observe the behaviour of different elements.

A hydraulic cylinder can be decomposed into several elements that all contribute to the stiffness and potential damping of the system:

- Pressurized cylinder (often oil as a fluid)
- Piston with connecting rod to the valve
- Tubes for transport of fluids towards the cylinder

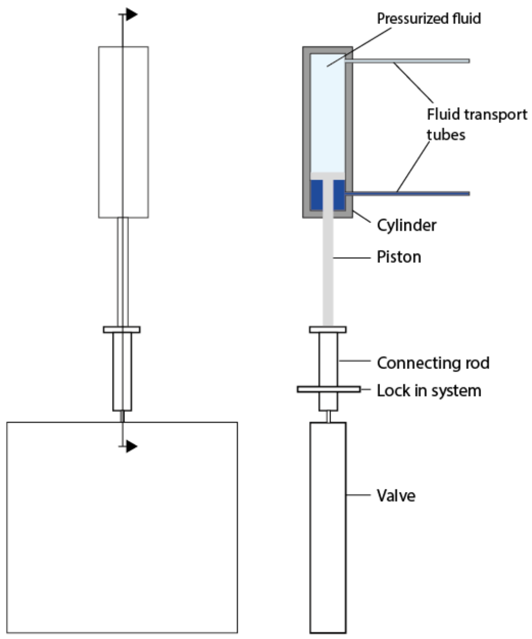


Figure 25 Schematic overview of the driving mechanism

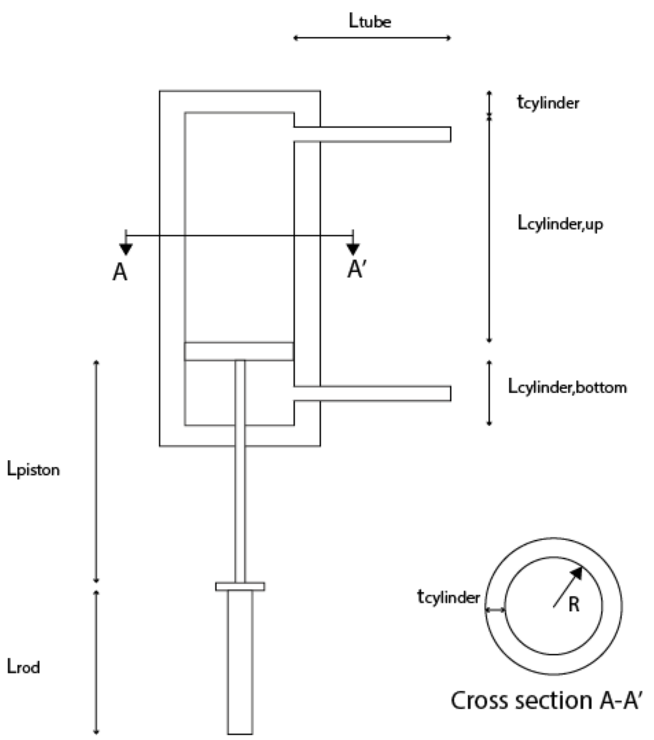


Figure 26 Relevant variables for dynamic quantities of driving mechanism

Relevant parametrized formulas according to the literature study in chapter 5 are given below.

$$k_{cylinder} = \frac{B\pi D^2}{4(L_{cylinder,up} + L_{tube})} \tag{53}$$

$$k_{piston} = \frac{E_{piston}A_{piston}}{L_{piston}} \tag{54}$$

$$k_{rod} = \frac{E_{rod}A_{rod}}{L_{rod}} \tag{55}$$

$$k_{tube} = \frac{B_{fluid}\pi}{L_{tube}} R^2_{cylinder} \tag{56}$$

$$c_{piston,material} = \zeta_{material}\sqrt{k_{piston}m_{piston}} \tag{57}$$

$$c_{rod,material} = \zeta_{material}\sqrt{k_{rod}m_{rod}} \tag{58}$$

In which:

B	= Bulk modulus of fluid	B_{oil}	= 1.5 GPa
E	= Young's modulus		
μ_{piston}	= Friction coefficient piston – cylinder		
ζ_{steel}	= Material damping coefficient steel		= 0.004
$\zeta_{aluminium}$	= Material damping coefficient aluminium		= 0.018

Relevant parameters for analysis

In order to obtain a sensitivity analysis, first the relevant parameters must be investigated in terms of dependency on dynamic quantities of the driving mechanism. Table 4 gives an overview of relevant parameters and which dynamic quantities they influence directly and indirectly.

Table 4 Relevant parameters driving mechanism - Hydraulic Cylinder

Parameter	Default value	Direct influence	Indirect influence
Rod length	3000 mm	- Stiffness rod - Damping rod	- Stiffness system - Damping system - Max gate opening - Strouhal number
Rod diameter	250 mm	- Stiffness rod - Damping rod	- Stiffness system - Damping system
Rod thickness	20 mm	- Stiffness rod - Damping rod	- Stiffness system - Damping system
Rod material	Steel	- Stiffness rod - Damping rod	- Stiffness system - Damping system
Piston length	3500 mm	- Stiffness piston - Damping piston	- Stiffness system - Damping system - Max gate opening - Strouhal number
Piston material	Steel	- Stiffness piston - Damping piston	- Stiffness system - Damping system
Piston diameter	100 mm	- Stiffness piston - Damping piston	- Stiffness system - Damping system
Piston thickness	50 mm	- Stiffness piston - Damping piston	- Stiffness system - Damping system
Cylinder fluid	Oil	- Stiffness cylinder	- Stiffness system
Cylinder length	330 mm	- Stiffness cylinder	- Stiffness system
Cylinder radius	250 mm	- Stiffness cylinder - Damping cylinder	- Stiffness system - Damping system
Cylinder thickness	20 mm	- Stiffness cylinder - Damping cylinder	- Stiffness system - Damping system
Friction piston - cylinder	0	- Damping cylinder	- Damping system

The range of which these variables are varying is given in terms of percentages of the default value and is listed below.

- Cylinder radius : -100% - 400%
- Cylinder length : -100% - 175%
- Rod thickness : -100% - 1200%
- Rod length : -100% - 350%
- Piston diameter : -100% - 1000%
- Piston length : -100% - 300%
- Tube length : -100% - 2000%

5.1.3 **Boundary conditions**

To obtain a velocity in the culvert of the gates, the water levels and geometry of the culverts must be known. Figure 27 gives a schematic overview of the gates in their equilibrium position. The gate opening is based on the filling and empty timeframes and the water volume that must be displaced. Furthermore the boundary conditions state some constant parameters listed below.

- Water level difference $\Delta H = 4$ m
- Gate opening height $\delta = 250$ mm
- Inclination angle $\alpha = 45^\circ$
- Friction coefficient gate $\mu_{dynamic} = 0.1$

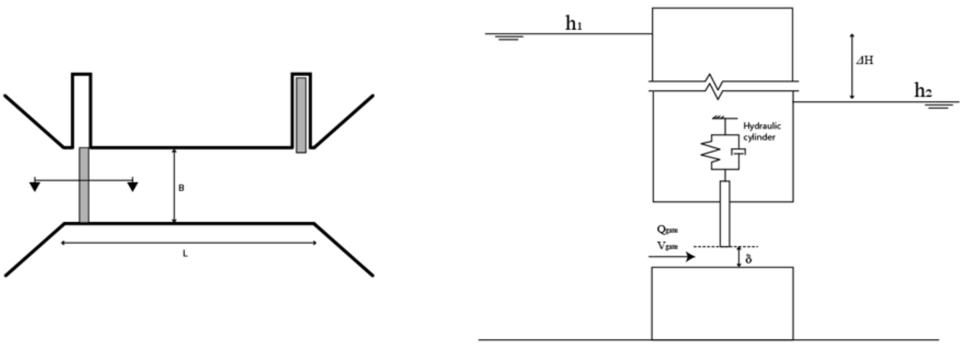


Figure 27 Top view of lock installation and cross section of gate including culvert and valve

Mind that these values are not used for computing pressure fluctuations but rather to give an impression on the order of flow velocities that may pass a sluice.

The relative importance of dynamic characteristics for varying boundary conditions(§6.3) takes a range of values as input. These are taken as:

- Water level difference $\Delta H = 0 – 8$ m
- Gate opening height $\delta = 0 - 2000$ mm

5.2 Modelling approach

Analysing the relative importance of different stiffness and damping components is done by means of analytical models. These models are implemented in Python and contain expression for the different damping and stiffness components as found in the literature study. By means of a fictional time varying forcing acting underneath the gate in the vertical direction, the FSI components can be calculated and compared to the dynamic components of the system.

5.2.1 **Modelling of vertical sliding valve**

Modelling of structures is the basis of a dynamic response analysis. Gates and valves that are driven by a hydraulic cylinder can be mainly modelled as a Single Degree of Freedom(SDOF) system. Figure 28 gives a representation of how such a model is constructed.

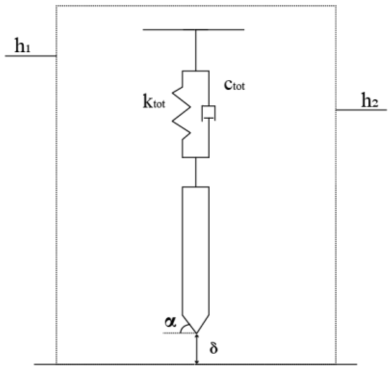


Figure 28 SDOF system to be considered

The total mass, damping and stiffness of the system, which are recognized by the subscript 'tot', can be subdivided into the characteristics of the system itself and characteristics due to the presence of water. Rewriting these characteristics we find:

$$m_{tot} = m_{gate} + m_{driving} + m_w \tag{59}$$

$$c_{tot} = c_{driving} + c_w + c_{friction} \tag{60}$$

$$k_{tot} = k_{driving} + k_w + k_{friction} \tag{61}$$

In the case of damping and stiffness, we find several components to be installed in series or parallel. Equations to obtain the total damping and stiffness in this case are given below.

$$c_{tot} = \sum c_i \tag{62}$$

$$k_{tot} = \frac{1}{1/\,k_{rod} + 1/k_{piston} + 1/k_{cylinder} + 1/k_{tube}} + k_w + k_{friction} \tag{63}$$

Elaborated in chapter 5, we find the following relevant characteristics of a system:

- Stiffness
- Damping
- Natural Frequency
- Damping ratio
- Dynamic amplification factor

5.2.2 **Model set up**

The model can be split up in several components: 1) Parameter determinations, 2) Formulas and 3) Calculations and plots. For all the analyses it is possible to transform a parameters into a range of variables to investigate the influence of that parameter. This holds for both the gate characteristics and the boundary conditions. Further information about the used python script can be obtained from the figures in appendix D.

5.2.3 **Model output**

Goal of the analysis is to obtain insight in the influence of the different driving mechanism components in terms of:

1. Damping of the system
2. Stiffness of the system
3. Resonance behaviour
4. Self-excitation

5. Dynamic amplification

The variables mentioned in Table 4 are checked on their sensitivity in the model schematization of a vertical sliding valve. Each variable is checked on its influence of the above mentioned 5 relevant subjects. The boundary conditions are also subjected to an analysis that investigates the behaviour of the different system characteristics for varying boundaries. This study will focus on the damping and stiffness of the different components of the system (Figure 22). A third analysis will focus on the influence of the elements on the natural frequency of the total system and will eventually give an optimal tweaking parameter as result.

6 Analysis

This chapter will focus on the analysis and the results from the different scripts that were made for this additional thesis. The following results will be displayed:

Sensitivity analysis	which element is most sensitive for damping and stiffness of the system
Relative importance	Which components are important for different sets of boundary conditions
Tuning	Results in an overview of components that mostly influence the natural frequency to a desired value

6.1 Dynamic characteristics reference situation

The reference situation exists of the default values for the cylinder geometry as listed in Table 4. Using these values we find the following system characteristics:

- Natural frequency ω_n = 44.435 rad/s
- Total damping c = 347747 Ns/m
- Total stiffness k = 12754707 N/m
- Damping ratio ζ = 0.606
- Excitation frequency ω = 66.794 rad/s

To get insight in the several stiffness and damping components, that lead to the total values, an overview of the magnitudes in given in the list below(see Figure 22 for elaboration of the different components) .

Damping (x 10^4 Ns/m)
$c_{coulomb}$ = 35
$c_{material}$ = 0.77
$c_{suction}$ = -1.77

Stiffness (x 10^6 N/m)
k_{rod} = 5527
k_{piston} = 2356
$k_{cylinder}$ = 96
$k_{archimedes}$ = 0.004
k_{flow} = 0.15
k_{sudden} = 0.188
k_{tube} = 14.7

6.2 Sensitivity analysis – results

This paragraph will present the results that are obtained during the sensitivity analysis. Each variable is examined in a range of values around its own default value as mentioned in Table 4Table 1. Each of the variables on its turn is analysed in terms of the 5 components in §6.2.1 - §6.2.8.

6.2.1 Case 1 - Hydraulic cylinder thickness

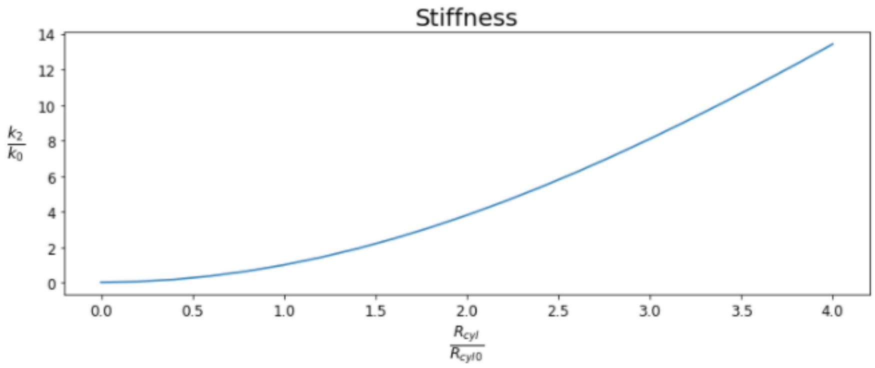
The thickness of the hydraulic cylinder in general doesn't influence the dynamic components of the system. The mass of the cylinder is increased, but this is not incorporated in the equations of motion since it is detached from the system.

