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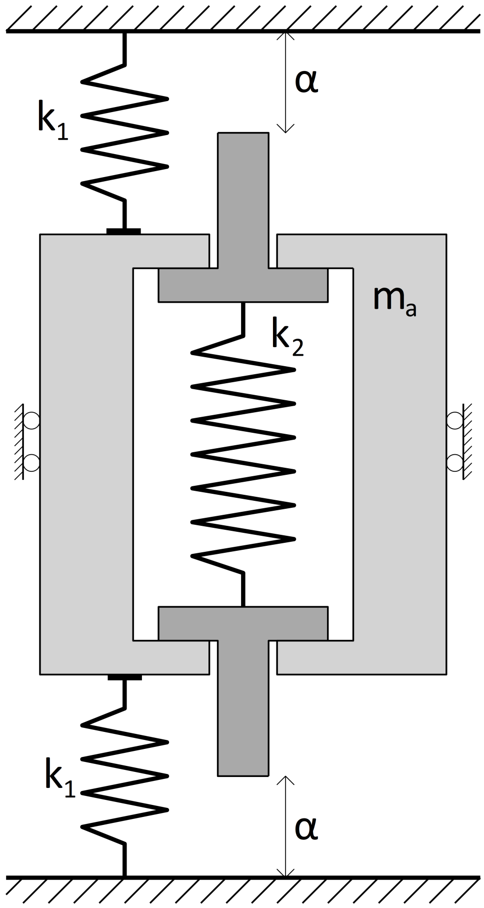

Investigation of feasibility of Auto Tuning Vibration Absorber

Master's thesis in Applied Mechanics

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Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
An illustration of the Auto Tuning Vibration Absorber.

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Abstract

The thesis is an investigation of the Auto Tuning Vibration Absorber (ATVA) developed by Swerea IVF. Main focus is to improve the absorber so it becomes active for more frequencies and thereby get a more stable behavior and also to adapt it for different systems. Here the focus is on three systems; a pneumatic breaker, a test rig and a reciprocating saw.

The investigation is made mainly by performing simulations in MATLAB, but also experiments on the test rig. To find out how stable the systems are the simulations are done in three different ways; one with increasing frequency, one with decreasing frequency and one with each frequency by itself. The experiments were done in different ways, but only with increasing and decreasing frequencies.

The ATVA consists of an auxiliary mass that shall counteract the vibrations from the main mass with help of a nonlinear spring system. Earlier it has been a gap between the main mass and the auxiliary mass which makes the masses to not be in contact all the time and this has given the system an unstable behavior, especially when working horizontally without affect of the gravity. To get it more stable it is tested to decrease the gap or to add a spring instead of the gap. This is shown to be very successful and give the system a more stable behavior without worsen other results as much. But the springs and gap needs to be carefully tuned for the systems to work well in their working frequency and to keep the advantages that the nonlinearity has.

Keywords: Auto Tuning Vibration Absorber, nonlinear springs, impact machines

Preface

This master's thesis is submitted as the final requirement for my master's degree in Applied Mechanics from Chalmers University of Technology. The project was executed during a period of 20 weeks, from September 2017 to January 2018. It was conducted on behalf of and in close cooperation with Swerea IVF, a Swedish research institute within production and product development. My supervisor at Swerea IVF was Lic. Eng. Hans Lindell and Professor Viktor Berbyuk was my examiner at Chalmers.

Acknowledgements

There are three people I would like to thank for the opportunity to be part of this interesting work and for great help along the way to successfully finishing this thesis. My first thanks goes to my examiner Viktor Berbyuk, for his good feedback, ideas and general help. I am also truly grateful to Hans Lindell, my supervisor at Swerea IVF, for being there with insightful thoughts and encouraging enthusiasm for this thesis and engineering thinking in general. And also a big thanks to Snævar Leó Grétarsson at Swerea IVF for his support and introducing me to the equipment. Many thanks to all three of you.

Nomenclature

Abbrevations

ATVA	Auto Tuning Vibration Absorber
LTVA	Linear Tuned Vibration Absorber
TVA	Tuned Vibration Absorber
WOTVA	Without Tuned Vibration Absorber
PB	Pneumatic Breaker
TR	Test Rig
RS	Reciprocating Saw
RMS	Root Mean Square

Variables

a	Length of gap between main mass and auxiliary mass
α	Length to second spring between main mass and auxiliary mass
c_a	Damping between auxiliary mass and main mass
c_h	Damping between housing mass and main mass
c_m	Damping between main mass and ground
c_p	Damping between user and housing mass
f	Excitation frequency
f_{ref}	Reference frequency when measuring force
F_0	Preload on second auxiliary spring
F_c	Damping force
F_e	Excitation force
$F_{e,ref}$	Measured force on the main mass at reference frequency
F_k	Nonlinear spring force acting on main mass and auxiliary mass
k_1	Spring stiffness of the first/extra spring between main mass and auxiliary mass
k_2	Spring stiffness of the second spring between main mass and auxiliary mass
k_a	Total nonlinear spring stiffness between auxiliary mass and main mass
k_h	Spring stiffness between housing mass and main mass
k_m	Spring stiffness between main mass and ground
k_p	Spring stiffness between user and housing mass
m_a	Auxiliary mass
m_h	Housing mass
m_m	Main mass

t	Time
x	Global coordinate
x_1	Displacement on main mass
x_2	Displacement on auxiliary mass
x_3	Displacement on housing mass
x_{rel}	Relative displacement between auxiliary mass and main mass

Contents

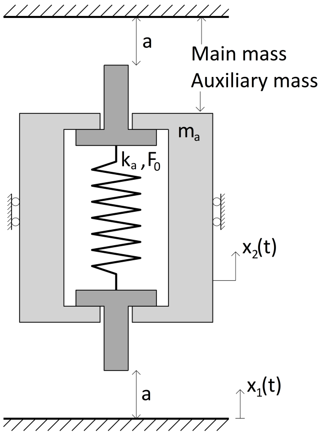
1 Introduction

This master's thesis is part of a bigger project at Swerea IVF, where the goal is to eliminate vibration injures from machine tools. The thesis is carried out together with Swerea IVF and Chalmers University of Technology and the goal is to investigate the range of machines' size and frequency where an Auto Tuning Vibration Absorber (ATVA) can be implemented.

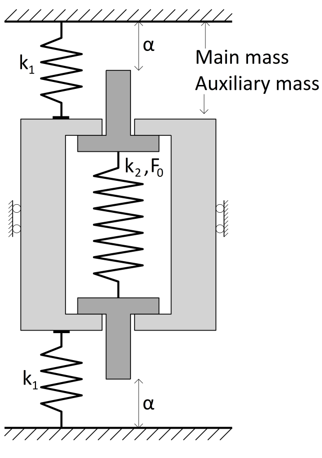
1.1 Background

One of the most common injuries on humans in the industry is caused by vibrating tools and is called Hand-Arm Vibration Syndrome (HAVS) or White Finger Syndrome. These injury occur when one uses tools that are vibrating in dangerous frequencies and amplitudes. Symptoms that can appear are for example numbness and cold intolerance in the fingers. Swerea IVF is running a project with a final goal of eliminating the dangerous vibrations from the machines translating into the user. [1, 2]

By focusing on reciprocating machine vibrations Swerea IVF has developed technology of an ATVA, which reduces the vibrations for the tools. There are other ways of decreasing the vibration, one common thing is to isolate the handle with system of springs or dampers. The main idea of how the ATVA works is that an auxiliary mass produces a counter force that counteracts the primary vibrations. In Figure 1.1 there is a basic illustration of the technique. [3, 4] Earlier one have tested with a linear system, but that has been only working in a narrow range of frequencies. It is often calibrated to work well in one frequency but it can be very sensitive for frequencies close by. By adding a nonlinear part it has been proven that it works in a wider frequency range than before. The ATVA is a kind of nonlinear tuned vibration absorber (NLTVA) but Swerea IVF has called it ATVA because it is adjusting itself after the tool's frequency. One can say it is auto tuning its frequency mechanically after the tool's working frequency. The parameters which make it nonlinear are the gap (a) and the preload in the spring (F_0). When the gap is replaced with a spring, see Figure 1.1b, that distance is named α.



(a) *ATVA with a free gap.*



(b) *ATVA with an extra spring instead of a gap.*

Figure 1.1: *A basic illustration of the ATVA. The auxiliary mass is supposed to move up and down in counter phase to the main mass.*

It is shown that this ATVA technology works for medium sized hand-held tools, as rammers and jackhammers. But they have a problem for some tools with having an unstable behavior of the auxiliary mass, especially when the ATVA is working horizontally. Next step is to investigate in which range of tools the ATVA technology can be implemented and how to get it more stable. For example how low or high can the frequency be and what size, when it comes to the tool's mass, can it handle. It is also in focus an analysis of what values on the gap (a and α), mass (m_a), total spring stiffness (k_a) and preload in spring (F_0) are needed to make the ATVA work and have a stable behavior.

There are also some more advanced systems with MR-damper or equal. But these solutions increase the complexity of the system and can make it more sensitive for other resonance frequencies which can lead to an increase of the vibrations. Because of that those options are not used in this project.

1.2 Purpose

The project's purpose is to investigate the range of sizes and frequencies in machine tools the Auto Tuning Vibration Absorber can be implemented in and how to adapt it. This investigation is mainly made with mathematical models and experiments in that range available equipment at Swerea IVF is capable of. The main focus will be on three systems: pneumatic breaker, test rig and reciprocating saw.

- What is activating the ATVA?
- What can be done to get the ATVA activated?
- How low/high frequencies can the ATVA work with?
- What kind of changes when it comes to mass, spring stiffness, gap length and preload in the spring are needed to be done to get the ATVA to work in other sizes of tools?

1.3 Limitations

- Only looking at characteristics of existing machine tools. For example, only frequencies the tools use not all of them.
- Limited time of 20 weeks of work.
- Experiments will only be done on the test rig.

2 Theory

In this chapter one can read about the theory behind what is used in the project. For example basics about tuned vibration absorbers, equations of motions and more theory about the systems (pneumatic breaker, test rig and reciprocating saw).

2.1 Basics about TVA

Tuned Vibration Absorber (TVA) or also called tuned mass damper has been used for a long time to suppress vibrations. It is commonly used in buildings, bridges and machines. The basic idea is to add an extra mass to the system which moves and suppresses the vibration of the main system. This can for example prevent fatigue failures and discomfort. The TVA can be shaped in many different ways. One of the most famous example is the pendulum in Taipei 101 and one can go back to 1911 when Hermann Frahm's patent *Device for damping vibrations of bodies* was granted, which was the first patent for a TVA. [5] A basic model of a TVA is shown in Figure 2.1 where the excitation force (F_e) is causing the main mass to move and by the connection to the auxiliary mass it starts to move with some delay because of the spring. This makes the masses to not work in same phase which lead to m_a counteracts m_m and decreases the movement in m_m. The damper is not necessary but with damping the resonance peaks decreases which is good, but the frequencies one want to suppress does not become completely zero which is possible without damping. [6]

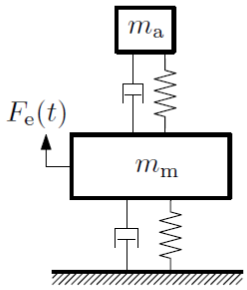


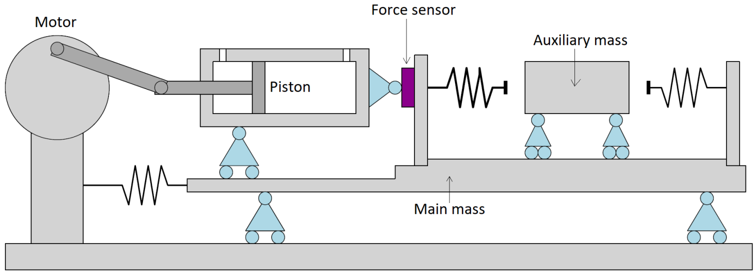
Figure 2.1: *An engineering model of a TVA with a modelled spring and damper connection.*

The TVA is often adapted to suppress the vibrations for specific frequencies. Because it can be sensitive and is often only able to suppress the movement in a close range of frequencies. Often the TVA is designed to decrease the working frequencies or resonance frequencies for the main system, because those frequencies makes the biggest risk for the system.