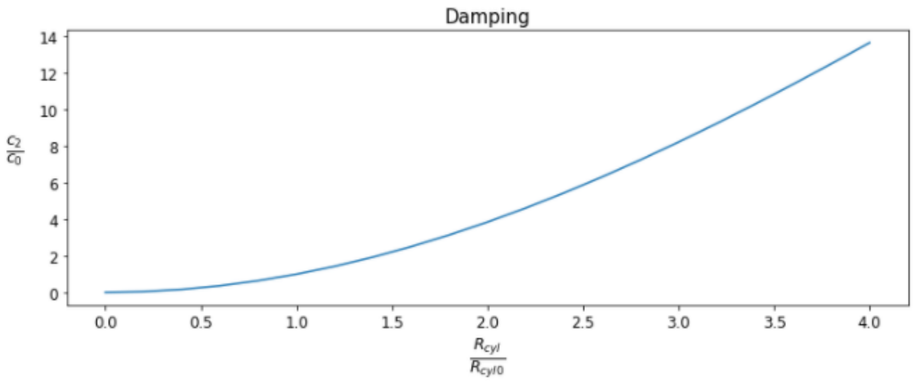
6.2.2 Case 2 - Hydraulic cylinder radius

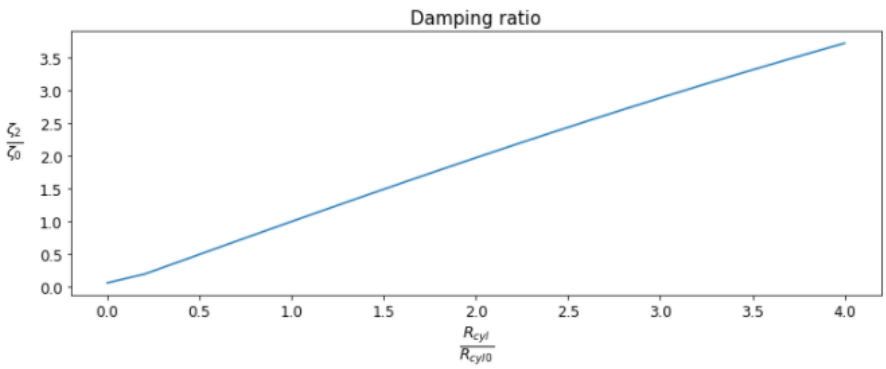
Varying the radius of the cylinder leads to a range of values for the stiffness, damping, mass and correspondingly to the natural frequency of the system. The variations is best expressed in graphs which are given in the graphs below. Overall it is observed that the stiffness and damping increase with a higher radius. Nevertheless we see that important parameters for dynamic design, the dynamic amplification, damping ratio and natural frequency, are increasing. The natural frequency thus tends to move further away from the forcing frequency corresponding to a Strouhal number of 0.3.

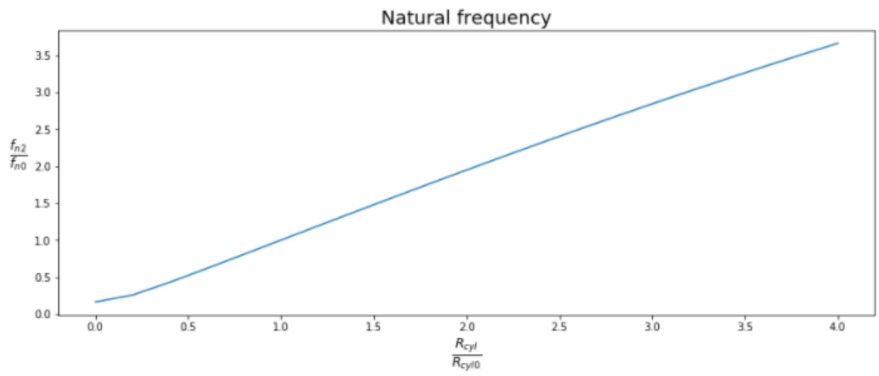
From the normalized graphs below, it can be observed that an increasing radius of the cylinder leads to an increase in the stiffness of the system. The damping is not affected. An increasing stiffness leads to the following:

- Increasing natural frequency.
- Reducing dynamic amplification factor, due to an increasing natural frequency.
- Peak in the dynamic amplification due to resonance, where after a steady decline is observed.







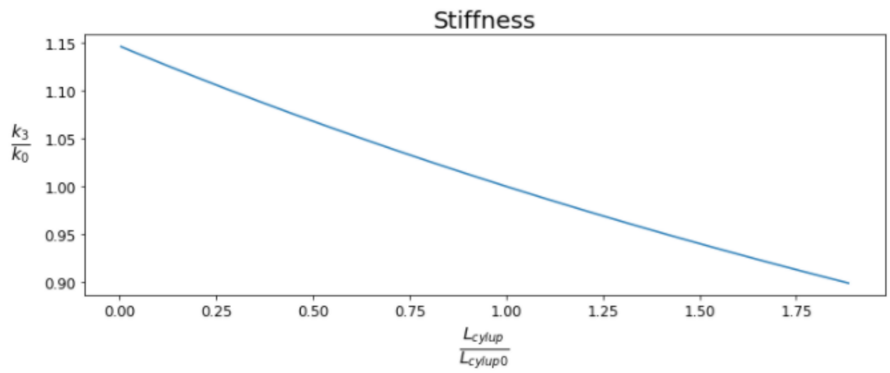


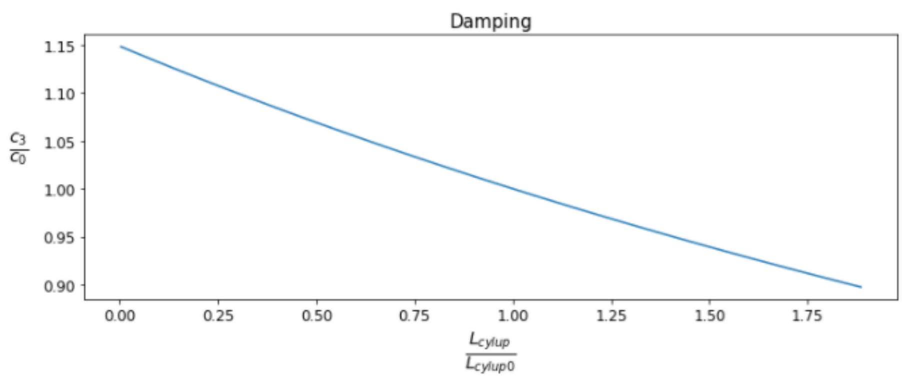
6.2.3 **Case 3 - Hydraulic cylinder length**

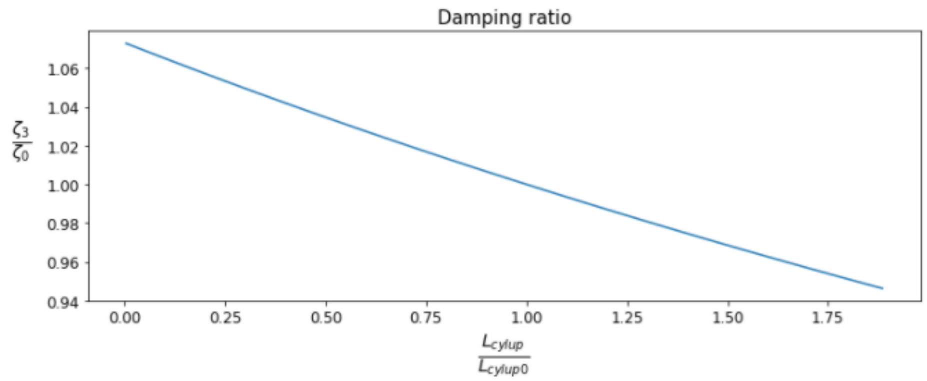
This case focusses on the length of the cylinder above the piston. This is denoted as $L_{cylinder,up}$ in Figure 26. It is assumed that the fluid tubes close off when the gate is in equilibrium position, thus not influencing the length of the cylinder.

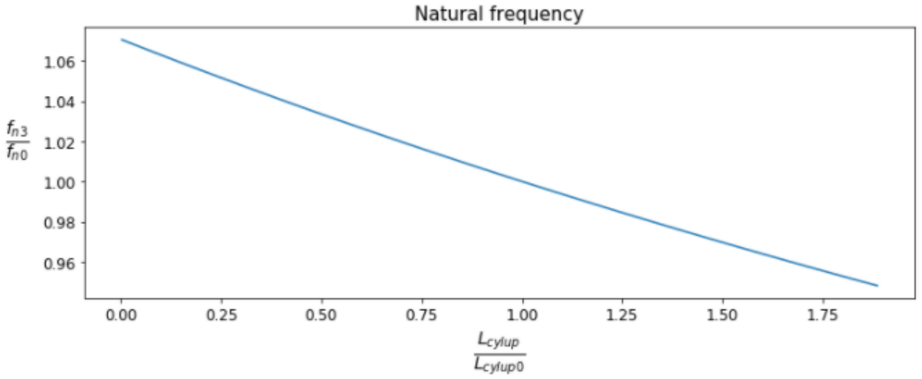
Again it is observed that the damping is influenced compared to the reference value, but more in a linear trend. The stiffness is also decreasing linearly with an increasing cylinder length. This might be declared by the fact that the volume of the cylinder is increasing, thus an increasing change in volume should be achieved for a constant bulk modulus. An decreasing stiffness results again in:

- Decrease of natural frequency, shifting towards the excitation frequency.
- Decrease in stiffness and damping
- Decrease in damping ratio.



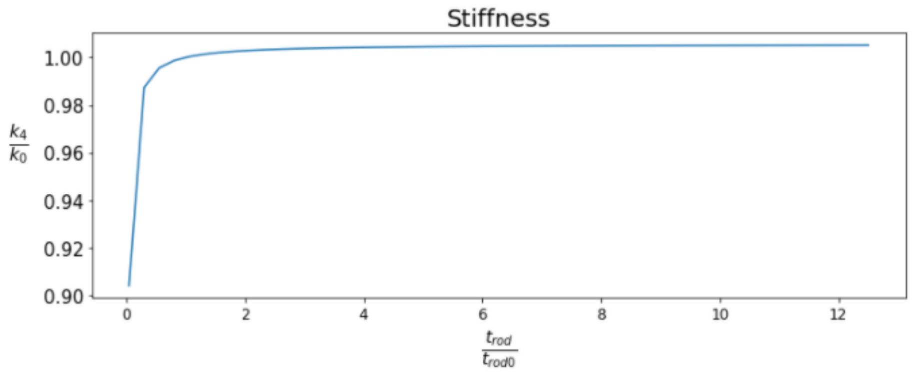


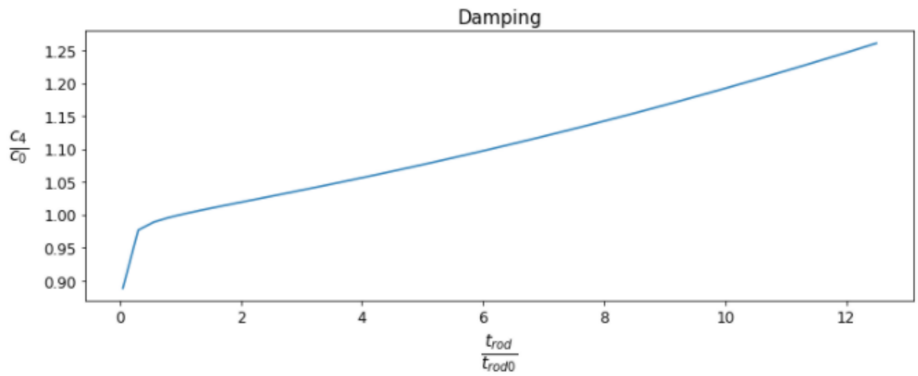


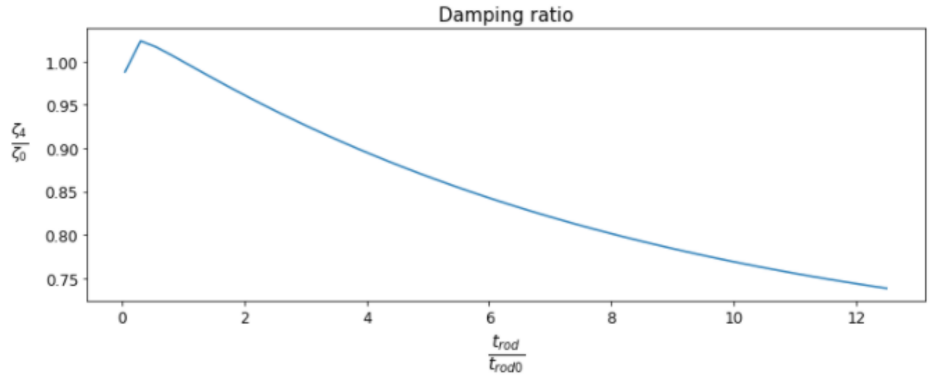


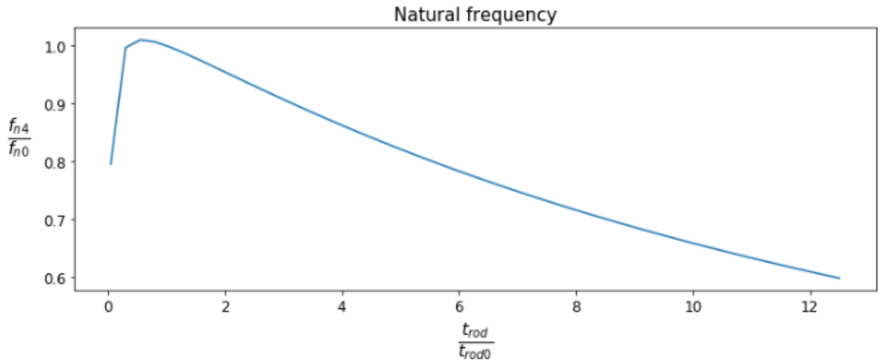
6.2.4 Case 4 - Rod Thickness

Adjusting the thickness of the rod leads to an interesting observation for the damping of the system. Since material damping is acting on the system, increasing the amount of material in the rod will lead to an increase in damping induced by the rod, which is best expressed as exponential for smaller values. The results of the sensitivity analysis show a more or less linear increase in damping with an increasing rod thickness for large dimensions. The damping ratio tends to decrease.