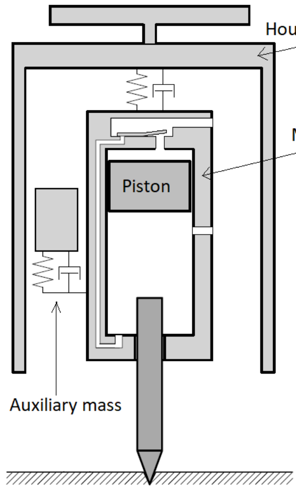
2.2 Models

A basic model of the test rig and pneumatic breaker can be seen in Figure 2.2. Pictures of all tools can be seen in Appendix A. As one can see they are structured in a similar way with a piston that affects the main mass and an added auxiliary mass. The pneumatic breaker also has a housing, which is not of interest in this project because the focus is on the auxiliary and main mass. The reciprocating saw also has the same basics, but unfortunately there is no illustrated model of it. Because of these similarities it is possible to make a simplified model of all reciprocating tools. The ones that are of interest for this project has a basic system with two masses, a main mass and a housing mass, with a connection to the ground, what it acts on, and to the person who uses the tool. The contact between the ground and the main mass is modelled as a linear spring and damper. Similar model is used for the contact between both the main mass and housing mass and the housing mass and user. This is illustrated in the left system in Figure 2.3 and the other two show systems with an added auxiliary mass.

To suppress the vibration generated in the main mass one add an auxiliary mass to the main mass with a modelled spring and damper connection. This can be seen in the middle and right system in Figure 2.3, where the middle system has a linear spring and the right system has a nonlinear spring. These are then modelled to make the auxiliary mass to counter act the movement in the main mass and by that decrease the vibrations translating into the user.



(a) *Illustration of the test rig. Where the motor drives the piston in the cylinder with small air inlets. This causes movement of the main mass and auxiliary mass. The force sensor is between the cylinder and the main mass.*



(b) *Illustration of the pneumatic breaker.*

Figure 2.2: *Basic drawings of the test rig and the pneumatic breaker.*

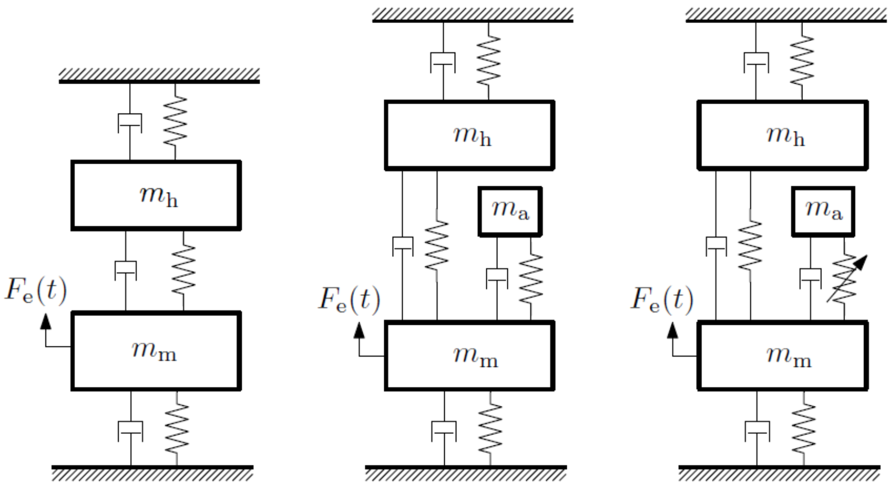
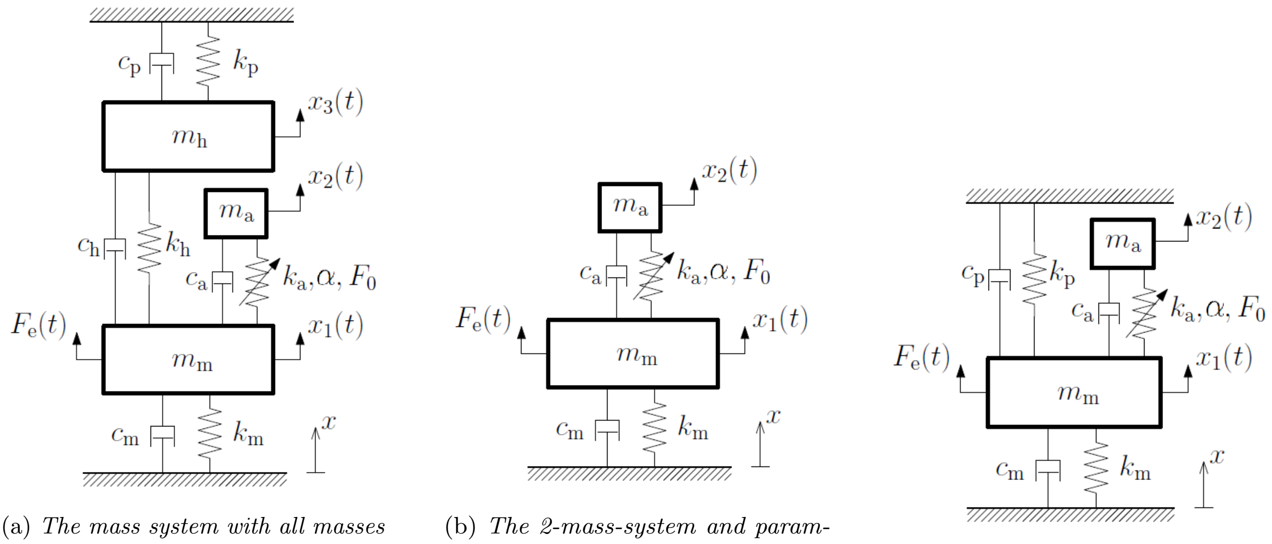


Figure 2.3: *A basic model of the tools. The first from the left is without a TVA, second one is with a Linear TVA (LTVA) and the third with an ATVA. There the upper boundary is the person holding the tool and the lower boundary is the subject the tool works on.* m_m = *main mass,* m_h = *housing mass and* m_a = *auxiliary mass.*

For the ATVA, Swerea IVF has used a gap and a preload to make it nonlinear, as one can see in Figure 1.1. A problem that has appeared when using a gap is that the small displacements on the main mass do not affect the auxiliary mass, because they need to be in contact to have affect. Especially if working in a horizontal direction where the gravity is not affecting. Which do that the auxiliary mass do not start move and cannot then create a counter force to counteract the vibrations. To activate the ATVA the main mass needs to get in contact with the auxiliary mass by getting enough displacement. Otherwise one probably can add some activator on the auxiliary mass to get it started. This activator can be for example an extra spring instead of the gap, but there is a risk to lose wanted qualities to add that.

2.2.1 Equations of motion

The equations of motion are used to compute the movement of the masses in the system. Earlier master's thesis focused on decreasing the vibration in the housing mass, m_h. [3] But in this project the focus is on the auxiliary mass, m_a, and decreasing the vibration in the main mass, m_m. Especially, as mentioned before, to activate the m_a for all frequencies. With that as background the system is simplified to a 2-mass-system as can be seen in Figure 2.4.



(a) *The mass system with all masses and parameters there are used to get the equations of motion for the whole tool. [3]*

(b) *The 2-mass-system and parameters there are used to get the equations of motion for the parts which is of interest in this project.*

(c) *The system for the test rig and the reciprocating saw and their parameters in the equations of motion.*

Figure 2.4: *The engineering model for the whole tool (a), the simplified 2-mass-system (b), the test rig and the reciprocating saw (c).*

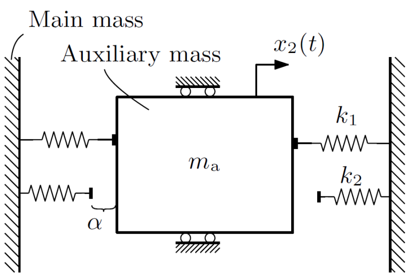


Figure 2.5: *Basic model of how the nonlinearity appears in the system. Where k_1 is the stiffness for the first spring and k_2 the stiffness for the second spring. Together they become the total spring, k_a.*

Pneumatic breaker, 3-mass-system with gap

This system can be seen in Figure 2.4a, where the nonlinear spring is made by a gap, $k_1 = 0$, and a spring with a preload, F_0. It looks like Figure 2.5 without k_1, which also turn α into a because of the free gap. The equations of motion then becomes as Equation 2.1 with definitions in Equations 2.2-2.5. [3]

$$\begin{bmatrix} m_m & 0 & 0 \\ 0 & m_a & 0 \\ 0 & 0 & m_h \end{bmatrix} \ddot{\boldsymbol{x}} + \begin{bmatrix} c_m + c_h & 0 & -c_h \\ 0 & 0 & 0 \\ -c_h & 0 & c_h + c_p \end{bmatrix} \dot{\boldsymbol{x}} + \begin{bmatrix} k_m + k_h & 0 & -k_h \\ 0 & 0 & 0 \\ -k_h & 0 & k_h + k_p \end{bmatrix} \boldsymbol{x} = \begin{bmatrix} F_{e,PB} + F_{k,PB3} + F_c - m_m g \\ -F_{k,PB3} - F_c - m_a g \\ -m_h g \end{bmatrix} \tag{2.1}$$

$$x_{rel} = x_2 - x_1 \tag{2.2}$$

$$F_k(\boldsymbol{x})_{PB3} = F_{k,PB3} = \begin{cases} F_0 + k_a(x_{rel} - a) & , x_{rel} > a \\ -F_0 - k_a(-x_{rel} - a) & , x_{rel} < -a \\ 0 & , -a \le x_{rel} \le a \end{cases} \tag{2.3}$$

$$F_c(\dot{\boldsymbol{x}}) = F_c = c_a \dot{x}_{rel} \tag{2.4}$$

$$F_e(t,f)_{PB} = F_{e,PB} = F_{e,ref}\Big(\frac{f}{f_{ref}}\Big)^2 sin(2\pi f t) \tag{2.5}$$

Pneumatic breaker, 2-mass-system

Before one has looked at the 3-mass-system but now one wants to look at specific qualities of the vibration absorber. To spare calculation time and have more focus on the damper. Now one only look at the 2-mass-system with the main mass and the auxiliary mass. The change can be seen in Figure 2.4. Also to compare what kind of affect the ATVA has on the main mass there will also be shown results for a system with only the main mass to see its vibration without any absorbers. Results from a LTVA will also be calculated to see if the ATVA perform better than the more commonly used LTVA. To compute these results Equations 2.6-2.11 with Equations 2.2 and 2.5 are used.