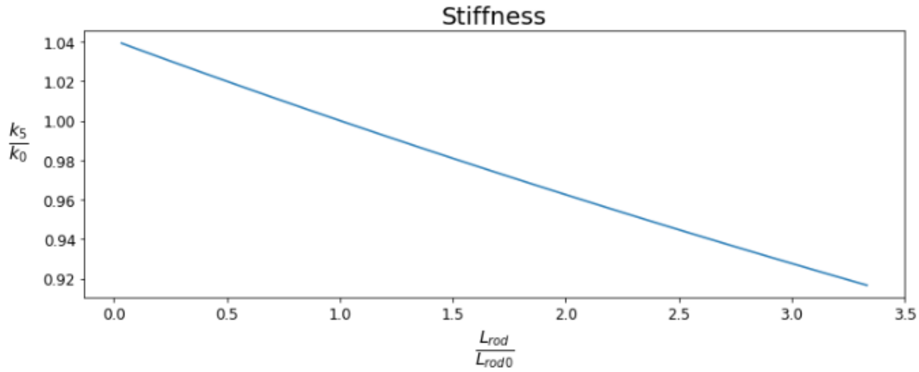


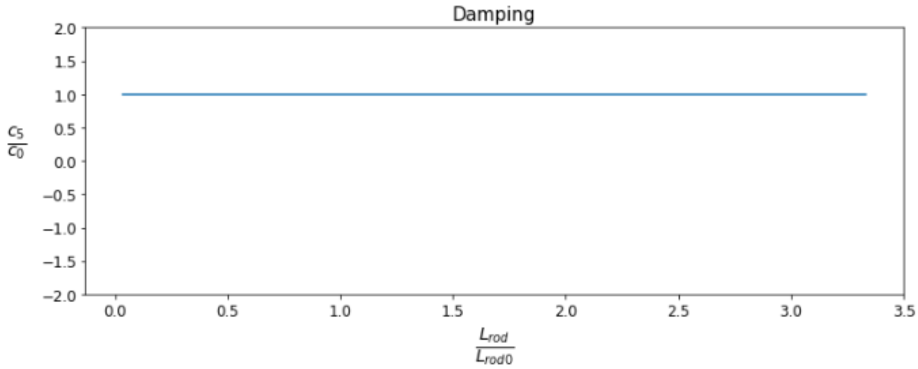


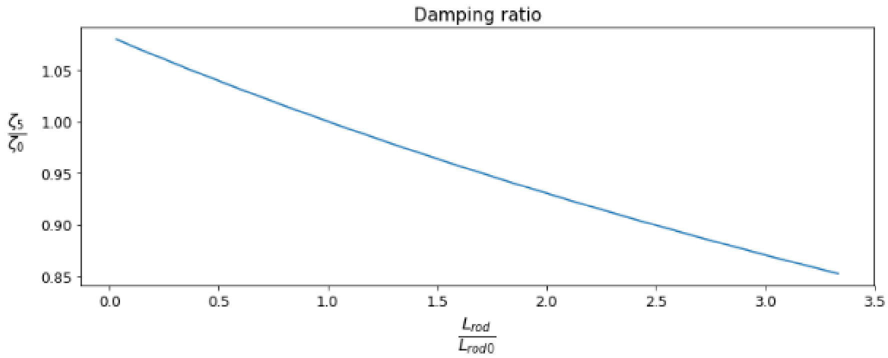


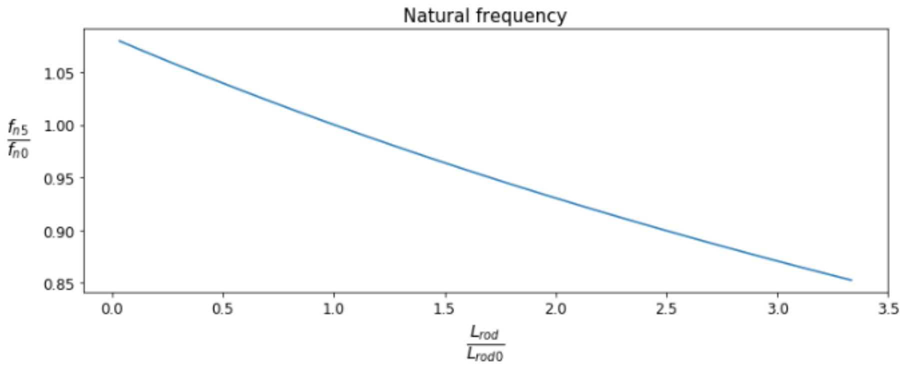
6.2.5 Case 5 - Rod length

Increasing the length of the rod leads to a slightecrease of the stiffness induced by the rod on the system. It turns out that a small rod induces a linear growth of the total stiffness of the system. This might be interesting for cases where the excitation frequency and the natural frequency of the system are close to each other. Simple measure like 'locking' the rod close to the valve leads to an increase in stiffness, and thus an increase of the natural frequency. The natural frequency shows a trend of an increase about 4 times the natural frequency of the reference case.



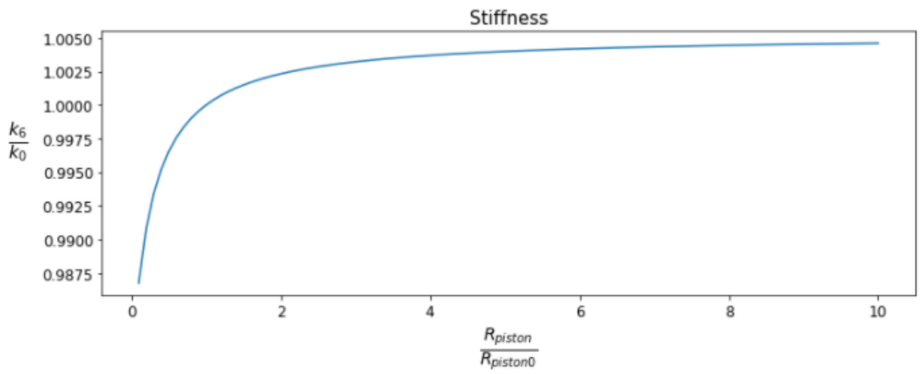


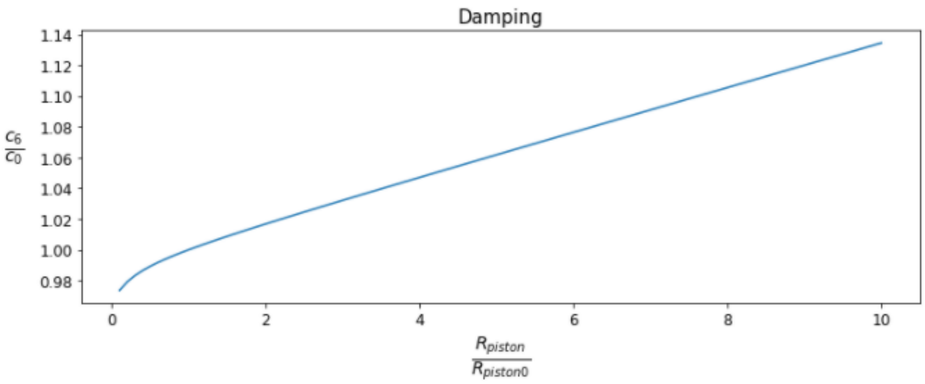


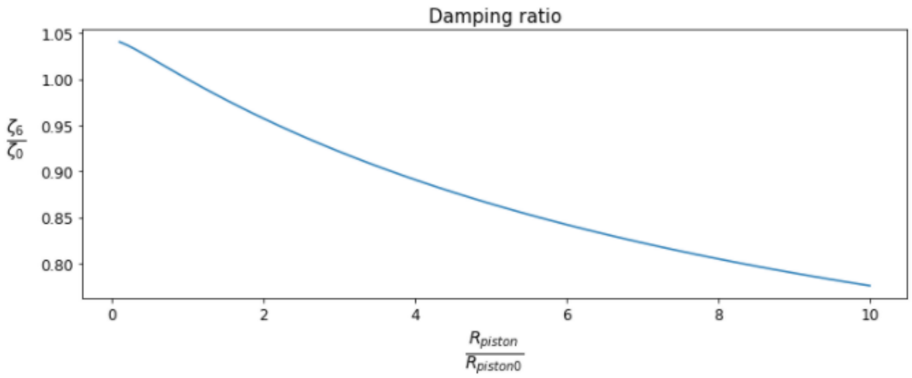


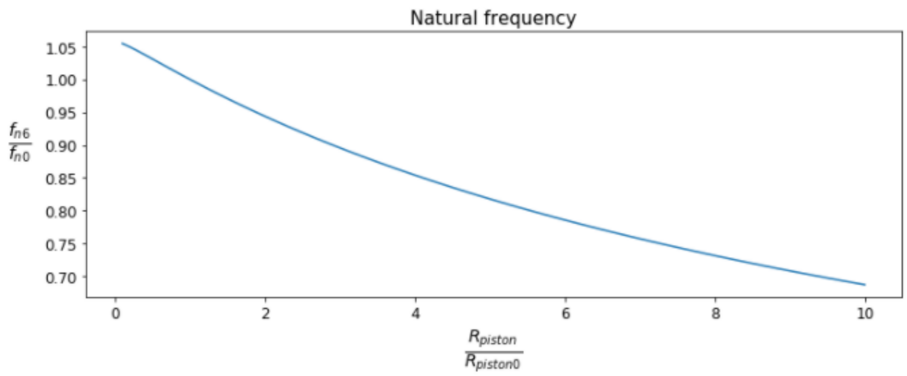
6.2.6 Case 6 - Piston diameter

Increasing the diameter of the piston doesn't influence the stiffness, only to a very limited value. A maximum of 0.5% increase is observed. The damping tends to have a larger variability. A linear relationship with a maximum increase of 14%. The natural frequency is declining in its value since the mass is growing faster than the stiffness, leading to a negative gradient for increasing piston radius. The damping ratio is negatively influenced due to the extra mass that is introduced with a thicker piston.



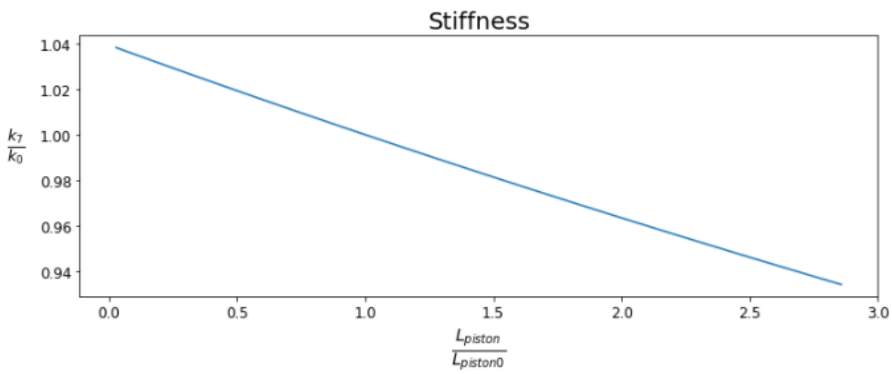


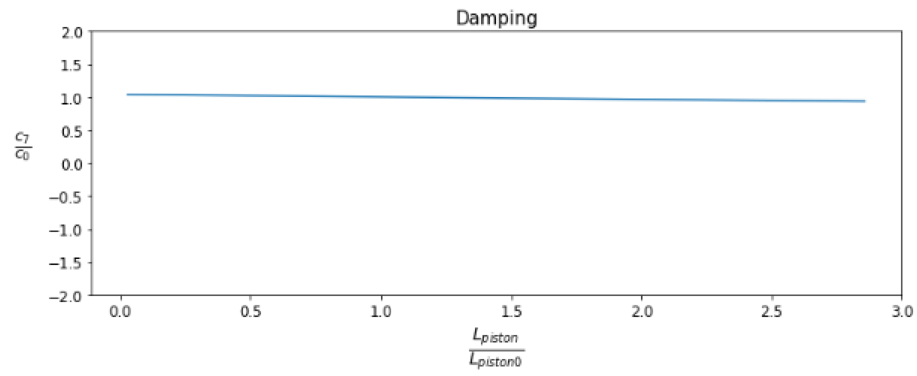


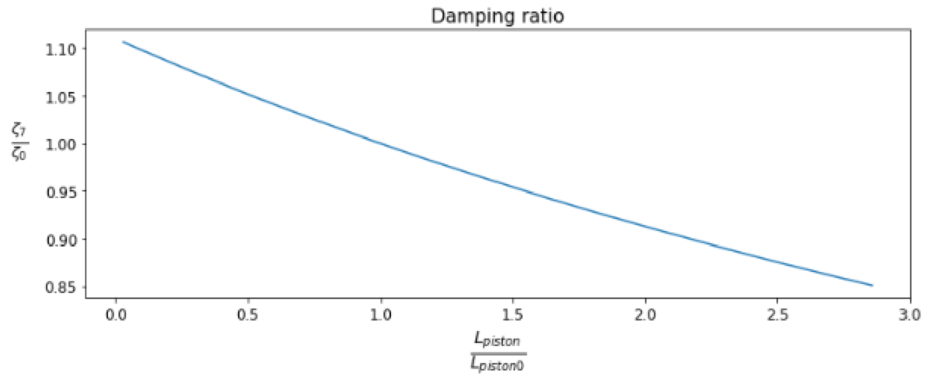


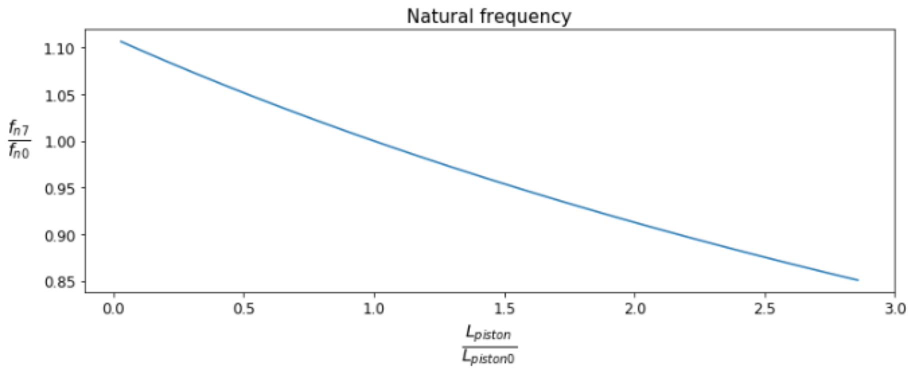
6.2.7 **Case 7 - Piston length**

Taking the piston length as a variable leads to the same effects as for the length of the rod. In general it can be concluded that the length of the axial elements influences the stiffness and damping in a negative sense. Once again a linear trend is observed for the stiffness of the total system. A negative gradient which leads to a minimum value for the stiffness of 94% of the reference value.



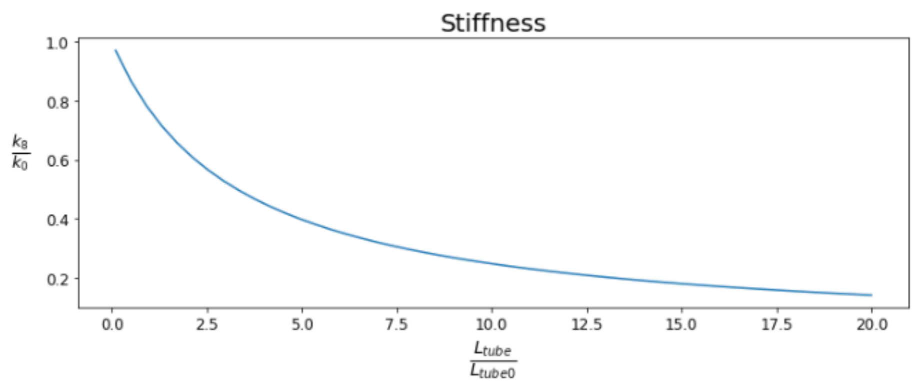


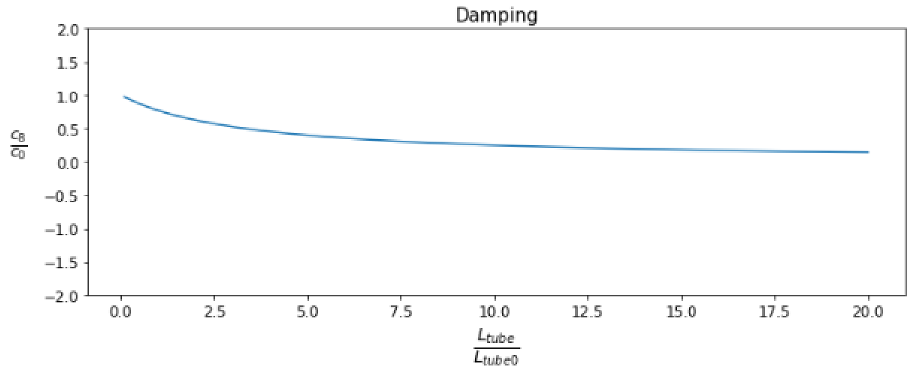


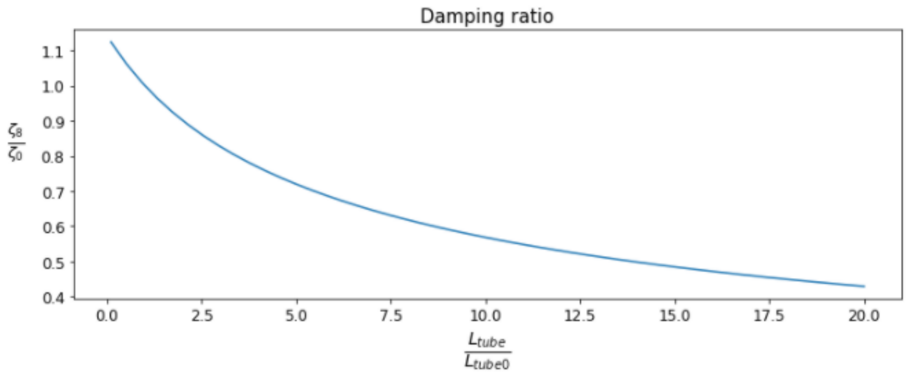


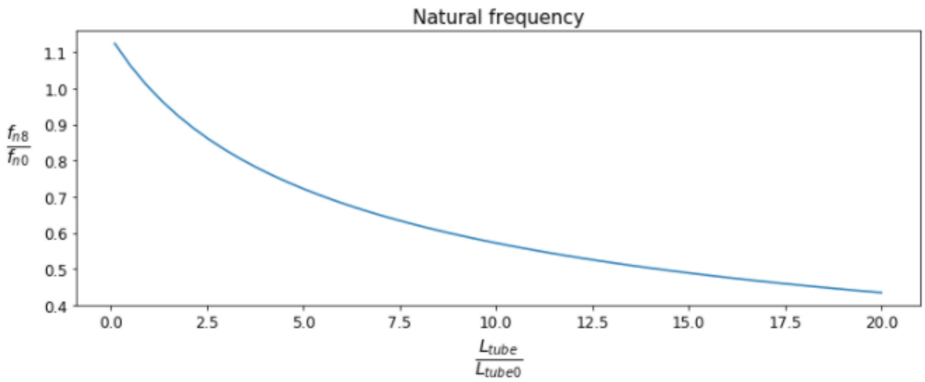
6.2.8 **Case 8 - Tube length**

The stiffness in the tubes is obtained in the same manner as for the fluid in the hydraulic cylinder, by means of the bulk modulus explained in §4.3.1. Just as for the cylinder, an increase in volume leads to a decrease in stiffness. This is also clearly observed in the graphs below. A lot of potential in tweaking the natural frequency is present in the variations in tube length. They do not count for the total mass of the system so the natural frequency solely depends on the stiffness of these tubes, which turns out to be a sensitive relationship.









6.2.9 Sensitivity matrix

This paragraph presents the results from the sensitivity analysis in a qualitative manner. Each case is checked on its performance for each dynamic characteristic of the system, which are known to be:

- Stiffness
- Damping
- Damping ratio
- Natural frequency

The cases are graded using 5 different classifications:

1. '- -' : Very sensitive , decreasing characteristic with increasing geometry
2. '-' : Sensitive , decreasing characteristic with increasing geometry
3. 'o' : No dependency
4. '+' : Sensitive , Increasing characteristic with increasing geometry
5. '++' : Very sensitive , Increasing characteristic with increasing geometry

Table 5 provides an overview of the classifications per case using the grades expressed above. Mind that a ‘- -‘ classification might hold that a smaller elements leads to a larger value for the prescribed characteristic.

Table 5 Results sensitivity analysis

Case	k	c	ζ	ω_n
Cylinder thickness	o	o	o	o
Cylinder radius	++	++	++	++
Cylinder length	-	-	o	o
Rod thickness	+	+	-	--
Rod length	o	o	-	-
Piston diameter	o	o	-	-
Piston length	o	o	-	-
Tube length	--	o	--	--

6.3 Relative importance dynamic characteristics for varying boundary conditions

This paragraph will focus on the variables of the surrounding elements for a valve in a lock gate. Important variables are:

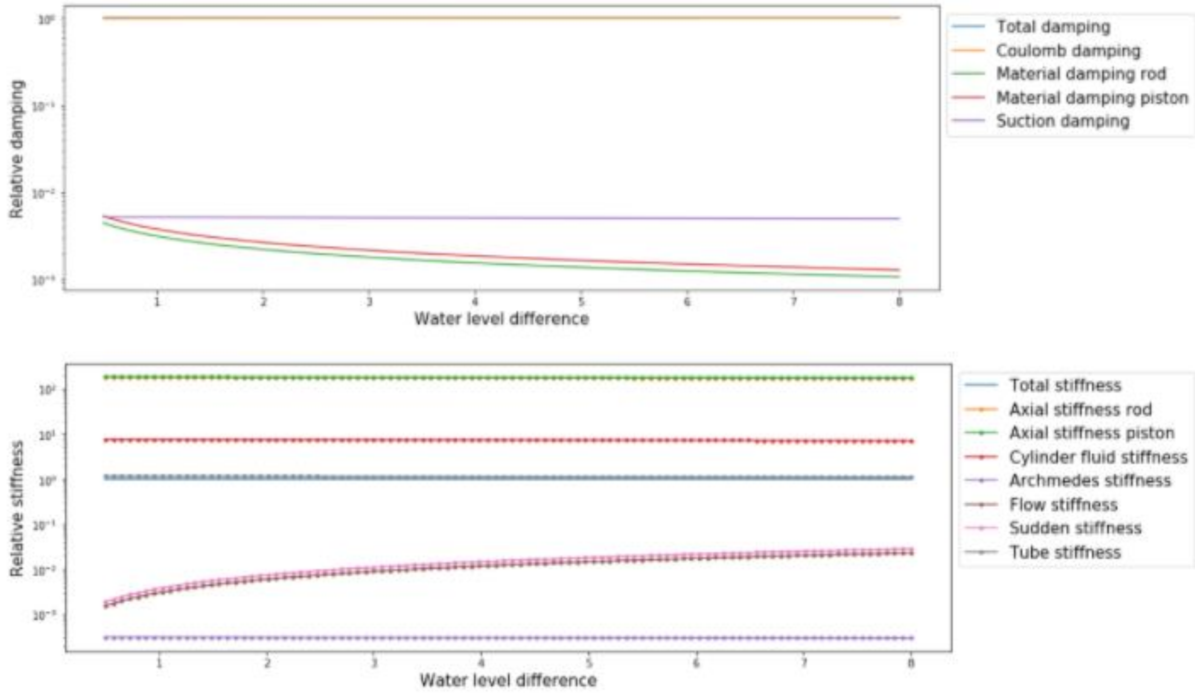
- Water level difference : Δh
- Flow velocity : v_{gate}
- Gate opening : δ
- Pressure amplitude : p_0

The result will give insight in which damping and stiffness components are relevant while designing a vertical sliding valve. The reference situation is the same as elaborated in §6.1 with the mentioned default values for the variables belonging to the valve itself. The graphs that are presented visualize the magnitude of the stiffness/damping compared to the total value of these characteristics.

6.3.1 Varying water level difference

When considering the damping of the system, the coulomb damping of the valve-gate interface is the most dominant for all water level differences. The other damping components are a factor 100 smaller. Furthermore a more or less constant magnitude of the damping components is observed across the water level difference range.

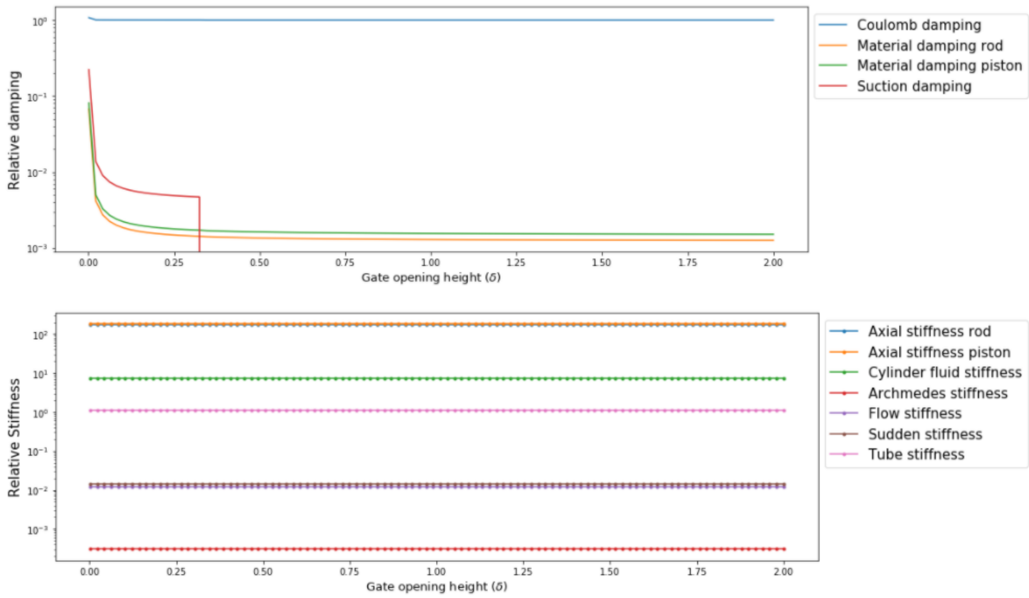
For the stiffness it is shown that the axial stiffness of the piston and rod and the cylinder fluid stiffness are dominant. The fluid stiffness is rather small, which might be because of the relative large length of the tubes. The added water stiffness can be neglected.



Graph 1 Relative magnitudes of damping and stiffness components with varying water level difference

6.3.2 **Varying gate opening height**

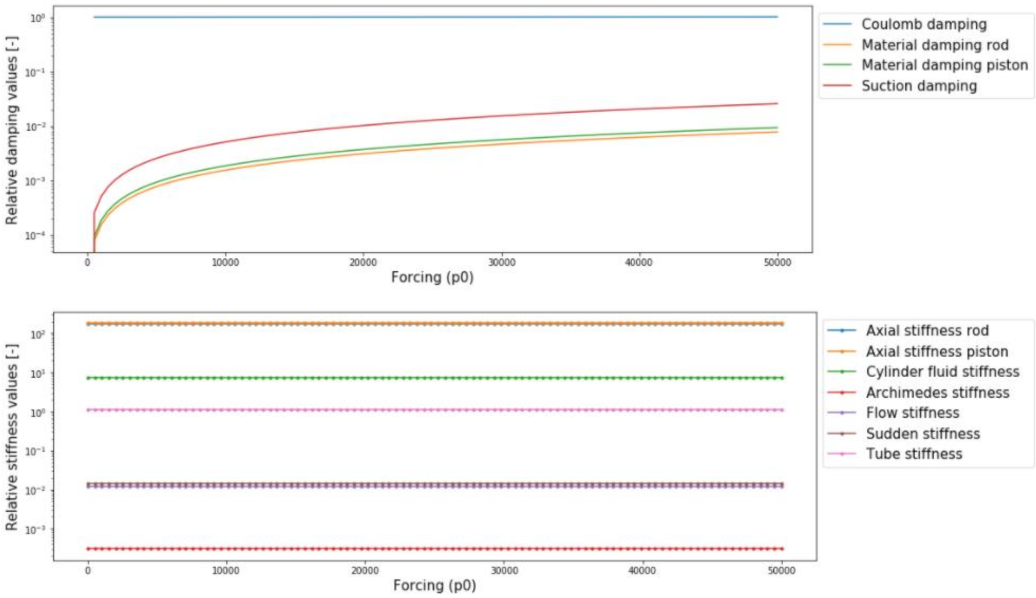
Relative large fractions of the suction and material damping components are seen for small gate openings and they decline with an increasing height. The coulomb damping has a rather steady fraction over the gate heights and is dominant, with a factor 10-100 difference with the other damping components. Stiffness wise the system again experiences large stiffness contributions from the axial and fluid components. The rest may be neglected since they are a fraction 100 - 1000 smaller. The suction damping reduces to zero for a gate opening height larger than 65% of the gate width.



Graph 2 Relative magnitudes of damping and stiffness components with varying gate opening height

6.3.3 Varying pressure

Varying water pressure may occur in a vertical direction underneath the gate. This is due to velocity fluctuation at this location, as described in paragraph 4.8.2. The coulomb damping has a rather steady fraction over the gate heights and is dominant, with a factor 10-100 difference with the other damping components. Stiffness wise the system again experiences large stiffness contributions from the axial and fluid components. The rest may be neglected since they are a fraction 100 - 1000 smaller.

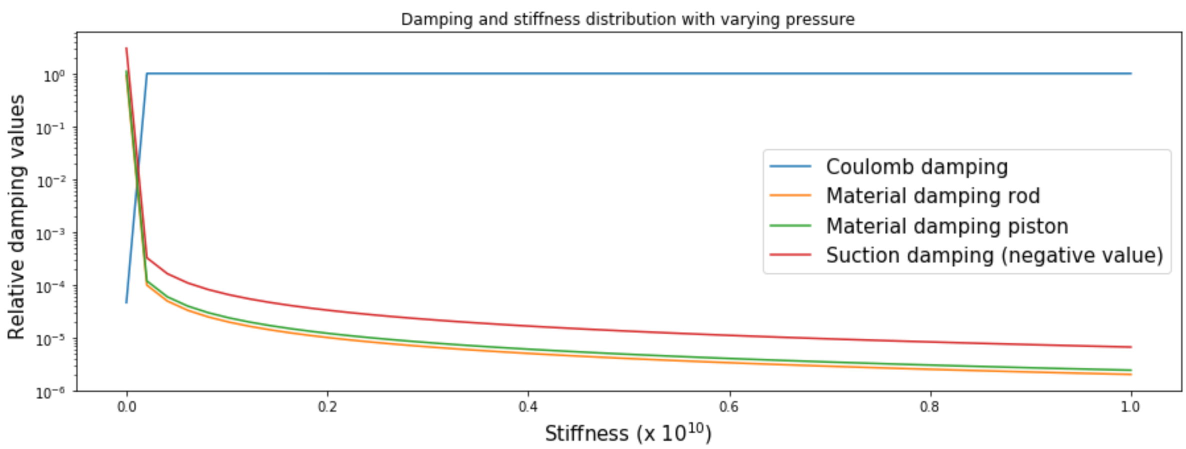


Graph 3 Relative magnitudes of damping and stiffness components with varying pressure amplitude

6.3.4 Varying stiffness

It is observed that the damping due to the friction (coulomb damping) is highly sensitive with respect to the stiffness of the system. This can be explained due to the fact that the coulomb damping depends on the reciprocal of the displacement amplitude. Graph 4 gives an overview of the relative importance of the damping components.

It can be seen that for smaller stiffness the material and suction damping are dominant and the coulomb damping is significantly small. This however switches rapidly for which the coulomb damping becomes dominant.



Graph 4 Relative magnitude of damping components with varying system stiffness

6.4 Tuning dynamic characteristics in design

Based on the design guidelines it might be concluded that the most important requirement is that the natural frequency of the system is sufficiently far from the excitation frequency(Kolkman et al, 1996). This paragraph will therefore focus on how to adjust the natural frequency of the system most efficiently.

The natural frequency of the system is determined by both the mass and the stiffness. However, the mass and stiffness are linked to each other via the dimensions of the different elements. The goal is to search for optimizations in terms of material use and a maximum natural frequency increase.

6.4.1 **Stiffness**

The results from the sensitivity analysis show that a limited number of components are sensitive for an increasing stiffness. These are known to be:

- Increasing cylinder thickness
- Decreasing cylinder length
- Increasing rod thickness
- Decreasing tube length

The values for all variables vary from 10% to 1000% of the reference value. The relative increase in stiffness is given in Figure 29.