Without TVA The system without any absorber becomes one degree of freedom and the equation of motion for this case can be seen in Equation 2.6

$$m_m \ddot{x} + c_m \dot{x} + k_m x = F_{e,PB} - m_m g \tag{2.6}$$

With LTVA With a LTVA the system is two degrees of freedom and it has constant spring stiffnesses and damping factors. The equation of motion for this is written in Equation 2.7.

$$\begin{bmatrix} m_m & 0 \\ 0 & m_a \end{bmatrix} \ddot{\boldsymbol{x}} + \begin{bmatrix} c_m + c_a & -c_a \\ -c_a & c_a \end{bmatrix} \dot{\boldsymbol{x}} + \begin{bmatrix} k_m + k_a & -k_a \\ -k_a & k_a \end{bmatrix} \boldsymbol{x} = \begin{bmatrix} F_{e,PB} - m_m g \\ -m_a g \end{bmatrix} \tag{2.7}$$

With ATVA This system is similar to the LTVA but has a nonlinear spring force connection to the auxiliary mass. As one can see in Figure 2.5 the connection is made of two springs with different stiffnesses and length. Where α is the distance to the second spring starts to act. The equations of motion and conditions can be seen in Equations 2.8, 2.9, 2.2 and 2.5. One can also add a preload, F_0, in the springs to get a nonlinear spring force.

$$\begin{bmatrix} m_m & 0 \\ 0 & m_a \end{bmatrix} \ddot{\boldsymbol{x}} + \begin{bmatrix} c_m + c_a & -c_a \\ -c_a & c_a \end{bmatrix} \dot{\boldsymbol{x}} + \begin{bmatrix} k_m & 0 \\ 0 & 0 \end{bmatrix} \boldsymbol{x} = \begin{bmatrix} F_{e,PB} + F_{k,PB2} - m_m g \\ -F_{k,PB2} - m_a g \end{bmatrix} \tag{2.8}$$

$$F_k(\boldsymbol{x})_{PB2} = F_{k,PB2} = \begin{cases} F_0 + k_1\alpha + k_a(x_{rel} - \alpha) & , x_{rel} > \alpha \\ -F_0 - k_1\alpha - k_a(-x_{rel} - \alpha) & , x_{rel} < -\alpha \\ k_1 x_{rel} & , -\alpha \le x_{rel} \le \alpha \end{cases} \tag{2.9}$$

Test rig with ATVA

The equations of motion for the test rig is a combination of the 3-mass-system without the housing mass and a new excitation force. The results from these equations can be validated by doing experiments with the test rig. The equations are Equation 2.10, 2.11 together with Equation 2.2 and 2.9.

$$\begin{bmatrix} m_m & 0 \\ 0 & m_a \end{bmatrix} \ddot{\boldsymbol{x}} + \begin{bmatrix} c_m + c_a + c_p & -c_a \\ -c_a & c_a \end{bmatrix} \dot{\boldsymbol{x}} + \begin{bmatrix} k_m + k_p & 0 \\ 0 & 0 \end{bmatrix} \boldsymbol{x} = \begin{bmatrix} F_{e,TR} + F_{k,PB2} - m_m g \\ -F_{k,PB2} - m_a g \end{bmatrix} \tag{2.10}$$

$$F_e(t,f)_{TR} = F_{e,TR} = \begin{cases} \Big(\frac{28}{f-4.75} + 36.5\Big) sin(2\pi f t) & , 5.681 < f \\ (19.67f + 4) sin(2\pi f t) & , 2.5 < f \le 5.681 \\ 0.8 f sin(2\pi f t) & , 0 \le f \le 2.5 \end{cases} \tag{2.11}$$

Reciprocating saw with ATVA

The equations for the reciprocating saw is very similar as the equations for the test rig but has another excitation force. For this system one use Equation 2.12, 2.13 together with Equation 2.2 and 2.9.

$$\begin{bmatrix} m_m & 0 \\ 0 & m_a \end{bmatrix} \ddot{\boldsymbol{x}} + \begin{bmatrix} c_m + c_a + c_p & -c_a \\ -c_a & c_a \end{bmatrix} \dot{\boldsymbol{x}} + \begin{bmatrix} k_m + k_p & 0 \\ 0 & 0 \end{bmatrix} \boldsymbol{x} = \begin{bmatrix} F_{e,RS} + F_{k,PB2} - m_m g \\ -F_{k,PB2} - m_a g \end{bmatrix} \tag{2.12}$$

$$F_e(t,f)_{RS} = F_{e,RS} = F_{e,ref}\left(\frac{f}{f_{ref}}\right)^2 sin(2\pi f t) \tag{2.13}$$

2.2.2 Parameters

In Table 2.1 one can see the parameters for each different case which are tested during the project. The most data is given from earlier measurements by Swerea IVF, with a few updates that are described in this report. The biggest change from before is as written, that the 2-mass-system is in focus and not the 3-mass. A "-" in the table means that the parameter is not used.

Table 2.1: Parameters for the different systems.

Parameter	Unit	Pneumatic breaker	2-mass Pneumatic breaker	Test rig	Reciprocating saw
m_m	$[kg]$	2.7	2.7	4.292	3.9
m_a	$[kg]$	1.0	1.0	1.010	0.5
m_h	$[kg]$	3.1	-	-	-
k_m	$[kN/m]$	0.5	0.5	1.255	0.4
k_1	$[kN/m]$	0	1.4	17.8	25
k_2	$[kN/m]$	164	164	52.8	40
k_a	$[kN/m]$	164	165.4	70.6	65
k_p	$[kN/m]$	14	-	1	1
k_h	$[kN/m]$	0.1	-	-	-
c_m	$[Ns/m]$	100	100	19.55	100
c_a	$[Ns/m]$	1	1	5	1
c_p	$[Ns/m]$	60	-	60	60
c_h	$[Ns/m]$	20	-	-	-
α	$[m]$	0.010	0.010	0.0011	0.0012
F_0	$[N]$	98.2	0	0	0
m_{pist}	$[kg]$	0.530	0.530	0.673	0.673
A_{pist}	$[m]$	0.030	0.030	0.030	0.030
$F_{e,ref}$	$[N]$	351	351	-	156
f_{ref}	$[Hz]$	26.1	26.1	-	44
g	$[m/s^2]$	9.81	9.81	0	9.81

2.2.3 MATLAB code

The main script starts with defining all parameters, as can be seen in Table 2.1 and which piston frequencies one wants to look at. Then it calls on the function TVA2_fr_XX, where XX is PB, TR or RS depending on what system one look at (PB=pneumatic breaker, TR=test rig, RS=reciprocating saw). This function gives the frequency response of the system. It starts with defining the initial state of the system. Then state how the spring force behaves depending on x_{rel}, with a smooth transition when the derivative is changing, see Figure 2.6, to decrease calculation time.

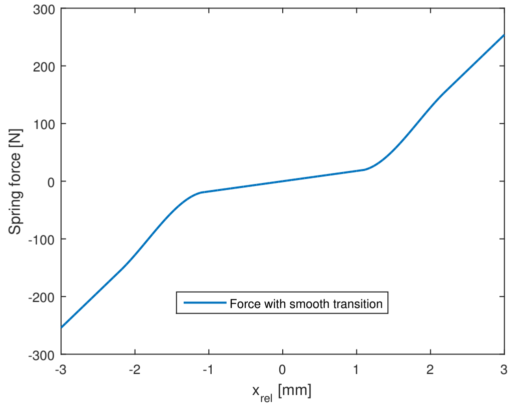


Figure 2.6: *Graph of how the spring force behaves in the simulations for the pneumatic breaker without a preload in the springs. A smooth transition is made when the slope is changing to save calculation time. The test rig and reciprocating saw has similar behavior but other values.*

Next step is to solve the ordinary differential equation (ODE) using an ODE solver in MATLAB. This will be done for each frequency and to limit the time the process ends when steady-state is found or when maximum allowed time is reached. From ODE solver one get the displacement and time points of the main mass and auxiliary mass for each frequency. These values are then used to calculate the RMS values of the displacement. Then the frequency response is calculated and one go back to the main script where chosen values is plotted.

3 Methodology

Here is the methodology for the thesis presented. It is divided into different sections depending on the subject and it follows in chronological order seen to when it has been done during the project.

3.1 Literature study

For the literature study focus was to find new papers and observations that was not published last time Swerea IVF did a study, which was when earlier master thesis was done in 2015. For the study one starts with a general search about tuned vibrations absorbers and vibrating tools. To get a basic knowledge about what the problem is and how the vibrations are generated. Some of these papers were given from the supervisor Hans Lindell and they are about earlier work that has been done at Swerea IVF. After that one need to do a deeper search in article registers and patents. A start was to check Google scholar and then go to ScienceDirect, Chalmers library and also Espacenet for specific patents. This search was to find related articles about LTVA and NLTVA. To be updated on the latest discoveries and products. But also to find articles that that can confirm the theory behind this project.

Because the NLTVA is called many different things, for example ATVA, it was a good thing to use all these names when searching. There are also a lot of applications in buildings and bridges, but the focus was on tools and machines which also was important to know when doing the search.

3.2 Mathematical model

With known information about the systems an engineering model was constructed and from that the mathematical model was derived. This has been done in earlier master thesis [3], but the new models are not exactly the same as the earlier ones. The new models are derived from that earlier model. This because most of the changes are simplifications of the earlier system. It was a few different systems that were derived and this because the pneumatic breaker, test rig and reciprocating saw has similar but not exactly the same behavior. For comparison every system has different cases for example without an absorber, with a LTVA and with an ATVA to make it easy to see the differences and improvements.

When the mathematical models were derived they are inserted into the MATLAB code, which was given from Swerea IVF. The code was also adapted after each system and case. The code was also verified by calculating cases where one know the results. In this case the known results are from the book *Structural Dynamics Control* by Viktor Berbyuk [7]. After a debugging of the code it was time for simulations.

3.3 Simulations

Simulations were performed in MATLAB for different systems and cases. Starting with the pneumatic breaker to get a feeling of what is happening and also to verify the results after adapting the MATLAB code to the 2-mass-system. Then a deeper analysis was done to see what needs to be done to get the ATVA to work for more frequencies. This was done for all three systems (pneumatic breaker, test rig and reciprocating saw). The three parameters which will be changed are k_1, k_2 and α. To make it easy one will always start to solve the issues by changing k_1 but if that is not enough one have a look on α and k_2. When starting the simulations all the systems have the same initial conditions which is depending on the gravity. With gravity one starts with a small displacement on m_m and m_a as the main spring was compressed by the weight of the tool, but x_{rel} is still zero. For the case without gravity all masses and springs starts from rest. To read more about how the simulation scripts work one can see section 2.2.3.

3.3.1 Pneumatic breaker

The simulations starts with running the script with a successive increasing frequency, always starting at 1 Hz and going passed the systems working frequency until it fails (m_a stops) or the vibrations gets worse in the m_m than without an ATVA. When the frequency is changing one use the last displacement and velocity values from the earlier frequency as initial conditions for the new frequency, which give a smoother transition and a more real behavior of a tool. Start the investigation of how stiff the first spring, k_1, needs to be by running through large range of stiffnesses. For the pneumatic breaker a range of stiffness around $0 - 5000N/m$ is enough. Then choose the lowest stiffness which is needed to activate the ATVA and keep it activated for the desired frequencies. If it seems to be hard to find a satisfying result by only changing k_1, one can start changing α and later also k_2 to see if that helps.

Then do the same thing but with a decreasing frequency. Starting at a frequency where it has failed and then go down to 1 Hz to see at which frequency it will reactivate. This time one loop over a closer range of stiffnesses, because hopefully the stiffness from earlier is also working for this case. Otherwise one needs to increase the stiffness until it gives satisfying results.

The third case is to restart the system for each frequency. For every new frequency one starts from the initial conditions. This shows how stable the ATVA is. Like if the ATVA would stop at a frequency one can see if it can be restarted at that frequency or if one need to change frequency to get it activated again. One run the simulation for the same frequencies as earlier cases and also starts with the stiffness which was satisfying at the last case. If the results are not satisfying for this case one change the stiffness until the requirements are fulfilled, one probably need a stiffer spring if that is the case.

At last, one simulate all the cases again, but this time with the stiffness from the last case. This is because of one need to see if all the cases give satisfying results. Because a too stiff, long or short first spring can give a very linear response, which is not desired. When the results are satisfying one compare the new simulated results with other systems, which are an ATVA with a gap, a LTVA and the system without an ATVA. This is to see if the changes really have been an improvement or not.

3.3.2 Test rig

The simulation of the test rig is very similar as for the pneumatic breaker. One go through the three steps with increasing, decreasing and restarting frequencies and after that check that everything is fine and compare to see the improvements.

But for this system one needs to check a wider range of values on k_1, $0 - 30000N/m$ and also change α to reach a satisfied result. The results were reach by manually iterating, tested different values and saw how the results changed. But still following the three steps as mentioned before.

Because of the test rig does not have a clear working frequency, but it has a maximum frequency around 50 Hz, it was decided to adapt the system for a frequency around 25 Hz. One can then easily reach desired frequencies when doing experiments and compare the results with each other.

3.3.3 Reciprocating saw

For the reciprocating saw the simulation strategy is also similar to the pneumatic breaker and test rig. One do the three steps and starts with testing different values on k_1, here the range are approximately $0 - 40000N/m$. But for this system one also needs to change both α and k_2 as well. This because of keeping the working frequency around $45 - 50Hz$. And as for the earlier systems a manual iteration is done to find parameter values which give satisfying results.

Satisfying results for the reciprocating saw is to have a minimum for the m_m around $45 - 50Hz$, have a stable movement on the m_a and suppress the vibration from as low frequency as possible. At last when comparing with other TVA-systems showing that it is an improvement.

3.3.4 Verification

To verify that the results are correct the simulation code will be used to solve elementary cases and less complex cases that can be solved analytically. The results is also compared with results from earlier measurements on the same or similar systems. If these test give good results the code is seen as correct and can be used for other cases with high credibility.

3.4 Experiments

Experiments was only done with the test rig. The test rig can be seen in Figure 3.1 and an illustrated model has been shown earlier in Figure 2.2a. The main idea of how the test rig works is that the motor drives the piston which affects the main mass and as an affect of that the auxiliary mass hopefully starts moving and everything is well monitored by sensors. Under section 3.4.1 one can see what kind of equipment that was used.

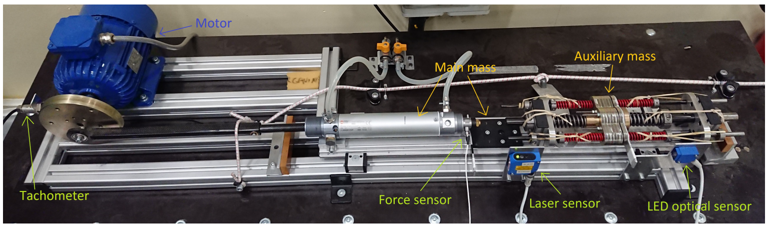


Figure 3.1: *Test rig setup.*

First step was to get a hint of how the system will respond. To get that one can manually change the rotation speed on the motor by turning a nob on the frequency inverter. By doing this one can in an easy and fast way see at which frequency the m_a will be activated, stop and be reactivated. With this knowledge one can adapt the settings for when it is time to do the measurements.

When doing measurements everything is controlled and monitored by LabVIEW SignalExpress in the computer. It is programmed to control the speed of the motor by sending voltage to the frequency inverter. For this experiments one will start at $200mV$ and go up to $8V$, which correspond to $40Hz$. The increase will be done with 200 steps between $200mV - 8V$ and every step will run for 2 seconds. As for the simulations different runs will be done, but because it is not possible to restart at each frequency that case will not be done. But both with an increasing and decreasing frequency will be done. One will run each case at least twice to be sure that no major error has happened during the measurements. Also run the cases separately not after each other, which is possible, to make it easier to separate the different cases in the data processing.

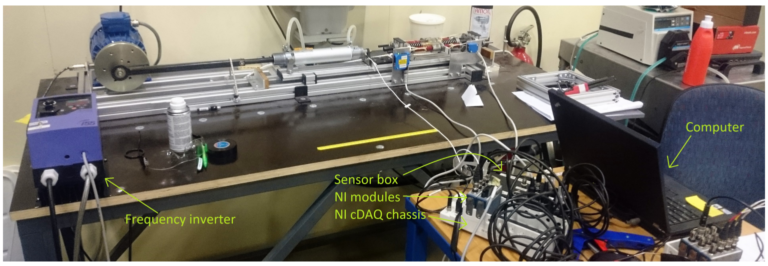


Figure 3.2: *Test rig setup with all equipment.*

3.4.1 Laboratory equipment

Here is a presentation of the laboratory equipment which is used. A picture of them can be seen in Figure 3.2 and 3.1.

Tachometer

The tachometer measures the rotation speed of the engine, which gives the frequency for the tool. The rotating axle it is measuring on is divided into four parts and each part gives a signal, which give a higher resolution of the frequency.

Laser sensor

The laser measures the displacement of the main mass. The laser sensor is fixed on the rig and are facing a flat area which comes out from the main mass.

LED optical sensor

This sensor measures the displacement of the auxiliary mass and are like the laser fixed on the rig and facing a flat area, but this one is on the auxiliary mass.

Force sensor

This is used to measure the force from the piston on the main mass. It is really the force from the compressed air, which are being compressed by the piston.

Sensor box

This box is directly connected to the sensors on the test rig. It processes the signals and send it to the Modules in the NI cDAQ.

Frequency inverter

The frequency inverter is used to control the rotating engine. It inverts the input voltage to a frequency, which the engine rotation speed will have. It can be controlled both by using the nob and the display on the box. Or it can be connected to a computer where one can program how the frequency shall change. This frequency inverter is a BEVI Type: ODE-2.

NI C Series modules

Two of these are used. The NI-9263 and the NI-9234, both of these are connected to the NI cDAQ chassis and receive signals from the sensor box. The NI-9263 is a simultaneously updating analog output module. The NI-9234 can measure signals from integrated electronic sensors such as accelerometers and tachometers.

NI cDAQ chassis

The NI cDAQ-9172 controls the timing, synchronization, and data transfer between NI C Series modules and the computer with Signal Express.

Computer

A Lenovo PC with the program LabVIEW SignalExpress installed.

3.4.2 Data processing

The sensors send signals to the sensor box which process it and send it forward to the NI modules which measure and convert the data together with the chassis also synchronize the time and from that it goes in to the computer and LabVIEW SignalExpress. From that relative data is manually exported to MATLAB where the graphs and comparisons with simulation data was done.

It is measuring the total movement of m_a, but the more interesting displacement is the relative movement against the main mass, to see how much the springs act. To get that one easily subtract the value from the laser senor from the values from the optical sensor. This is done in MATLAB with the RMS values which is exported from SignalExpress.

3.4.3 Validation

By comparing the experiment results with the results from the simulations one can validate computational model. The simulations are often first done with ideal input data. But those values are hard to get exactly right when doing it in practice. When the setup for the experiment is done one can control measure the distances and check that things that should be fixed are fixed and so on. If something incorrect is found, one change it or if not possible one change the simulation data to the actual data. For example if the gap should be $1mm$ and becomes $1.2mm$ it is good enough and one change the simulation data instead. Because moving by hand a lock ring $0.2mm$ is very hard. After all the adjustment and the simulation data is consistent with the setup it is time to run. One run the simulation and the experiment and then compare the results, which hopefully is similar to each other. If the results are similar the model is validated otherwise one need to investigate what is wrong.

This particular test rig has been validated carefully in earlier work [3, 8, 9] so this part do not have a big weight in this thesis. Because earlier work is trustful, but the results will of course be controlled if something unexpected would happen.

4 Results and discussion

In this chapter the results from the simulations and experiments are presented. The results are divided into different sections depending on what system is in focus. All parameters that are used can be seen in Table 2.1, if other values are used it is mentioned in the text. Also for all graphs that shows the frequency response it is the RMS value of the displacement, which is plotted on the y-axle.

4.1 Literature study results

There are very few papers related to this subject. Especially if one looks specific on machines connected to a NLTVA. There are a lot on buildings and wires, but those are often not applicable on machines. The main focus has also only been on new articles, those which have been published after the last study which was made in 2015. When searching more about tools and applications to reduce vibrations in tools and motors one discover that it is most common to use vibration reduction for eigenfrequencies. But this project is about to decrease the vibrations in a wider range, optimal would be to reach no vibration at every frequency.

4.2 Activate the ATVA

To activate the ATVA some different ways were tried, for example decrease the gap or adding an extra spring instead of a gap. It was soon found that the most effective and simple way was to add an extra spring. Because of that the following results have focus on what kind of a result it gets to add an extra spring to the systems. Early it has also been shown that the preload, F_0, has a very minor affect on the results. To make the calculations even easier F_0 is neglected in all of the calculations.

Already it is shown that the systems are pretty similar to each other. But the differences that have been affecting the results the most is the excitation force and the masses. As one can see in Figure 4.1 the forces are not the same. Especially the test rig which after 5Hz is almost constant compared to the others. The pneumatic breaker and the reciprocating saw have similar behavior but the force on the breaker is increasing faster. The force development of the test rig become like this because the piston has a constant amplitude and is then generating close to equal force for each stroke. While the other tools have an increasing amplitude of the piston when the frequency is increasing. This generates more power in each stroke. When it come to masses the main mass for each system (pneumatic breaker, test rig and reciprocating saw) are around $2.7kg$, $4.3kg$ and $3.9kg$ and the auxiliary masses are $1.0kg$, $1.01kg$ and $0.5kg$. Which give a rough approximation of the proportion to 1/3, 1/4 and 1/8. This affects the behavior of the system a lot and also make it harder to find a more general solution to the activating problem.