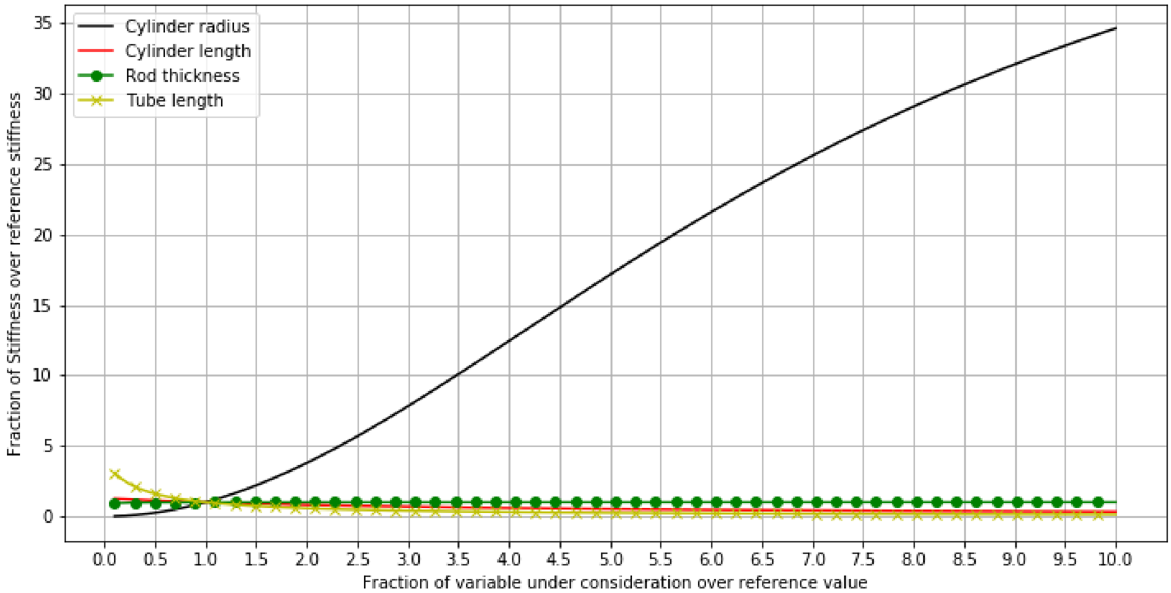


Figure 29 Relative increase in stiffness with increasing variable

On first sight it might instantaneously concluded that increasing the cylinder radius leads to an extreme stiff system. However, the cylinder radius will achieve unrealistic values. Nevertheless, in limits of realistic diameters the stiffness is increasing already by a factor 4-5.

The stiffness of the system also varies with the tube length. A ultimate increase of 4 times the reference value is obtained by minimizing the tube length.

6.4.2 **Mass**

To gain insight in a most optimal solution for tweaking the natural frequency, the mass has to be taken into account as well. Goal is to increase the stiffness and natural frequency with a minimum increase of the mass. Figure 30 shows that an increasing cylinder radius only adds limited mass to the system. Decreasing the cylinder length reduces the mass, as expected. This reduction might induce an increase in natural frequency as well, this is elaborated in §4.5.1.

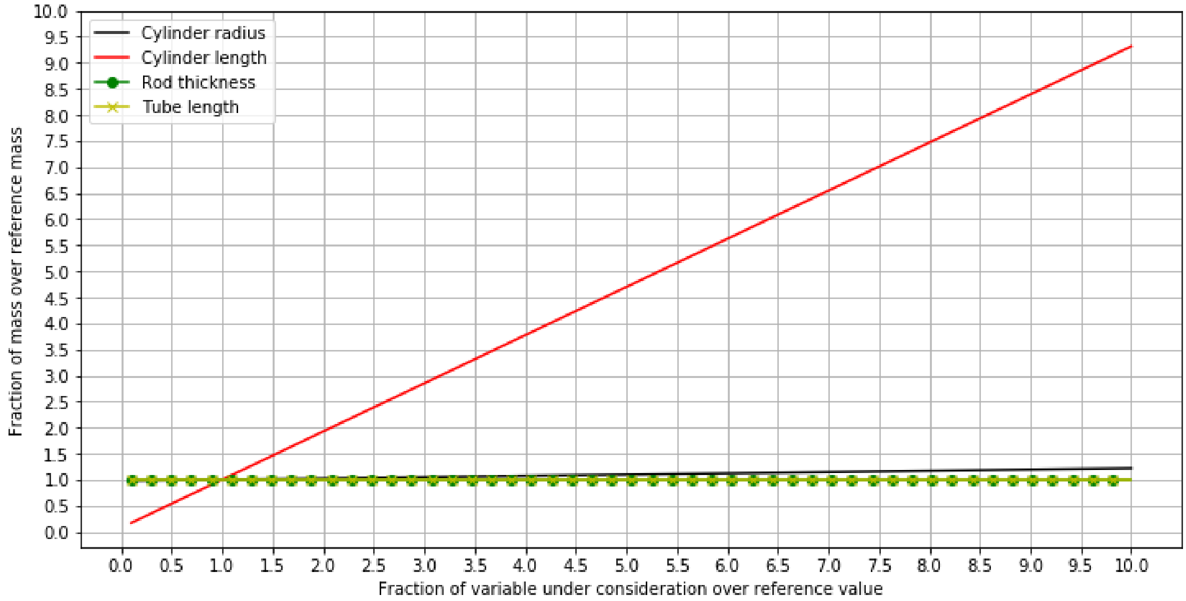


Figure 30 Relative increase in mass for the different tuning components

6.4.3 **Natural frequency**

In case of the stiffness and mass values, we may conclude that reducing the tube length and increasing the cylinder radius might give rise to an optimal increase of the natural frequency. This is shown in Figure 31 as well. Minimizing the tube length leads to an increase of 100% of the natural frequency. Increasing the cylinder radius eventually may lead to 300% for relevant diameters.

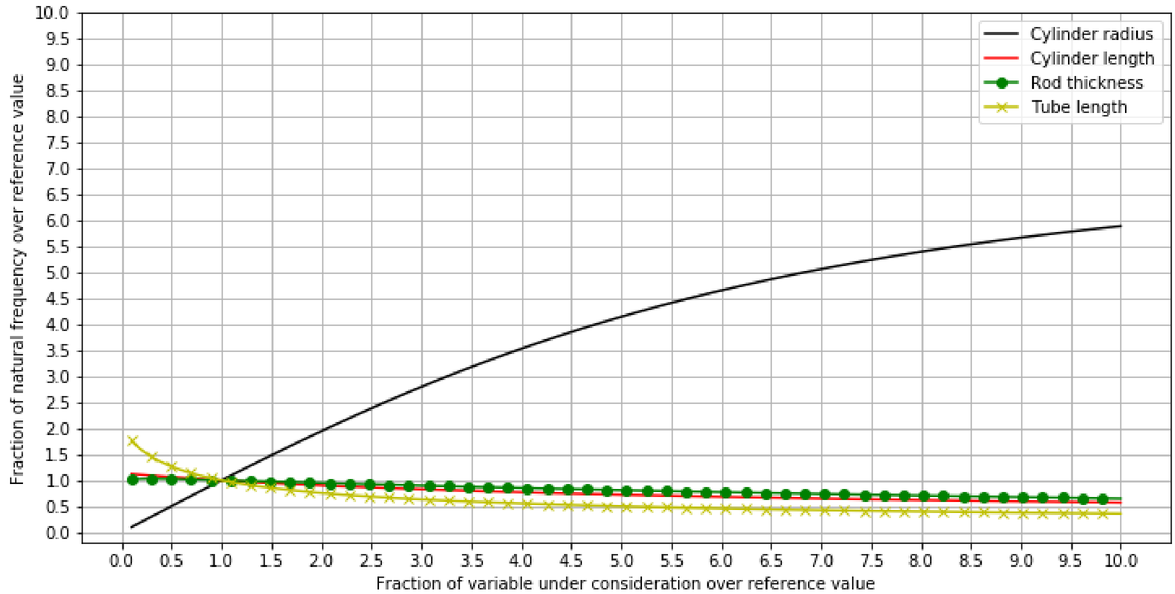


Figure 31 Relative natural frequency of the system for the different tuning components

6.4.4 **Damping**

In case of a wide range of excitation frequencies, it can be impossible to adjust the natural frequency in such a way that it is always greater than 2 or 3 times the largest excitation frequency. This is best seen in the relation of Belfroid (2017) and the Strouhal number(§4.6.3

& §4.7.1). They both state that the pressure fluctuation and excitation frequency are a function of the velocity. Holding on to a critical range of Strouhal numbers from 0.2 to 0.3, we find the following relations for the pressure and excitation frequency.

$$f = \frac{S\delta}{V} = \frac{0.2\delta}{\sqrt{2g\Delta h}} \tag{64}$$

$$p = \frac{0.01}{2}\rho V = \frac{0.01}{2}\rho\sqrt{2g\Delta h} \tag{65}$$

Using a critical Strouhal number of 0.3, a gate opening height can be determined via Kolkman et al(1996) equation 49. Together with the default values given in Table 4 we find:

$$S = 0.5 * \frac{\delta}{b} + 0.05 \rightarrow \delta = 2 * (S - 0.05) * b = 0.25\ m \tag{66}$$

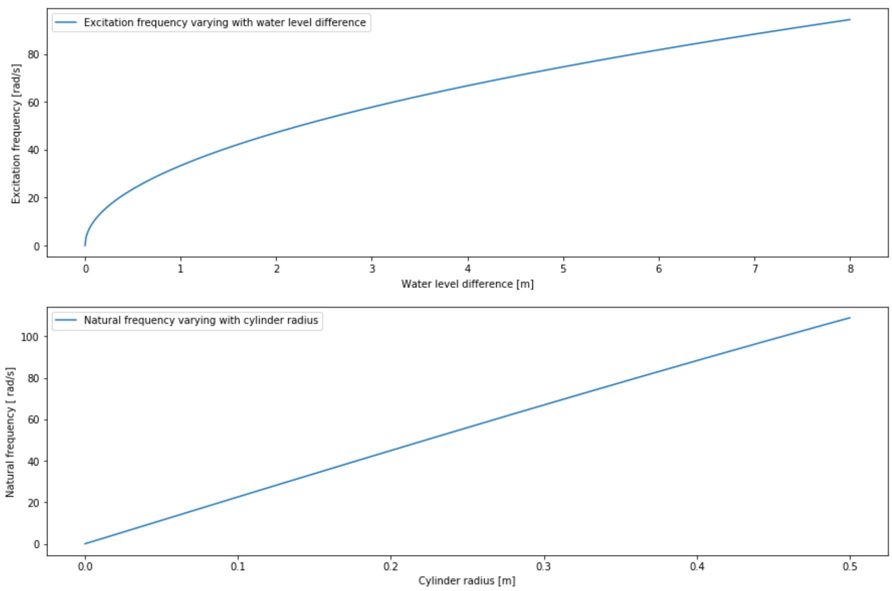
For a varying gate height the excitation frequency can be determined together with the pressures acting underneath the gate.

From the sensitivity analysis it was observed that the following components where most sensitive for the damping of the system:

- Cylinder radius
- Cylinder length
- Rod thickness
- Tube length

Cylinder radius

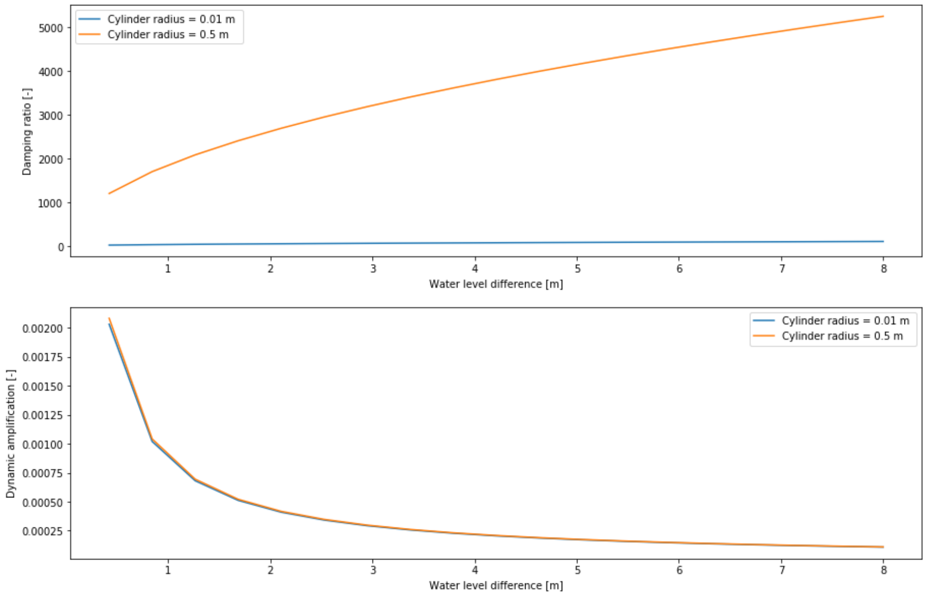
The optimal solution in terms of natural frequency is used for the damping problem, which was increasing the radius of the cylinder. Graph 5 gives an overview of excitation frequencies and natural frequencies of the system for different values of the cylinder radius. The hydraulic cylinder radius is limited to 0.5 meters which is already an extreme value.



Graph 5 Overview of excitation frequencies and natural frequencies for varying cylinder radius

The graphs show a excitation frequency that will always be the same at some point during the filling and emptying process, no matter what cylinder radius is used. Interesting is to observe the damping, damping ratio and the dynamic amplification factor for the different cylinder radius dimensions.

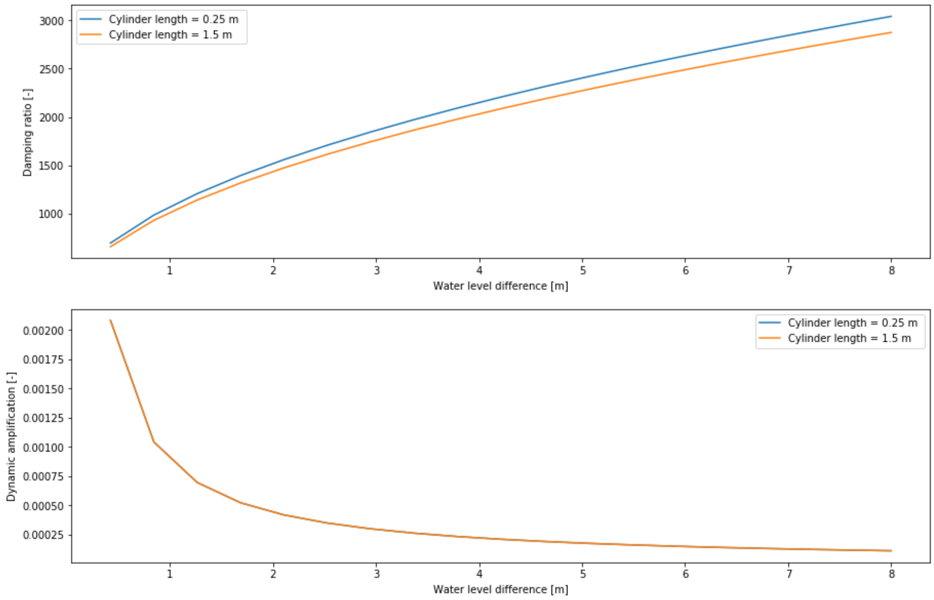
In case of resonance one might expect extensive vibration patters with potential large amplitudes. Nevertheless this can be counteracted by adding sufficient damping to the system. For different water levels the damping ratio and dynamic amplification factor are plotted in graph 6. It can be observed that in case of large water level difference the damping is large, whereas the damping is relatively small for smaller water level differences. This leads to a situation where dynamics can play an important role at the end of the empty and filling process, when water levels are small compared to each other. Mind that in all cases the damping ratio is sufficiently high (extreme) to reduce all excitation amplitudes up to the bare minimum.



Graph 6 Damping ratio and dynamic amplification as function of cylinder radius, for different water levels

Cylinder length

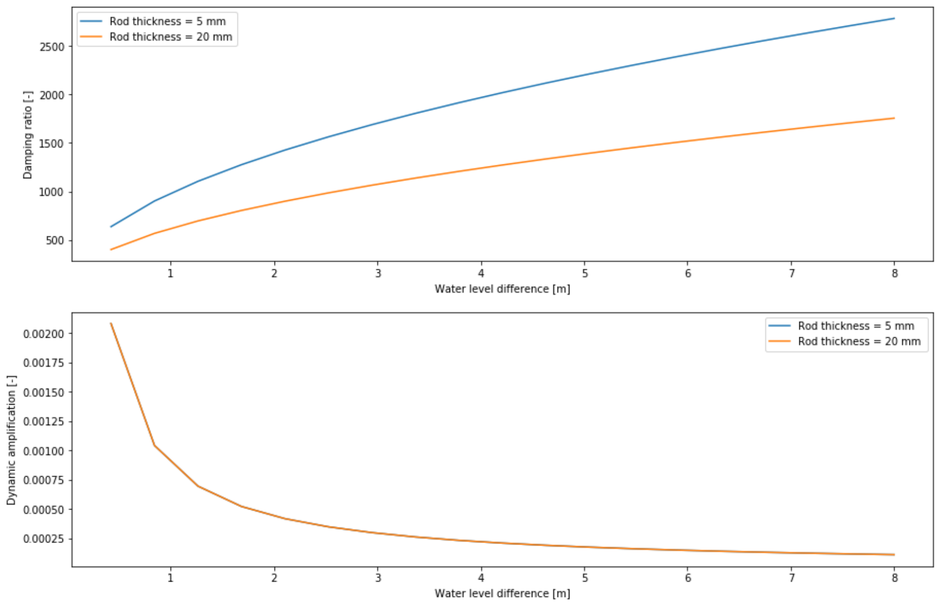
The cylinder length shows a more or less independent relation with the damping ratio. If we take a look at the dynamic amplification we find two exactly the same values for the different cylinder lengths. It can be concluded that varying the length of the hydraulic cylinder doesn’t really influence the damping features of the system.



Graph 7 Damping ratio and dynamic amplification as function of cylinder length, for different water levels

Rod thickness

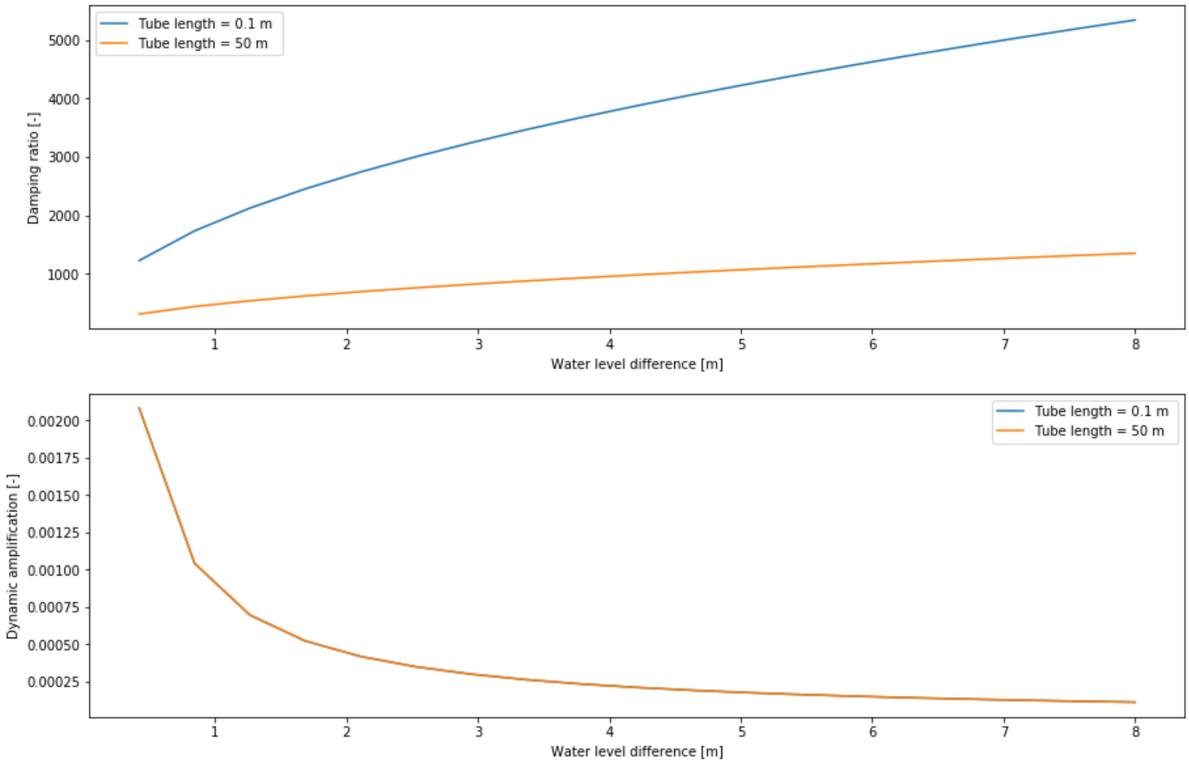
The damping ratio is increased by decreasing the thickness of the rod. This seems somewhat odd compared to the other results, but can be explained by a decrease in stiffness and mass. The dynamic amplification is again not influenced.



Graph 8 Damping ratio and dynamic amplification as function of rod thickness, for different water levels

Tube length

As expected based on the previous analysis regarding the tube length, it shows an increase for the damping ratio by a decreasing tube length. The magnitude is somewhat the same as the case with the cylinder radius, however this feature can be implemented after construction as well which makes it a very powerful measure. The dynamic amplification is again not influenced in this case study. In case of smaller damping ratio values a more extensive influence in the dynamic amplification can be expected.



Graph 9 Damping ratio and dynamic amplification for varying tube length

7 Discussion

The research that was done was subject to multiple assumptions. These assumptions are discussed regarding their potential influence on the outcome of the research model.

Single case for a vertical sliding valve examined

Although it might be expected that the outcomes of the models have broad support across different configurations of a vertical sliding valve with a hydraulic cylinder, it cannot be said with full assurance. The system investigated had relatively large dimensions with respect to smaller locks and their filling and emptying system. In case of scaling down the results will hold, but in other cases the results may turn out differently.

The basic conclusions in terms of increasing stiffness and damping potentially hold for all configurations. The natural frequency, damping ratio and dynamic amplification depend on the relative magnitudes of the stiffness, damping and mass as well. Therefore these characteristics might lead to other conclusions for different configurations. It should be tested across a broad spectrum of valves whether the conclusions drafted in this thesis are solid for all valves.

Stiffness in cylinder and tube from bulk modulus of fluid

The stiffness induced by the hydraulic cylinder itself, is assumed to come from a decrease in volume of the fluid in the cylinder when pressurized. Important assumption is that the cylinder will not deform radially under increasing pressures. If this deformation occurs, the relation for the bulk modulus stiffness doesn't hold completely anymore. Another stiffness is then introduced, which has the same basis as the axial stiffness of the piston and rod. The radial deformation might also lead to an extra damping component in the system due to material damping. Nevertheless, the dependencies of the different components with respect to the dynamic characteristics will hold their shape as derived in this thesis. The magnitude however, might differ when radial deformation is introduced.

Strouhal number

A Strouhal number of 0.2 for wide 2D objects subjected to a flow is assumed. This was based on Belfroid(2017). The validity of this assumptions must be certified across multiple studies and measurements in real life structures. The Strouhal number however doesn't influence the stiffness and corresponding natural frequency of the system itself.

Axial expansion tubes

For this study it is assumed that the tubes of the hydraulic cylinder do not undergo a axial expansion. This leads to an assumption that there is no stiffness component that relates to this deformation direction.

Fluid reservoirs for cylinder

In this study the reservoirs for the fluid that is in the closed system of the hydraulic cylinder are not taken into account. It is assumed that they play no role in terms of stiffness and damping due to their stiffer composition compared to the tubes. Furthermore it is assumed that they can be closed so that they are disconnected from the tubes and are not part of the total system.

8 Conclusion

The conclusion of the research is based on answering the main and sub research questions that where drafted in the introduction. All conclusions are based on the case study of a vertical sliding valve which is performed during this thesis.

First of all it was examines which components of a hydraulic cylinder have the most influence on the dynamic characteristics of the total system. These characteristics are the stiffness and damping terms as well as the natural frequency of the system. A sensitivity analysis performed in §6 forms the basis of the answer to this question. Several conclusions can be drafted from this paragraph.

1. Increasing the diameter of the hydraulic cylinder leads to an increase in stiffness and natural frequency. This relation for the stiffness is not linear but has a more or less exponential increase of the stiffness with increasing diameter. The relation the natural frequency is more or less linear for an increasing diameter.
2. Increasing of the cylinder length leads to a reduction of the stiffness, damping and natural frequency. On the other hand, a decrease in cylinder length leads to an increase of those parameters. This relation is best described as exponential with a rapidly increasing value for the dynamic characteristics for a decreasing cylinder length. Important side note is that most of the time the cylinder has a prescribed length, due to the minimal opening height of the valves.
3. A decreasing rod thickness leads to a potential disastrous decline of the damping ratio and natural frequency of the system. It turns out that this value is very sensitive for deviations and should therefore be kept in mind all the time.
4. Minimizing the tube length in which the fluid is transported from a pressurized chamber towards the cylinder, leads to a relative easily obtained increase in stiffness and natural frequency. The extra effort that has to be put into the structure is minimal since a minimum of extra components are needed to obtain a minimal tube length.

Besides these four other components where tested as well, but turned out to have limited influence in the stiffness, damping and natural frequency of the system. To conclude the following dependencies are drafted.

$k \sim R^2_{cylinder}$	$c \sim R^2_{cylinder}$	$\omega_n \sim R_{cylinder}$
$k \sim L^{-1}_{cylinder}$	$c \sim L^{-1}_{cylinder}$	$\omega_n \sim L^{-0.5}_{cylinder}$
$k \sim \arctan(t_{rod})^{0.5}$	$c \sim \arctan(t_{rod})^{0.5}$	$\omega_n \sim -0.1 \text{ x } t_{rod}$
$k \sim L^{-0.3}_{tube}$	$c \sim L^{-0.3}_{tube}$	$\omega_n \sim L^{-0.3}_{tube}$

Secondly the damping and stiffness terms that play the most important roles for a vertical sliding valve where investigated. The goal was to focus on the elements that can be excluded on forehand under certain circumstances, like: gate opening height, water level difference, stiffness of the system and pressure variation. For all four situations graphs were presented where the magnitude of the damping and stiffness components, relative to the total value, where plotted for a varying variable representing one of the four circumstances. The overall conclusion, that was observed for all situations, is that the stiffness is mostly determined by the axial stiffness of the rod and piston together with the fluid stiffness in the cylinder. Other component where at least a factor 10 smaller. Important note here is that in case of a small stiffness coming from the fluid tubes, it might lead to an extreme flexible system. It is therefore important to increase the stiffness of the tubes in such a way that it increases the lower limit of the system stiffness. Concluded on this research it seems that for larger structures the FSI doesn't play a role in the stiffness of the system.

In case of damping it was found that in all cases coulomb damping from the friction between the valve and guiding rails was dominant. Even for small friction coefficients. The coulomb damping is on the vibrations of the valve itself, which on its turn depends on the stiffness of the system. It turns out that the coulomb damping becomes dominant for already relative small values for the stiffness. Therefore it can be concluded that suction damping plays a minor role at all times, so self-excitation is not likely to happen. Even when the coulomb damping is small compared to the suction damping, the material damping provides enough buffer capacity to overcome self-excited vibrations.

Thirdly it was examined whether the hydraulic cylinder could influence the natural frequency of the system significantly. Several components from the hydraulic cylinder can be used to adjust the natural frequency. Looking at the sensitive elements of the cylinder it was found that four components where potential candidates to influence the natural frequency. A stiffness analysis and the corresponding natural frequencies for a range of variables was performed and shown in the graphs of the different cases of the sensitivity analysis. It turned out that the cylinder radius had a more or less linear relation with the natural frequency. The tube length was found to be more or less exponential for smaller values than the reference value. The length of the cylinder and rod thickness were also found to be sensitive, but turned out to be less sensitive than the already mentioned components. The overall conclusion can be drawn that for an effective increase of the natural frequency, one has to increase the cylinder radius. Decreasing the tube length is a more elegant way of increasing the natural frequency, but might lead not to the desired value due to the limited influence.

Looking at the two variables that came from the previous question, hydraulic cylinder and tube length, it is clear reducing the tube length is the most effective in terms of mass reduction. Using this option, it might be clear what the consequences are on beforehand. In principle only one single dynamic characteristic is adjusted, the stiffness. The mass of the system is not influenced, and therefore one might immediately observe the consequence of this measure. The hydraulic cylinder also influences the mass of the system and therefore it might be unclear what the effect is on beforehand. For the situation investigated in this thesis it was clear that the cylinder radius was the most effective in terms of both mass and stiffness, and consequently the natural frequency.

In contrast to the stiffness and natural frequency, the fifth question looked at the damping and how this could be influenced by adjusting the hydraulic cylinder. The damping ratio and amplification factor are influenced by multiple factors and not solely by the damping, natural frequency and forcing frequency. As elaborated in §4.5.1 and §0 the mass and stiffness of the system play a crucial role in these characteristics as well. Using the Strouhal number and pressure fluctuations based on the approximation of Belfroid (2017) it was found that it is not possible to prevent resonance to occur. To reduce the excitations it is therefore crucial to increase the damping ratio to a maximum. The hydraulic cylinder and the tube length showed the largest increase in the damping ratio over the different values for the water level difference. An increase in the cylinder radius(up to 0.5 meters) and a decrease in the tube length(up to 0.1 meter) resulted in an increase of the damping ratio by a factor 2 – 5, depending on the water levels. A decreasing tube length can be obtained by installing valves in the tube close to the valve of the gate. Since the damping ratios rapidly grew to enormous values, the effect on the dynamic amplification was not seen. All variables showed a maximum dynamic amplification, as described in equation 41, in the order of 0.002. This may also be an indication that the stiffness and damping of the system in the reference situation was already enough to counteract the excitations and limit them to a minimum. The conclusion therefore might be that in terms of dynamic amplification, the system is not subjected to large vibration patterns across all natural frequencies and forcing frequencies. Therefore it plays a minor role in the design process for large hydraulic structures.