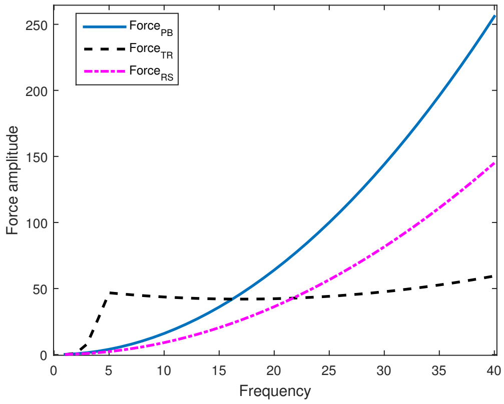
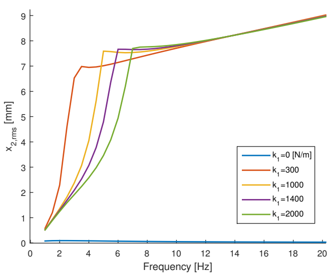


Figure 4.1: *Comparison of the forces applied in the three systems. (PB=pneumatic breaker, TR=test rig, RS=reciprocating saw)*

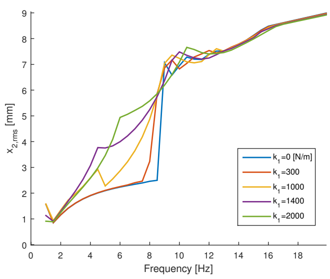
Another parameter that is problematic is the gravity. The test rig is always working in a horizontal direction, which is doing that the gravity is not affecting. But both the breaker and the saw can be used in all directions and due to that the affect of the gravity is varying a lot. From experience of the results, which one can see in the next sections, the worst case to get the ATVA activated is when there is no gravity. Because with gravity m_a is falling down on the springs and gets the desired contact. But with no gravity m_m needs to move enough to reach m_a. This makes the investigation more complex and for the affected systems results from cases with and without gravity will be shown, see next sections.

4.2.1 Pneumatic breaker

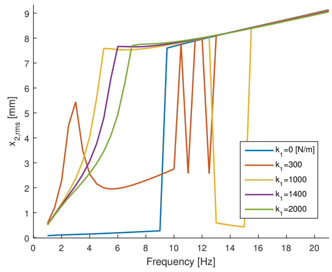
The first investigation was to look at how stiff k_1 needs to be to activate the ATVA already from low frequencies. Because the excitation force is increasing fast it activates with a gap from 10Hz or higher but to activate it from low frequencies it is enough to add a spring. In Figure 4.2 one can see results from different values on k_1 in different cases. For example, one can see a value of $300N/m$ is enough for case (a) but it is unstable in the two others but a value of $1400N/m$ is the lowest value for a stable behavior in all three cases. Because of those results $k_1 = 1400N/m$ is chosen.



(a) *Increasing frequency.*



(b) *Increasing frequency with gravity.*



(c) *Restart at every new frequency without gravity.*

Figure 4.2: *Comparison of the frequency response for m_a with different values on k_1 and cases.*

With chosen k_1 a comparison of the cases with and without gravity is done and the results can be seen in Figure 4.3. One can see that the behavior differ up to 17Hz but then they are very similar. This is probably because of in lower frequencies the masses move slower and then the gravity has enough time to affect the motions but after 17Hz it moves too fast and the forces become too big for the gravity to affect. The line in the middle of the figure shows the vibration of m_m if no m_a was added. As expected the vibrations are much lower with a m_a than without. The minimum RMS value is reached at $22Hz$ and measure $0.334mm$ without gravity, with gravity it is little higher $0.342mm$.

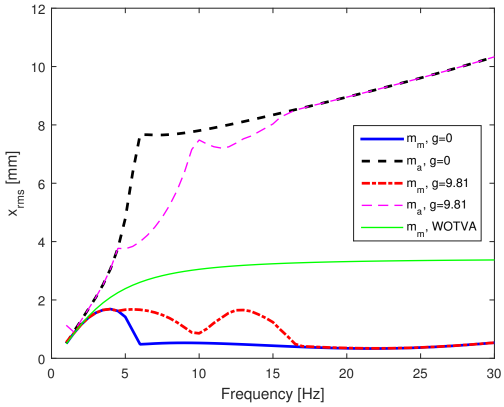
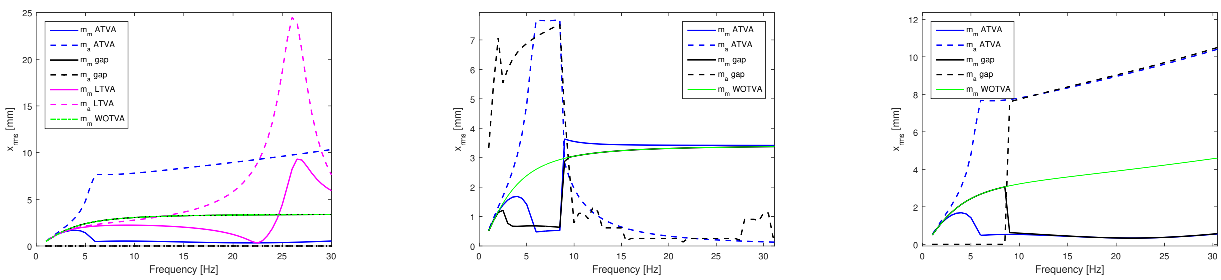


Figure 4.3: *Frequency response for the system with and without gravity.* $k_1 = 1400N/m$

In Figure 4.4 one can see three cases. As described in the method this three cases is (a) with an increasing frequency starting at 1Hz, (b) with a decreasing frequency started at 80Hz and (c) where the system restarts for each frequency from initial conditions.

The working frequency for the tool is around $25Hz$, which make this results satisfying. Especially that they keep this low level for a long time. It is first when reaching 50Hz when the vibrations gets worse than without m_a and it fails first over $70Hz$ for increasing frequency and reactivates at $44Hz$, with gravity, when decreasing the frequency which is way over the working frequency. A problem is that without gravity the ATVA reactivates first at $8Hz$ when decreasing the frequency. But to reach an acceptable level even for that case one need a first spring with stiffness around $20kN/m$ but then the vibrations for the lower frequencies are not suppressed as much. And this kind of pneumatic breaker is very rarely used in a horizontal direction. Also in case (c) the ATVA gets activated for every frequency up to $55Hz$, which shows that there are a chance of the ATVA to be reactivated at a higher frequency even when decreasing the frequency. When weighted all facts it was decided that a $k_1 = 1400N/m$ is good enough, when the pneumatic breaker is used properly.

If comparing the results with systems with an ATVA with a gap, LTVA or without a TVA one can see in Figure 4.4 and 4.5 that they have different behavior. With increasing frequency the ATVA with a gap never becomes activated while the LTVA and the ATVA with an extra spring become activated right away, when looking at systems without impact from the gravity. But as expected the LTVA is not suppressing the vibrations as much as the ATVA. Only at the minimum value it reaches similar value as the ATVA. In ideal conditions, without damping, the LTVA will completely suppress the vibration at the working frequency. For decreasing frequency the results do not seems to be as good as wanted when no gravity is included. There the LTVA has same behavior as for increasing frequency but the both ATVA has pretty similar behavior as each other and are not reactivated until reaching $8Hz$. When restarting at each frequency ATVA with gap is first active from 8Hz and over, while with an extra spring it get activated from 1Hz.



(a) *Increasing frequency.* (b) *Decreasing frequency.* (c) *Restart at every new frequency.*

Figure 4.4: *Comparison of the frequency response for the system without gravity for three cases. The LTVA is only in (a) because it gives equal results for the other cases as well.*

When one include the gravity the results for each case are very similar. Why the behavior is uniform for all three cases is probably because of the gravity makes m_a get in contact with m_m and by that become affected of the excitation force much easier. As one can see in Figure 4.5 m_a starts to move for all system already from 1Hz but for the first frequencies they are following the linear behavior. ATVA with gap is leaving the linear behavior at 8Hz and is then starting to suppress the vibrations more effective. While the one with an extra spring starts to suppress the vibrations much earlier. And for frequencies up to 40Hz they behave almost the same, but for higher frequencies m_a is stopping in different frequencies for each case. But because they are failing in much higher frequencies than the working frequency, it is not of interest.

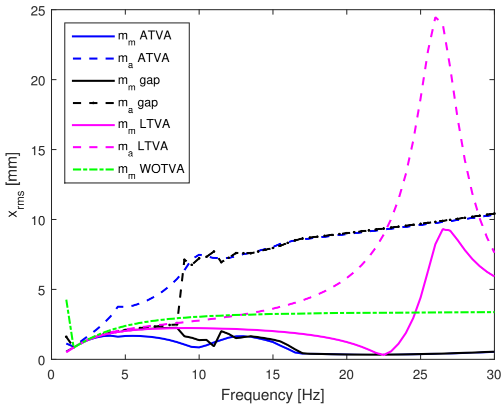
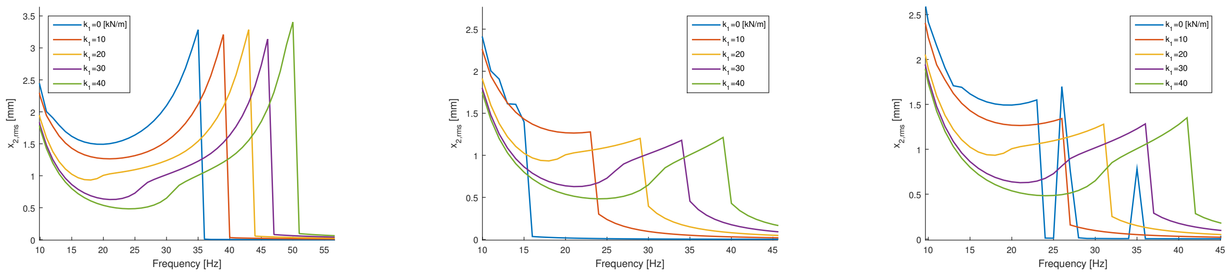


Figure 4.5: *Comparison of the frequency response for the system with gravity. In this graph the frequency is increasing, but the result with decreasing and restarting frequencies are very similar when looking at these frequencies. For frequencies over 40Hz they start to differ.*

4.2.2 Test rig

First off all it was checked if one could activate the ATVA with an extra spring even for the test rig. This seemed to be possible but it got a very linear behavior and to not get that also the length α was decreased from $3.25mm$ to $1.1mm$. This because when adding a spring the relative displacement of m_a did not reach over $3mm$ so the second spring never got in contact with m_a and it became a LTVA. But with decreasing the distance they got in contact and the nonlinear behavior could be observed. The first purpose was to have $\alpha = 1.0mm$ but when preparing the test rig for experiment it became $1.1mm$ which was good enough.

When to choose the stiffness of the first spring a wide range of stiffnesses was included. In Figure 4.6 one can see some results of the m_a when k_1 is varying. The working frequency for the test rig is set to be around $25Hz$ so when deciding on stiffness for the first spring m_a needs to be activated over $25Hz$. When looking at the results a stiffness around $20kN/m$ seemed to be a good choice. Because there are also experiments done on the test rig one needed to find a spring in proper dimensions that can be implemented in the rig. The final spring stiffness became: $k_1 = 17.8kN/m$.