The answers from the sub question give clear insight in the behaviour of a hydraulic cylinder in terms of the dynamic characteristics of the total system. They serve as a basis for answering the main research question:

- *What role does a hydraulic cylinder play in combination with a vertical sliding valve in terms of stiffness, damping and the corresponding dynamic characteristics?*

It was shown that a hydraulic cylinder and its different components do influence the dynamic characteristics of the system significantly. The **cylinder radius** and the **tube length** showed to be the most effective components in terms of:

- Increasing the stiffness
- Increasing the natural frequency
- Increasing the damping ratio of the system

Furthermore it was observed that large hydraulic structures vibrating in the vertical plane due to an horizontal flow, might not be prone to large vibration amplitudes. Nevertheless this conclusion should be validated by means of empirical data. In all case the dynamic amplification was negligible. Furthermore it was observed that the damping of the system never reached a negative value, which excludes the fact that self-excitation might occur. Mainly the relative large components of the cylinder that where contributing to the material damping, as well as the coulomb damping from the interface where sufficiently high to counteract the suction damping.

The stiffness and damping terms resulting from the FSI, excluding the suction damping, where negligible compared to the structural dynamic characteristics of the system. This is powerful information regarding the design of large structures. It shows that, at least for the vertical sliding valve, damping an stiffness of the system itself is dominant. This reduces the uncertainty in the calculations since the FSI components of the damping and stiffness values are mostly empirical.

9 Recommendations

During the study some assumptions where made regarding the damping and stiffness components as well as the geometry of the system.

First of all it was assumed that the oil tubes leading to the cylinder where not experiencing to any radial expansion of the material. This implies that any damping and stiffness from the expanding material was not taken into account. To compensate for this it was assumed that the oil inside the pipes was compressed under high pressure. Using the bulk modulus a stiffness term was found which was used for the stiffness of the tubes. This procedure is also applied on the cylinder itself. Nevertheless this procedure should be validated. It may well be that the stiffness term from the radial expansion of the tubes differs significantly from the one obtained via the bulk modulus. In case of a dramatic decrease of the stiffness value the system might encounter a significant decline in stiffness.

Secondly the axial deformations of the oil tubes where neglected. It may well be the case that due to pressure at one side of the tube, the tube itself expands axially. For now the axial deformation was set on 0 and therefore didn't participate in the dynamics of the system. In case of axial deformation, extra stiffness terms come into play and might influence the stiffness in a negative way.

Furthermore assumptions regarding the overall geometry of the cylinder where made. It was assumed that the fluid reservoir at one side of the oil tubes where not participating in the dynamics. It should be verified whether this is true.

This sections is finalized by mentioning the fact that this additional thesis made use of programming models that should be validated by means of empirical data. Therefore it is recommended to perform flume tests or observe data from real life structures that can help validate this research.

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Appendices

Appendix A – Hydraulic gates and valves

Rolling gate

Rolling gates are widely applied in navigation locks worldwide. Famous examples are the new sealock in IJmuiden and the lock in the Panama Canal. The undergo a horizontal translation over a rails which is installed on the sill. In closed position the system is locked into the lock heads.

Figure 32 New Sealocks at IJmuiden (Dyckerhoff Basal, n.d.)

Mitre gates

These types of gates are, just like the rolling gates, widely applied in navigation locks. They ensure a tight sealing of water at one lock end due to the way of installation of the gates. They are also popular since they don't induce height limitations for ships to pass and because of their relative simple installation and operation. In closed position the gate has a hinged support in the lock head and a more fixed, but still more hinged like, support in the middle of the lock where it meets the other gate.

Segment gate

This gate has a curved shape and is also known as a radial gate. It is not often applied in navigation locks due to the space it needs to open and close. An advantage is that one can easily fill and empty the lock by means of lifting the gate to create a flow underneath the structure. Segment gates are often applied in flood defence structures like the Haringvliet sluices which are depicted in Figure 33. The gate is can undergo a vertical rotation and is supported by steel bars or trusses that are connected to a pivot point around which the gate rotates.

Figure 33 The Haringvliet sluice (Rijkswaterstaat , z.d.)

Appendix B - Driving mechanisms

Towing cables or chain

These kind of driving mechanism are widely applied in older structures and mostly in vertical translation doors like a vertical lifting gate and a vertical sliding gate.

Workflow

The towing cable of chain is installed on the top of the door and is connected to a counter weight on the other end. The cable flows over a lifting tower that is installed on both lock heads next to the gate. Due to the presence of the counter weight the absolute mass of the gate or valve is reduced which makes it less labour intensive to lift it up. The vertical translation of the counter weight may either be done by means of a hydraulic cylinder or a gearing system.

A cable or chain does contribute to the vertical stiffness of a driving mechanism, in both the stiffness and damping coefficients. Extension of the cables may be modelled as a spring with a certain stiffness depending on the Young's Modulus of the material.

Installation in structures

The towing cables are often seen in vertical lifting gate. As mentioned, the cables are installed over a lifting tower and connected to the top of the gate at one end and a counter weight on the other end. Besides the presence in lifting gates, they are also present in older rolling gates. A chain is connected to the bottom of the gate at one end, and to a gearing system in the lock head at the other end. The gearing system is driven by a mechanical system which may run on electricity which enables the gate to translate in horizontal direction. Figure 34 gives an overview of both gates and the installation of the driving mechanism.

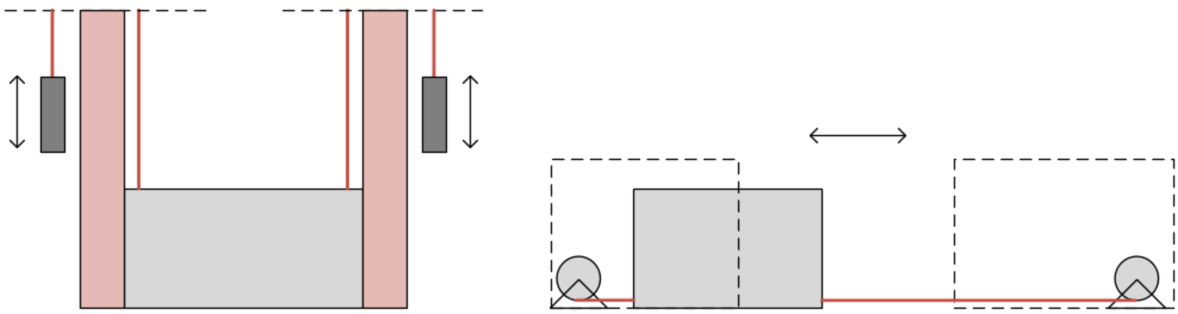

Figure 34 Installation of towing cable or chain on vertical lifting gate (left) and rolling gate(right)

Gear rack

A gear rack is commonly applied in older rolling types of gates that translate in the horizontal plane.

Workflow

The idea behind this driving mechanisms is that it a gearing wheel, which is driven by a motor, starts rotating and shifts the door in the horizontal plane due to the gear rack that is installed on the door. The gears on the wheel fall exactly together which enables them to translate the door in the horizontal plane.

One may observe that this systems' stiffness depends on the gearing wheel itself and the type of connection between the wheel and the gate. Among other the weight of the wheel and the degree of clamping of the wheel into the surrounding structure may contribute to the stiffness of the driving mechanism. A higher degree of clamping leads to more friction, which on its turn leads to a higher stiffness of the system.

Installation on structures

Often the gear rack is applied in rolling gates. A gearing wheel is installed beneath the gate into a lock head. On the gate itself a gear rack is installed from which the gears are matched to that of the gearing wheel. Rotating the wheel then drives the gate in horizontal direction.

Panama wheel

The panama wheel is not used in current structures since the amount of space that it needs. Therefore it will not be treated in this report.

Appendix C – Vibration patterns of different gates

Vertical lifting gate

Vertical lifting gates are often subject to strong flow patterns through the gate in case of filling and emptying locks. Besides that waves impacts are also found in especially coastal areas where they are part of a flood defence system. Vertical lifting gates may be designed for extreme storm conditions, for example in case of the Eastern Scheldt barrier. The below list gives an overview of the possible excitation sources that may occur together with the expected translation direction of the gate.

Excitation source	Excitation direction
Sudden force due to flow Turbulent flow Flow instability Amplification due to fluid resonance Self-excitation (in case of smaller gate)	Horizontal plane Vertical plane
Wave slamming Wave induced pressure variations	Horizontal Vertical (in case of smaller gate)

Rolling gate

Rolling gates are rarely subjected to flow induced excitations. They open and close only during situations in which water levels on both sides of the gate are equal to each other. Although they are not experienced often, they may happen in case of a filling and emptying system through the gate in navigation locks. Especially for large sized locks this may induce a problem since the discharges through a gate can be extreme. In this thesis however it is assumed that the masses of the gates are such that they do not impose any dynamical problems in case of strong flows.

The wave sources however are of another kind for these gates. In case of, especially, structure that are part of coastal defence systems, design waves heights and impacts can be large. In this sense the gate may experience wave slamming which may impose horizontal excitations.

Excitation source	Excitation direction
Amplification due to fluid resonance Wave slamming	Horizontal plane

Mitre gate

Mitre gates are installed in locks and have a hinged support at one end and a fixed support where the gates meet each other. In case of a physical connection between the gates, they are hindered to excite in a radial direction in the horizontal plane. However, they might undergo vibrations in the horizontal plane anyways.

Depending filling and emptying system this type of gate may experience flow induced excitations. Again, depending on the gate geometry, it may experience unstable flow.

Furthermore, it can be neglected that severe wave conditions occur, since these types of gates are often installed in river locks.

Excitation source	Excitation direction
Sudden force due to flow	Horizontal plane
Turbulent flow	Radial plane
Flow instability	
Amplification due to fluid resonance	

Segment gate

Segment gates are applied in both locks, water control and water defence structures. Independent of the situation for which the gates are implemented, they always experience flow induced vibration sources. Often gates experience a strong flow underneath the gate when they are lifted and openings arise.

Gates that are located near coastal zone may also experience severe wave conditions, but the number of these structures in direct contact with the sea is limited. One of the flood defence system that is located close to sea are the Haringvliet sluices. At this sluices the wave impact is considerably less compared to the flood defence system located at the coast, therefore they will not induce the same dynamic risks.

Excitation source	Excitation direction
Sudden force due to flow	Horizontal plane
Turbulent flow	Vertical plane
Flow instability	
Amplification due to fluid resonance	
Self-excitation	
Wave slamming (not often)	Horizontal
Wave induced pressure variations	Vertical

Appendix D – Python script equations and variables

As highlighted in chapter 5, the python script that is used can be split up into several parts. This appendix will focus on the first two parts which are the defined variables and the formulas. The sensitivity analysis and varying boundary conditions contain multiple variable compositions which will all be highlighted in this appendix.

Variables sensitivity analysis

The approach for the sensitivity analysis is based on taking one variables as a range of values while the others are kept constant. This leads to an overview of different variables and their influence in the dynamic characteristics of the system.

Constant values

```
# Material variables
E_steel     = 210     # GPa     ,  Youngs modulus of Steel
E_alu        = 69      # GPa     ,  Youngs modulus of Aluminium
B_oil        = 1.5     # GPa     ,  Bulk modulus of oil in cylinder
B_water      = 2.1     # GPa     ,  Bulk modulus of water in cylinder
mu_gate      = 0.1     # [-]     ,  Friction coefficient gate - concrete
mu_cylinder = 0       # [-]     ,  Friction coefficient piston - cylinder
eta_steel    = 0.004   # [-]     ,  Damping ratio steel (material damping)
eta_alu      = 0.018   # [-]     ,  Damping ratio aluminium (material damping)
Cw           = 1.2     # [-]     ,  Drag force coefficient for vertical gate in flowing water
rho          = 1000    # [kg/m3], Density of fresh water
rhosteel     = 8050    # [kg/m3], Density of steel
rhoalu       = 2700    # [kg/m3], Density of aluminium


# Variables of valve -  Constants
delta      = 0.2          # meters  , Opening height of valve
b          = 0.2          # meters  , Thickness of the valve
blow       = 0.5          # meters  , Thickness of lower edge valve
W          = 2            # meters  , Width of the valve
h          = 1.5          # meters  , Height of the valve
h1         = 4            # meters  , Waterdepth upstream
h2         = 0            # meters  , Waterdepth downstream
dH         = h1 - h2      # meters  , Water level difference over valve
alpha      = 45           # degrees , Inclination angle lower edge valve
mu         = 1            #[-]      , Contraction coefficient of flow underneath valve
p0         = 100000
```

Case 0 – reference case

```
# Geometry variables of driving mechanisms

delta       = 0.25                # meters  , Opening height of valve
Ltube        = 20                # meters     , length of fluid tubes
Lcylup       = 3.3 - delta       # meters     , Length of cylinder compartment on top of piston
Lcyldown     = 3.3 - Lcylup      # meters     , Lenght of cylinder compartment beneath piston
Lpiston      = 3.5               # meters     , Length of piston including rod
Rpiston      = 0.1               # meters     , Radius of piston
tpiston      = 0.05              # meters     , thickness of piston
Lrod         = 3                 # meters     , Length of connecting rod from piston to top of valve
Rrod         = 0.05              # meters     , radius of connecting rod
trod         = 0.02              # meters     , thickness of rod edge
Rcyl         = 0.25              # meters     , Inner radius of the hydraulic cylinder
tcyl         = 20                # milimeters , Thickness of the cylinder
Lculvert     = 10         # meters      , Length of the culvert in which valve is installed (gate width)

# Design of driving mechanism

instal_direction            = 'horizontal'
material_piston             = 'steel'
material_rod                = 'steel'
material_cylinder           = 'steel'
```

Case 1 – Hydraulic cylinder thickness

```
# Geometry variables of driving mechanisms - Case 1

delta      = 0.25               # meters   , Opening height of valve
Ltube       = 20                # meters     , length of fluid tubes
Lcylup      = 3.3 - delta       # meters     , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup      # meters     , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5               # meters     , Length of piston including rod
Rpiston     = 0.1               # meters     , Radius of piston
tpiston     = 0.05              # meters     , thickness of piston
Lrod        = 3                 # meters     , Length of connecting rod from piston to top of valve
Rrod        = 0.05              # meters     , radius of connecting rod
trod        = 0.02              # meters     , thickness of rod edge
Rcyl        = 0.25              # meters     , Inner radius of the hydraulic cylinder
tcyl        = np.linspace(5,30,25)      # milimeters , Thickness of the cylinder
Lculvert    = 10        # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 2 – Hydraulic cylinder radius

```
# Geometry variables of driving mechanisms Case 2

delta       = 0.25               # meters   , Opening height of valve
Ltube       = 20                # meters     , length of fluid tubes
Lcylup      = 3.3 - delta       # meters     , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup      # meters     , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5               # meters     , Length of piston including rod
Rpiston     = 0.1               # meters     , Radius of piston
tpiston     = 0.05              # meters     , thickness of piston
Lrod        = 3                 # meters     , Length of connecting rod from piston to top of valve
Rrod        = 0.25              # meters     , radius of connecting rod
trod        = 0.02              # meters     , thickness of rod edge
Rcyl        = np.linspace(0, 1 , 21)        # meters     , Inner radius of the hydraulic cylinder
tcyl        = 20                # milimeters , Thickness of the cylinder
Lculvert    = 10        # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 3 – Hydraulic cylinder length

```
# Geometry variables of driving mechanisms Case 3

delta       = 0.25               # meters   , Opening height of valve
Ltube       = 20                # meters     , length of fluid tubes
Lcylup      = np.linspace(delta + 0.01 , 6 , 60) - delta      # meters , Length of cylinder
Lcyldown    = 3.3 - Lcylup      # meters     , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5               # meters     , Length of piston including rod
Rpiston     = 0.1               # meters     , Radius of piston
tpiston     = 0.05              # meters     , thickness of piston
Lrod        = 3                 # meters     , Length of connecting rod from piston to top of valve
Rrod        = 0.25              # meters     , radius of connecting rod
trod        = 0.02              # meters     , thickness of rod edge
Rcyl        = 0.25              # meters     , Inner radius of the hydraulic cylinder
tcyl        = 20                # milimeters , Thickness of the cylinder
Lculvert    = 10        # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 4 – Rod thickness