(a) *Increasing frequency-* (b) *Decreasing frequency.* (c) *Restart at every new frequency.*

Figure 4.6: *Here one can see how the stiffness of the first spring, k_1, is affecting the behavior of the auxiliary mass. As one can see a higher stiffness makes it able to reach higher frequencies before it fails and stops suppressing the vibrations.*

One challenge with the test rig is to estimate the excitation force correctly. In Figure 4.7 one can see three different estimations that have been used. The blue linear line is how Swerea IVF had estimated the force before the project. But from measurements Swerea IVF did before this project one could see the behavior that the red double doted line shows. This was the estimation which where used for the early simulations when investigating the spring stiffness. After the experiments one could see that the force was like the third line and it can also be seen in Figure 4.11. This estimation were then used for the rest of the simulations, when comparing with other results and systems.

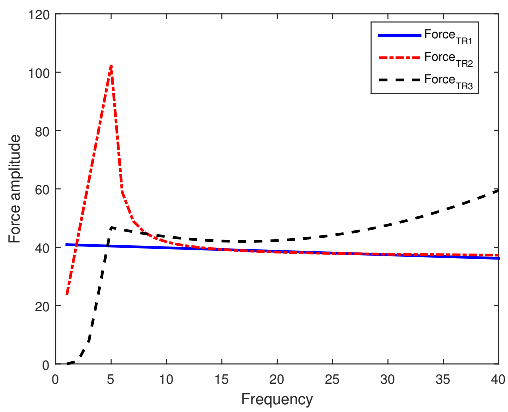
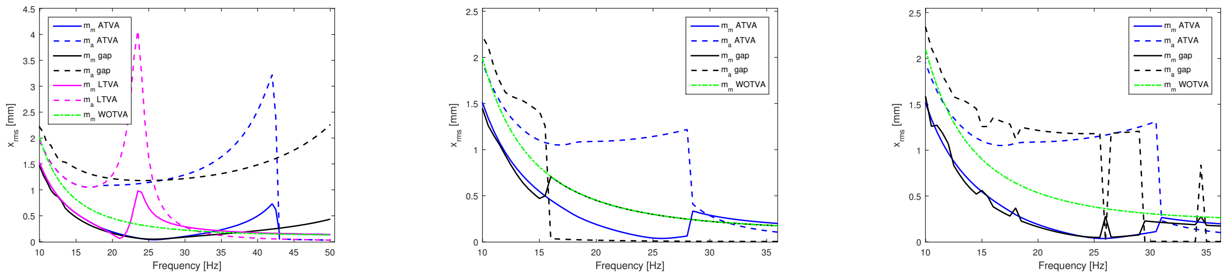


Figure 4.7: *Force development through the project. Comparison of older and the new estimations.*

The improvements of adding k_1 to the system can be seen in Figure 4.8, be aware of that the values on the axles in the plots are different. The low frequencies, $1 - 10Hz$, have very similar behavior for all cases. That is why they are not showed in the graphs but to see the extended versions of the graphs, from 1Hz, one can see them in Appendix B. If one look at the m_m without m_a one can see that the vibration amplitude is decreasing with increasing frequency. This is because of the excitation force is close to constant for higher frequencies and that depends on that the piston amplitude is constant. The goal is then to be under that line as much as possible, especially around the working frequency. Here the system with a LTVA is not adapted after the same working frequency than the others. It is linear with $k_a = 17.8kN/m$, the same as k_1 for the ATVA. So here it shows how much of the linear behavior the ATVA have. And for all three cases they behave the same up to around $17Hz$, after that the LTVA gets a resonance while the ATVA is still decreasing and relative slowly turning around. It is as first at $34Hz$ as the vibration amplitude become higher for the ATVA-system than without m_a. With a gap the vibrations gets worse at $37Hz$, but here $k_2 = 132kN/m$ to get same minimum value as the one with an extra spring. For the total vibrations one can say that it is good that the ATVA fails after $37Hz$ because then the system without TVA has lower vibration levels.

When comparing the ATVA with k_1 and the ATVA with a gap one can see an improvement. With an extra spring m_a is activated for more frequencies in two of three cases and are also more stable in the restarting case. In the case with increasing frequency both get activated but the one with a gap can go up to $56Hz$ before it fails and with an extra spring only up to $42Hz$. But it still higher than the working frequency which is enough. But as mentioned and which can be seen in Figure 4.8 the ATVA with an extra spring has a more stable behavior and are being activated up to at least $28Hz$ for each case. In total this means that the ATVA with k_1 gives a better result than the one with a gap.



(a) *Increasing frequency.* (b) *Decreasing frequency.* (c) *Restart at every new frequency.*

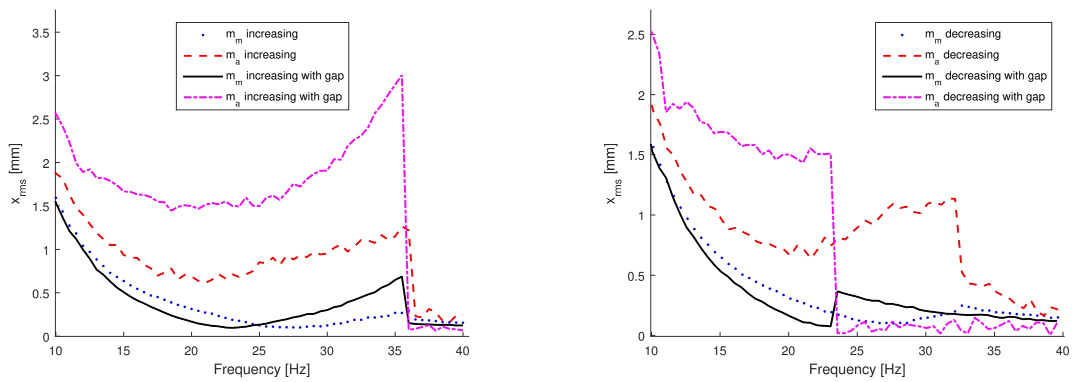
Figure 4.8: *A comparison of the frequency response for different systems in three different cases for the test rig.*

Experiments

The experiments are done as described in the methodology, section 3.4. In Figure 4.9 one can see the measured displacements for two cases and two different ATVA-systems. For the case with increasing frequency one can see that both systems are failing at the same frequency, right after $35Hz$. But for the case when decreasing the frequency the system with k_1 reactivates at 32Hz and with a gap is first reactivated at 23Hz. This shows that the system with an extra spring is more stable than with a gap.

One can also see that after the ATVA has stopped the vibrations get lower. This is because m_m starts to behave as m_a did not exist and as shown in the simulations the vibrations decreases when the frequency is increasing. Same as for the simulations it is good that the ATVA fails at $35Hz$ because for higher frequencies the vibrations get worse if it continues to be active.

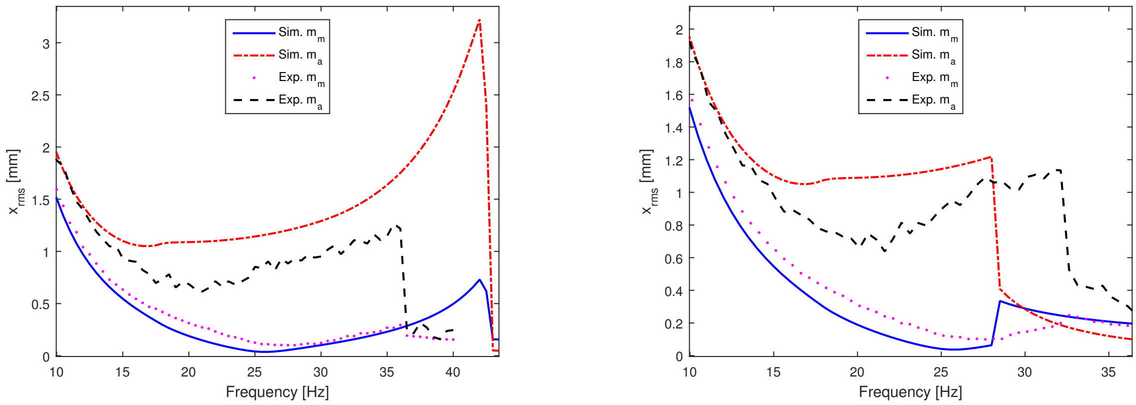
If m_a would stop for some reason at the frequencies between the reactivated frequency and the failing frequency, in this case $32Hz - 35Hz$, it can be activated again with a push. Over 35Hz it is not possible, this shows that if one just get enough movement on m_a it can be active. It is also very hard or impossible to stop the ATVA for lower frequencies then $32Hz$, which shows a stable running behavior.



(a) *Increasing frequency.* (b) *Decreasing frequency.*

Figure 4.9: *Calculated RMS values from measured data from experiment for two different kinds of runs. Also comparison between different cases. Minimum RMS value measured from the experiments with gap is* 0.0961*mm at 23Hz and with* k_1 *it is* 0.1101*mm at 28Hz.*

When comparing the results from the experiments with simulated results on get Figure 4.10, where one can see a pretty similar behavior. The biggest difference is the fail and reactivate frequency. This is probably because the conditions are more ideal in the simulations and parameters like damping can change from time to time in the experiments. And that are a parameter that have been shown to make a difference when it comes to this small changes. It can also be that α can vary between the springs and sides. As one can see in Figure 3.1 it is four α that needs to be tuned correctly and those can have some variation which can affect the final results.



(a) *Increasing frequency.* (b) *Decreasing frequency.*

Figure 4.10: *A comparison of the frequency response from simulations and experiments. The different figures shows different start conditions.*

The excitation force measurements can be seen in Figure 4.11. Here one can see that the force has a maximum at 6Hz which is the resonance frequency for the modelled user spring, k_p. After that it is decreasing and around $17Hz$ it starts to increase slowly till m_a fails and then it become constant around $36N$ for ATVA with extra spring and $40N$ when using a gap. This jump when it fails make it hard to anticipate how much the force will increase if not failing that early. But a good thing that the force become lower and constant when failing is that the total vibration will also decrease. One can also see that the force become greater for when using a gap, probably because m_a get a more distinct turn when changing direction. With an extra spring the bounce is over a longer time, due to the constant connection with the springs, which make the force to be applied for a longer time but is not reaching as high amplitude.