```
# Geometry variables of driving mechanisms Case 4

delta       = 0.25               # meters   , Opening height of valve
Ltube       = 20                # meters     , length of fluid tubes
Lcylup      = 3.3 - delta       # meters     , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup      # meters     , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5               # meters     , Length of piston including rod
Rpiston     = 0.1               # meters     , Radius of piston
tpiston     = 0.05              # meters     , thickness of piston
Lrod        = 3                 # meters     , Length of connecting rod from piston to top of valve
Rrod        = 0.25              # meters     , radius of connecting rod
trod        = np.linspace(0.001 ,Rrod)        # meters      , thickness of rod edge
Rcyl        = 0.25              # meters     , Inner radius of the hydraulic cylinder
tcyl        = 20                # milimeters , Thickness of the cylinder
Lculvert    = 10        # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 5 – Rod length

```
# Geometry variables of driving mechanisms Case 5

delta       = 0.25                   # meters   , Opening height of valve
Ltube       = 20                  # meters      , Length of fluid tubes
Lcylup      = 3.3 - delta         # meters      , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup        # meters      , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5                 # meters      , Length of piston including rod
Rpiston     = 0.1                 # meters      , Radius of piston
tpiston     = 0.05                # meters      , thickness of piston
Lrod        = np.linspace(0.1 ,10 , 100)# meters,Length of connecting rod from piston to top of valve
Rrod        = 0.25                # meters      , radius of connecting rod
trod        = 0.02             # meters      , thickness of rod edge
Rcyl        = 0.25                # meters      , Inner radius of the hydraulic cylinder
tcyl        = 20                  # milimeters , Thickness of the cylinder
Lculvert    = 10          # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 6 – Piston diameter

```
# Geometry variables of driving mechanisms Case 6

delta       = 0.25                   # meters   , Opening height of valve
Ltube       = 20                  # meters      , Length of fluid tubes
Lcylup      = 3.3 - delta         # meters      , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup        # meters      , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5                 # meters      , Length of piston including rod
Rpiston     = np.linspace(0.01 , 1 , 100)            # meters      , Radius of piston
tpiston     = 0.05                # meters      , thickness of piston
Lrod        = 3                 # meters      , Length of connecting rod from piston to top of valve
Rrod        = 0.25                # meters      , radius of connecting rod
trod        = 0.02             # meters      , thickness of rod edge
Rcyl        = 0.25                # meters      , Inner radius of the hydraulic cylinder
tcyl        = 20                  # milimeters , Thickness of the cylinder
Lculvert    = 10          # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 7 – Piston length

```
# Geometry variables of driving mechanisms Case 7

delta       = 0.25                   # meters   , Opening height of valve
Ltube       = 20                  # meters      , Length of fluid tubes
Lcylup      = 3.3 - delta         # meters      , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup        # meters      , Lenght of cylinder compartment beneath piston
Lpiston     = np.linspace(0.1 , 10 , 100)           # meters      , Length of piston including rod
Rpiston     = 0.1          # meters      , Radius of piston
tpiston     = 0.05                # meters      , thickness of piston
Lrod        = 3                 # meters      , Length of connecting rod from piston to top of valve
Rrod        = 0.25                # meters      , radius of connecting rod
trod        = 0.02             # meters      , thickness of rod edge
Rcyl        = 0.25                # meters      , Inner radius of the hydraulic cylinder
tcyl        = 20                  # milimeters , Thickness of the cylinder
Lculvert    = 10          # meters      , Length of the culvert in which valve is installed (gate width)
```

Case 8 – Tube length

```
# Geometry variables of driving mechanisms Case 8

delta       = 0.25                # meters   , Opening height of valve
Ltube       = np.linspace(0.1,20)# meters      , Length of fluid tubes
Lcylup      = 3.3 - delta         # meters      , Length of cylinder compartment on top of piston
Lcyldown    = 3.3 - Lcylup        # meters      , Lenght of cylinder compartment beneath piston
Lpiston     = 3.5                 # meters      , Length of piston including rod
Rpiston     = 0.1                 # meters      , Radius of piston
tpiston     = 0.05                # meters      , thickness of piston
Lrod        = 3                   # meters      , Length of connecting rod from piston to top of valve
Rrod        = 0.25                # meters      , radius of connecting rod
trod        = 0.02                # meters      , thickness of rod edge
Rcyl        = 0.25                # meters      , Inner radius of the hydraulic cylinder
tcyl        = 20                  # milimeters , Thickness of the cylinder
Lculvert    = 10          # meters      , Length of the culvert in which valve is installed (gate width)
```

Variables boundary conditions

This analysis is used to obtain insight in the behaviour of different damping and stiffness components for different boundary conditions of the system. The relevant boundary conditions are determined to be the 1) water level difference, 2) valve opening height, 3) pressure underneath valve and 4) varying stiffness of the system(damping only).

Water level difference

```
#Variables - cases
delta1      = 0.2                       # meters   , Opening height of valve
h11         = np.linspace(8,0.5,100)    # meters   , Waterdepth upstream
h21         = 0                         # meters   , Waterdepth downstream
dH1         = h11 - h21                 # meters   , Water level difference over valve
p1         = 10000                      # kPa      , pressure underneath valve
```

Valve opening height

```
#Variables - cases
delta2      = np.linspace(0.001,2,100)    # meters   , Opening height of valve
h12         = 4                           # meters   , Waterdepth upstream
h22         = 0                           # meters   , Waterdepth downstream
dH2         = h12 - h22                   # meters   , Water level difference over valve
p2         = 10000                        # kPa      , pressure underneath valve
```

Pressure

```
#Variables - cases
delta3      = 0.2                           # meters   , Opening height of valve
h13         = 4                             # meters   , Waterdepth upstream
h23         = 0                             # meters   , Waterdepth downstream
dH3         = h13 - h23                     # meters   , Water level difference over valve
p3         = np.linspace(50000 , 0 , 100)   # kPa      , pressure underneath valve
```

Stiffness

```
#Variables - cases
delta4      = 0.2           # meters   , Opening height of valve
h14         = 4             # meters   , Waterdepth upstream
h24         = 0             # meters   , Waterdepth downstream
dH4         = h14 - h24     # meters   , Water level difference over valve
p4         = 10000          # kPa      , pressure underneath valve

#Stiffness
k44 = np.linspace(1,10**10) # N/m    , Varying stiffness of the system
```

Variables tuning dynamic characteristics

Based on the sensitivity analysis an optimal solution is examined. Based on the relative influence on the natural frequency taking the additional mass into account, an attempt is done to find the variable that is adjusted best.

```
var1 = np.linspace(Rcyl/10 , Rcyl * 10 , 1000)      # cylinder radius , default = 20 mm
var2 = np.linspace(Lcylup/10 , Lcylup * 10 , 1000)  # cylinder length , default = 3.3 m
var3 = np.linspace(trod / 10 , trod * 10 , 1000)    # Rod thickness , default = 0.02 m
var4 = np.linspace(Ltube / 10 , Ltube * 10 , 1000)  # Tube length , default = 1 m
```

Equations

The formulas and relations that are mentioned in the literature study are all incorporated in the python scripts. An overview is given below.

Explanation of different formulas

```
# Explanation different formulas
#  added_mass     : Determines the added mass present underneath the valve using Kolkman (1996)
#  mass_rod       : Determines the mass of the rod and other circular shaped elements
#  mass_valve     : Determines the mass of the valve assuming it is a rectangular shaped body
# axial_stiffness: Determines stiffness due to axial deformation circular elements
# bending_stiff  : In case of horizontal installed piston
# fluid_stiffness: Makes use of Bulk modulus to determine the stiffness of the fluid in cylinder
# archimedes     : Floating stiffness of the valve
# flow_stiffness : Added water stiffness due to flow dFy/dy = k , Fy is flow force y direction
# sudden_stiffness: Stiffness due to sudden flow, derived from Fperm in x direction
# Coulomb_damping : Friction damping in cylinder and valve
# material_damping : Energy dissipation in axial deformation of elements
```

Mass formulas

```
# Mass formulas

def added_mass(b , rho , W , delta):
    a = 2 * b
    m = 2 / 3 * rho * a ** 3 * W / delta
    return m

def mass_rod(rho_rod , R , t , L ):
    rho = rho_rod
    R_in = R
    R_out = R + t
    A = np.pi * (R_out ** 2 - R_in ** 2)
    V = A * L
    m = V * rho
    return m

def mass_valve(rho_valve , b , W, h):
    rho = rho_valve
    m = b * W * h * rho
    return m

def mass_cylinder(Rcyl , tcyl , material , Lcylup , Lcyldown):
    L = Lcylup + Lcyldown
    R_out = Rcyl + tcyl
    R_in = Rcyl
    A = np.pi * (R_out ** 2 - R_in ** 2)
    if material == 'steel':
        rho = 8050
    if material == 'aluminium':
        c = 2700
    return A * L * rho
```

Damping formulas

```
#Damping formulas

def coulomb_damping( mu_gate , omega , k , dH , F , W , h):
    g = 9.81
    Fp = rho * g * dH * W * h
    Fw = mu * Fp / 2
    y_head = np.mean(F) / k
    c = 4 * Fw / ( np.pi * omega * y_head)
    return c

def material_damping(k , m , material):
    # m = mass of element
    # k = stiffness of element

    eta_steel   = 0.004
    eta_alu     = 0.018
    if material == 'steel':
        c = eta_steel * np.sqrt ( k * m )
    if material == 'aluminium':
        c = eta_alu * np.sqrt ( k * m )
    return c

def suction(delta , blow , W , dH , S , mu , rho , Lculvert ):
    m = mu                      # Discharge/Contraction coefficient of gate opening
    Ci = Lculvert / delta
    if delta < 0.65 * blow:
        Cs = delta
    else:
        Cs = 0
    c = m * Ci * rho * Cs * b * np.sqrt ( 2 * 9.81 * dH )
    return -c

def suction_array(delta , blow , W , dH , S , mu , rho , Lculvert ):
    Cs = np.zeros(len(delta))
    csuc = np.zeros_like(Cs)
    Ci = np.zeros_like(Cs)

    for i in range(len(delta)):
        if delta[i] < 0.65 * blow:
            Cs[i] = delta[i]
        else:
            Cs[i] = 0
        Ci[i] = Lculvert / delta[i]
    for j in range(len(delta)):
        csuc[j] = mu * Ci[j] * rho * Cs[j] * b * np.sqrt ( 2 * 9.81 * dH )
    return -csuc
```

Stiffness formulas

```python
#Stiffness formulas

def axial_stiffness(E , R , t , L):
    A_out = np.pi * ( R + t ) ** 2
    A_in  = np.pi * R ** 2
    A = A_out - A_in
    k = E * 10 ** 9 * A / L
    return k

def fluid_stiffness(B , R , Lcyl):
    D = 2 * R
    L = Lcyl
    k = B * 10 ** 9 * np.pi * D ** 2 / ( 4 * L )
    return k

def archimedes_stiffness(rho , W , b):
    k = rho * W * b * 9.81
    return k

def flow_stiffness(rho , dH , W , alpha , Cw):
    k = rho * 9.81 * dH * W * np.tan(alpha) * Cw
    return k

def sudden_stiffnes(delta , W , mu , rho , dH , Cw):
    #  mu = contraction coefficient
    A = delta * W * mu
    V = np.sqrt(2 * 9.81 * dH)
    Fperm = 0.5 * rho * A * V ** 2 * Cw
    k = 2 * Fperm / delta
    return k

def tube_stiffness(B , Rcyl , Ltube):
    return B * 10 ** 9 * np.pi * Rcyl ** 2 / Ltube
```

Excitation frequency

```python
# Forcing

def strouhal( delta , blow ):
    S = 0.5 * (delta / blow ) + 0.05
    return S

def omega_force(S , dH , delta ):
    V = np.sqrt ( 2 * 9.81 * dH)
    f = S * V / delta
    return f * 2 * np.pi

def Force(p0 , W , b):
    # n = resolution = step interval
    # Random = yes/Yes or no/NO  , returns a varying or constant force over frequencies

    F = p0 * W * b
    return F

def omega_natural( k , m ):
    return np.sqrt( k / m )
```

Appendix E – Python script calculations

This appendix serves as an overview of the different calculations that are made in terms of:

- Mass
- Damping
- Stiffness

These three system core characteristics eventually lead to other characteristics that describe the dynamics of the system. All cases of the sensitivity analysis as well as the tuning of dynamic characteristics python scripts are given in this appendix.

Sensitivity analysis & boundary conditions

Since all calculations are the same, except from the input variables, only the reference case will be treated in this section.

```python
# Dynamic characteristics  - Case 0

instal_direction            = 'vertical'
material_piston             = 'steel'
material_rod                = 'steel'
material_cylinder           = 'steel'

if material_piston == 'steel':
    E = E_steel
    material = 'steel'
else:
    E = E_alu
    material = 'aluminium'

#masses
m1 = added_mass(b , rho , W , delta)
m2 = mass_rod(rhosteel , Rrod , trod , Lrod )
m3 = mass_valve(rhosteel , b , W, h)
m4 = mass_rod(rhosteel , Rpiston , tpiston , Lpiston )
m5 = mass_cylinder(Rcyl , tcyl , material , Lcylup , Lcyldown)
m  = m1 + m3 + m2 + m4

#Stiffness
k1 = axial_stiffness(E , Rrod , trod , Lrod)
k2 = axial_stiffness(E , Rpiston , tpiston , Lpiston)
k3 = fluid_stiffness(B_oil , Rcyl , Lcylup)
k4 = archimedes_stiffness(rho , W , b)
k5 = flow_stiffness(rho , dH , W , alpha , Cw)
k6 = sudden_stiffnes(delta , W , mu , rho , dH , Cw)
k7 = tube_stiffness(B_oil , Rcyl , Ltube)
k = 1 / ( 1 / k1 + 1 / k2 + 1 / k3 + 1 / k7 ) + k4 + k5 + k6

#Forcing
omega_nat = omega_natural( k , m )
F = Force(p0 , W , b)
S = strouhal( delta , blow )
omega = omega_force(S , dH , delta )

# Damping
c1 = coulomb_damping( mu_gate , omega , k , dH , F , W , h)
c2_rod = material_damping(k1 , m2 , material)
c2_piston = material_damping(k2 , m4 , material)
c2 = c2_rod + c2_piston
c03 = suction(delta , blow , W , dH , S , mu , rho , Lculvert)
c0 = c1 + c2 + c03
```

Tuning dynamic characteristics

No further explanation about the calculations is needed since it can already be found in the previous section.