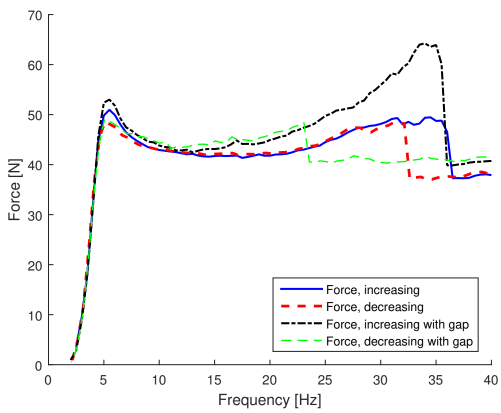
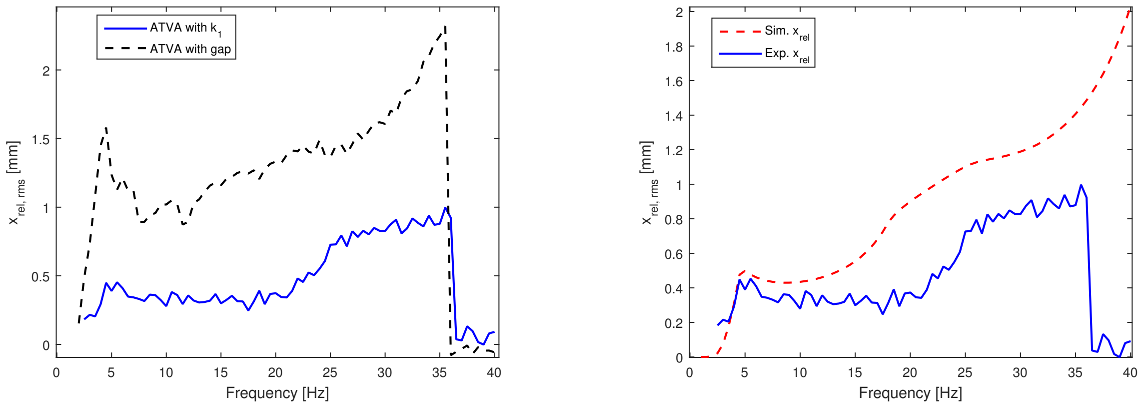


Figure 4.11: *Measurement of the force from experiments.*

When looking at the relative displacement of m_a to m_m one can see how much the springs, k_a, contract. As one can see in Figure 4.12 the RMS value for x_{rel} is below $1mm$ but as one know the RMS value is lower than the real peaks (approximately 29.6% lower). Frequencies lower than 23Hz is not reaching the second spring which make the behavior linear, but for frequencies over 23Hz it can reach the second spring and will then get the nonlinear behavior. Also the second spring makes it more stiff the increase of the displacement stops for some frequencies at least. In even higher frequencies the force become greater and it will lead to an increase in the displacement, see the simulated line in Figure 4.12b. One can also see that the ATVA with a gap has higher x_{rel} displacement and that is because it can move freely in the gap, while k_1 is preventing that free movement in the other case.



(a) *Comparison of x_{rel} between an ATVA with an extra spring and with a gap.*

(b) *Comparison of the simulated results and data from the experiment.*

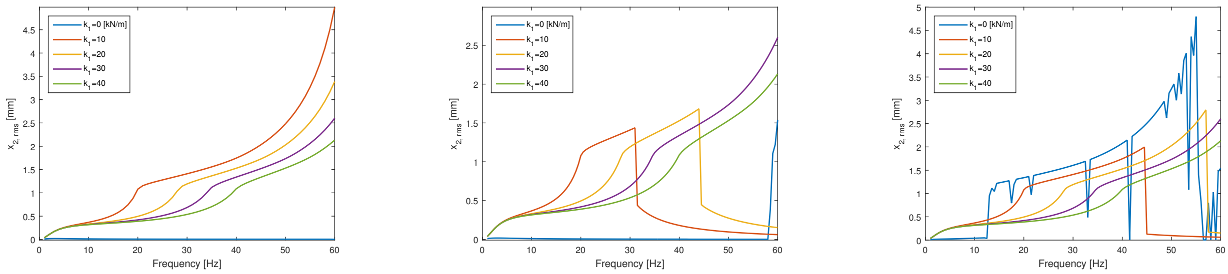
Figure 4.12: *RMS value of the displacement for m_a relative to m_m (x_{rel}).*

4.2.3 Reciprocating saw

The reciprocating saw has as shown similar simplified system as the other systems. The largest difference is that it is driven by an electrical motor and that will produce vibrations in all directions and not only in one as for the other systems. But the main vibrations seem to be the ones coming from the reciprocating movement of the saw and that movement should be possible to suppress with the ATVA. These other vibrations can also affect the ATVA and disturb its movements and by that get more unstable and have easier to stop. Also the low mass of m_a only $0.5kg$ makes it more sensitive.

Figure 4.13 shows the behavior of m_a with different stiffnesses on k_1. As expected it shows that a pretty stiff spring needs to be used to keep the ATVA active over the working frequency, which is around $45 - 50Hz$. With an increasing frequency one seems to manage to keep it active over the working frequency by adding a spring. When decreasing the frequency one need at least a stiffness around $30kN/m$ to reach satisfaction and it seems to also work for the restarting case. As one later can see in Figure 4.14 the behavior of system are very similar with and without gravity when using as stiff springs as one do here. That is why no further investigation of which spring stiffness suits best in case of gravity is presented.

After some more close range investigations it is decided to use $k_1 = 25kN/m$. This because it seems to pass the requirements good enough and is also starting to suppress the vibration in a lower frequency than stiffer springs would. It was also a balance to be able to adapt the system after the working frequency. With $k_1 = 25kN/m$, $k_2 = 40kN/m$ and $\alpha = 1.2mm$ one could get the minimum for m_m at $50Hz$ which is in the upper bound of the working frequency. Earlier parameters were $k_1 = 0kN/m$, $k_2 = 72.1kN/m$ and $\alpha = 1.0mm$ so all three of the parameters were needed to be changed for this tool to still have same working frequency. It is also better to be closer to the upper bound of the working frequency because as one can see in Figure 4.14 the vibrations are fast increasing when going to higher frequencies over $50Hz$.



(a) *Increasing frequency.* (b) *Decreasing frequency.* (c) *Restart for every new frequency.*

Figure 4.13: *Comparison of the frequency response on m_a for different values on k_1 and cases. Here $k_2 = 72.1kN/m$ and $\alpha = 1.0mm$.*

A very good thing for this system is that it behaves similar either one have gravity or not. Which can be seen in Figure 4.14, this is also correct for the different running cases. They behave different from case to case but the gravity has no influence. This is probably because of the relative stiff spring system, k_a. Because of the stiff springs the affect from the gravity can be neglected. Due to that one has reached a more stable and controllable system. What is also achieved is a lower vibration level for m_m especially for the working frequencies, when comparing with the system without a TVA.

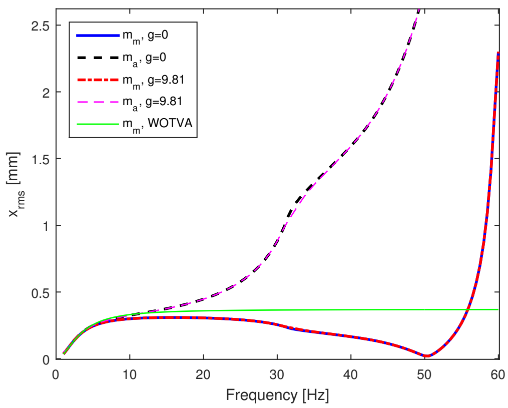
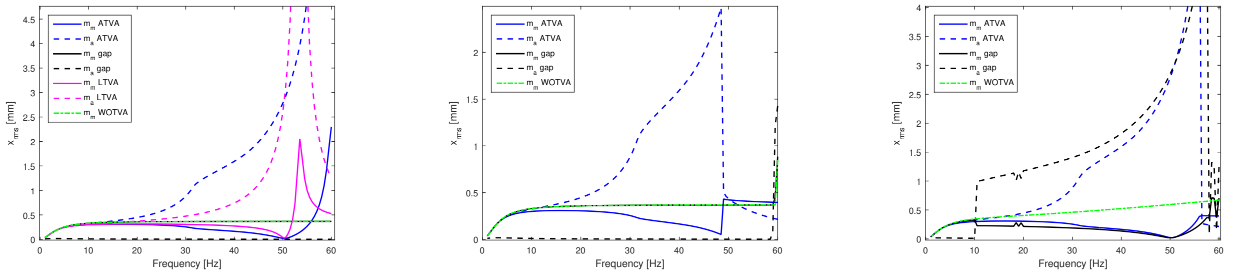


Figure 4.14: *Frequency response for the system with and without gravity. Here $k_1 = 25kN/m$, $k_2 = 40kN/m$ and $\alpha = 1.2mm$.*

When comparing with other systems with a LTVA, ATVA with a gap and without a TVA one can see an improvement especially in the stability to keep the ATVA active, see Figure 4.15 and 4.16. When comparing with a gap those are the old parameters are used for that system, which is $k_1 = 0kN/m$, $k_2 = 72.1kN/m$ and $\alpha = 1.0mm$, this because to really see the improvement. Also the LTVA is adapted to have the same working frequency, $50Hz$, which means that it has $k_a = 50kN/m$.

When looking at the frequency response for the different systems without gravity, see Figure 4.15, one can see that the system with a gap is not active at all for two of the cases. But interesting is that it seems to have ability to be active for some frequencies if one restarts from initial conditions. More discussion about how it can be this way can be found in section 4.3. The other systems like the LTVA behaves as predicted with a minimum at 50Hz and a maximum not far above. Without a TVA has a constant vibration amplitude over 10Hz, which make it easy to see if one get better results with other systems. The ATVA-system with an extra spring starts with a linear behavior but are for frequencies over 25Hz getting the nonlinear behavior and the vibrations get more suppressed. This shows that in case of no gravity it is an improvement to use the new parameters for the ATVA.

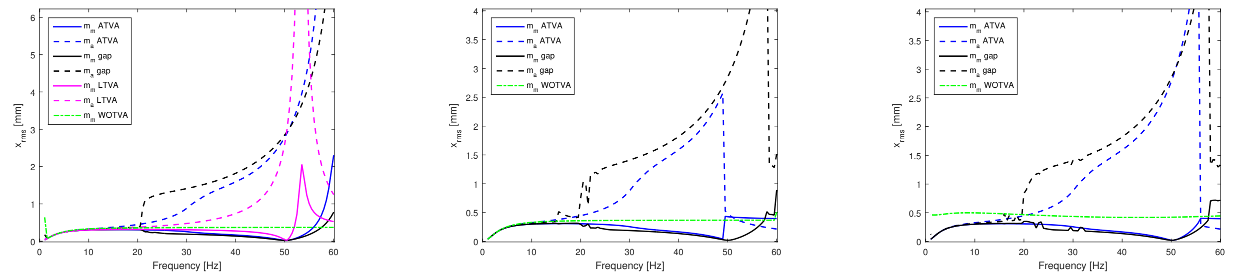
As one can see in Figure 4.15b and 4.16b the ATVA with an extra spring is not being reactivated until around $49Hz$ which is lower than the working frequency. But it needed to be this way to still have a minimum around $50Hz$. To have a higher reactivation frequency one need either stiffer springs or a smaller α but then it will also get a more linear behavior. To keep the nonlinear behavior and have a minimum at $50Hz$ it needed to be this way. But one can also see that for the other cases the failing frequency is higher than $50Hz$ which means that it can still have a chance to be reactivated in higher frequencies. For the case with increasing frequency the ATVA is failing when it reaches over $60Hz$.



(a) *Increasing frequency.* (b) *Decreasing frequency.* (c) *Restart for every new frequency.*

Figure 4.15: *Comparison of the frequency response for the system without gravity for three cases. The LTVA is only in (a) because it give equal results for the other cases as well.*

When comparing the different systems with the affect of gravity included one can see that all masses starts moving from $1Hz$ but up to 20Hz they move more or less as a fixed body. This behavior has also been seen for the pneumatic breaker and are probably depending on that the excitation force is not strong enough to give m_a enough energy to leave that phase. But with higher frequency the force also increases and then the phase shift can be done. One can see when the ATVA starts to move for itself that the one with a gap is suppressing the vibration little more and for a wider range than the ATVA with an extra spring. It is more slowly suppressing the vibration but are also doing a good job, especially compared to the LTVA and WOTVA. The ATVA with gap has also a higher failing frequency, after 63Hz, and also a higher reactivation frequency around 57Hz. Same for the restarting case but there it is little more unstable for some frequencies. The one with an extra spring is acting, as written before, very similar as without gravity. For this case with gravity included one can say that the ATVA-system with a gap is the best solution. But the ATVA with an extra spring is not far behind, but get a more linear behavior due to the stiff springs.



(a) *Increasing frequency.* (b) *Decreasing frequency.* (c) *Restart for every new frequency.*

Figure 4.16: *Comparison of the frequency response for the system with gravity for three cases. The LTVA is only in (a) because it give equal results for the other cases as well.*

To do a conclusion of what ATVA-system is best for the reciprocating saw used for all cases it will be the ATVA with an extra spring. Because it is performing so much better for the case without gravity and are very close to even beat the other system when gravity is affecting. The reciprocating saw is also often used in both horizontal and vertical directions which makes it important that the ATVA can be kept active even if changing work direction.

4.3 General discussion

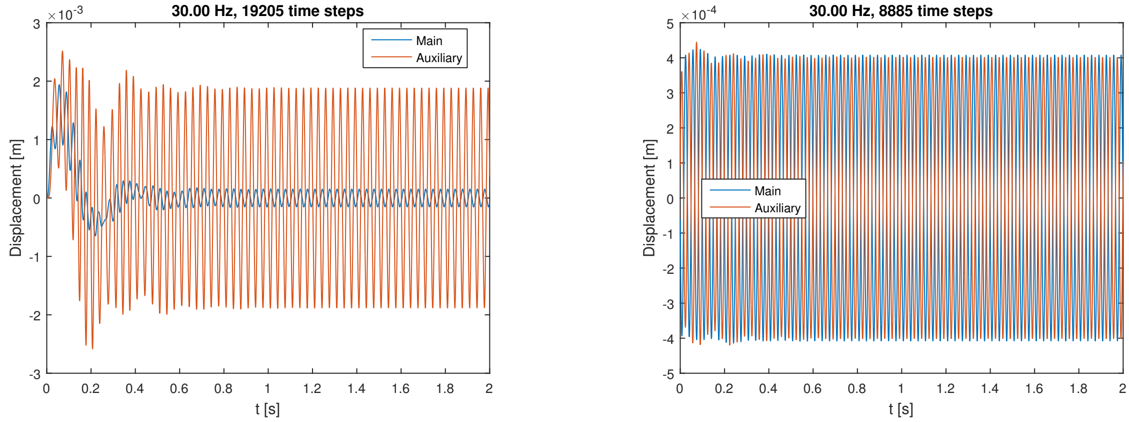
One of the main problems with keeping the ATVA activated is that it is stopping at high frequencies. It has been shown that it is possible to get it activated for low frequencies but next step is to keep it active. Some tries have been done during this project but only to reach good enough results for just that system. An investigation of why it stops for high frequencies is needed to be able to have a clear explanation of why this phenomenon appears. A hypothesis about why this happens is that when reaching high frequencies m_a needs to move faster and the momentum is increasing and at some point the system cannot keep pace of the required movement and is then starting to counteract the movement for m_a. Which make the ATVA to have close to no vibrations while the rest of the system loses the affect of having an absorber.

The behavior of the ATVA when it has failed and one is lowering the frequency it reactivates often at a lower frequency than it was failing at. This depend probably on the momentum, it is not possible to reactivate at so high frequency because the ATVA-system is too slow to start at so high speed. One need to lower the frequency more to get it started, but with a push it can be reactivated in a higher frequency and that is probably because one are then helping the system pass the threshold that the momentum needs. As a conclusion it is shown that it is easier to keep m_a active when it is already moving than to reactivate from zero.

If one look at the behavior of a LTVA it has the same behavior for the three cases (increasing, decreasing and restarting frequency). Then one can ask why it is not that way for the ATVA, it is still the same setup of springs for the three cases but the behavior differ. It is shown in other papers that it is pretty typical for a nonlinearity to behave different when running it different ways. For the systems in this project one can see that for low frequencies, approximately $1 - 10Hz$, it is only acting with the first spring and gets than a linear behavior while for frequencies around and over the working frequency m_a has contact with both springs and is then getting the nonlinear behavior. Then when decreasing the frequency after failing the system need to jump direct into the nonlinearity, which maybe is harder than start as linear and then go to nonlinear. In some cases it can also depend on the first spring, which is weaker than the second one, can start acting as an isolator in high frequencies and are then isolating the small movements when it tries to reactivate the ATVA so it will never get enough displacement to get affected of the second spring. But then when lowering the frequency even more the isolating behavior fades away and it starts acting as a TVA again.

Although the ATVA has different frequencies where it fails and become reactivated again the behavior of the system is the same as long the ATVA is active. It is no difference between the cases while it is active and that shows that the system has a stability and is acting the same as expected even if one need to decrease or increase the frequency while working with the tool.

One little strange thing one can see in the results is that the case when restarting from initial conditions for each frequency is often reaching a higher frequency before m_a can not be activated than the case with a decreasing frequency. Both is cases where one try to activate the ATVA from almost no movement to a frequency around or over the working frequency. They have pretty similar start conditions but are still given different results. This has probably to do with that the start conditions are not exactly the same. For the case with restarting at each frequency the system get some kind of kick-start where the whole tool is getting a displacement before it finds steady-state. With decreasing frequency the changes between the frequencies are smoother and the steady-state is found much faster. This can be seen in Figure 4.17, where the ATVA become active in the restarting case but not in the decreasing case. This difference in the start has probably to do with the momentum of m_a because when restarting from rest it follow the whole systems movement in the beginning, but for the decreasing case it has small movements already and is not following the whole systems change as much and are then not fully affected.



(a) *Restarting at* $30Hz$.

(b) *Decreasing the frequency to* $30Hz$.

Figure 4.17: *Comparison of when restarting and decreasing the frequency on the test rig. Be aware of the different values on the y-axle.*

As shown the ATVA seems to get a more stable behavior when adding the extra spring, while not affecting the suppression of the vibrations so much. For some system the range of great suppression has become little more narrow, but it still a wide range and much wider than for the LTVA. In the experiment the ATVA with a gap reach 5% lower frequency than the ATVA with k_1 but for many of the simulations of the other tools the ATVA with k_1 can reach up to 25% lower vibrations on the main mass than the ATVA with a gap can. Which is a nice improvement to a already good system. By adding the extra spring one get improvements in many ways and are not losing any good qualities that one had before.

Now it is shown that adding an extra spring helps the system to be activated. But what is the main parameters that affect m_a so it can be active? After doing all simulations and experiments it stood clear that it is three things that are important to keep the ATVA active. These are the vibration amplitude of the system and in particular the vibration amplitude of m_m, the force from the piston which affect m_m and the third thing is the frequency. For low frequencies are vibration amplitude and force more important. Because the system moves relative slowly in low frequency and then the force needs to be strong enough to get m_a to move and the amplitude large enough to get an initial displacement. The affect of the frequency is more visible in higher frequencies when it fails. Because then the vibration amplitude and force are not enough to keep m_a in pace. It is a balance of the three things, but one can say that the frequency is in command and the other two needs to adapt to keep it active. The amplitude needs to be high enough to make the whole system move and then the force needs to be large enough to move m_m and more significant also be transmitted to m_a so it can change phase and start to suppress the main vibrations. The first spring helps to transmit the force but can also counteract the phase shift, which means that it needs to be carefully tuned to be efficient.

4.4 Error sources

Things to have in mind when looking at the results are that all simulations are done with a simplified model of the system. It is verified to give reliable results, but it is still simplified and if one want to go more into details and investigate very small changes it can be a good thing to have a more detailed model. The simulations also only shows for ideal conditions if one would test it for real the result would probably be similar but not the same, for example see the results from the test rig.

Also when doing the experiments there are a few error sources. The sensors are maybe not exact perpendicular to the surfaces and are then not measuring exact correct displacement. But the biggest error found for the experiments was that the test rig was placed in a really dry and dusty environment which made the lube oil to change character pretty fast. The lube oil was placed on the bars where m_a was supposed to glide. The friction could differ between each day and gave small changes in the results. When doing the experiments it is important to have well greased equipment every time. With a well greased bar the results become closer to the simulated results.

Also the springs in the experiments are not fixed in the ends, only kept in place by rubber bands. This do that the springs can be bouncing around when losing contact with m_a and that can affect the results. Not a lot but still enough that one need to consider it.

5 Conclusion

To start with it is found that this area seems to be unexplored. When looking in the literature very little is done in this particular area, which shows that it is a pretty new area of research and there are a lot to discover about how NLTVA can be used in tools and machines.

One can say that to activate the ATVA the movement and the force from the main mass needs to be large enough so it get in contact with the auxiliary mass and affect it enough to start moving. And what is shown during the project is that adding an extra spring makes that m_a always has contact with m_m which fulfill that requirement and then only the transmitted force needs to be enough to get m_a moving. Which also shows that the first spring cannot be too weak otherwise the force would not transmit. It is also possible to make the gap smaller to make it work, but in some cases it needs to be so small that the system gets a very linear behavior which is not what one want. It makes it also easier if the masses, m_m and m_a, considering the weight are pretty similar. At least a quota of $1/4$ is desired.

The extra spring makes it also possible to activate the ATVA from very low frequencies and by keeping the springs stiff enough they are able to keep it activated over the working frequency. Which is good enough for these tools.

As a conclusion one can say that it is possible to make the ATVA to be activated for a wider range of frequencies by adding an extra spring instead of the gap. And it is in many systems also an improvement of the vibration suppression which is a bonus.

5.1 Future research

The next step for this research is to investigate more of why the ATVA stops when reaching a certain frequency. Because to solve the problem of keeping the ATVA activated one need to know the reason why it stops.

This project has also only had focus on three systems and what can be done to make these work better. For later research one need to investigate other tools, especially much larger and smaller tools like big breakers and small drills. To see if it is possible to implement the ATVA for that range of tools.

Also for the three systems that have been tested here it would be good to run an optimization code to generate better parameter values. Before one run the optimization one need to decide what to optimize. If it is the ability to keep the ATVA active for as much frequencies as possible, to get as low vibration levels as possible, a combination of them both, or something else.

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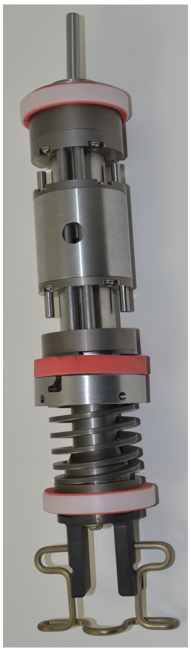
Appendices

A Pictures of the systems

(a) *The pneumatic breaker.*



(b) *The inside of the pneumatic breaker with ATVA.*

Figure A.1: *The pneumatic breaker and the inside with the ATVA.*

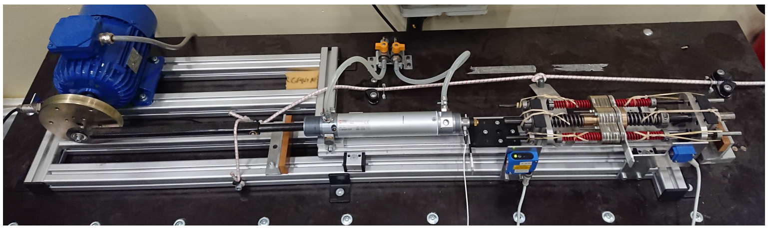


Figure A.2: *The test rig.*

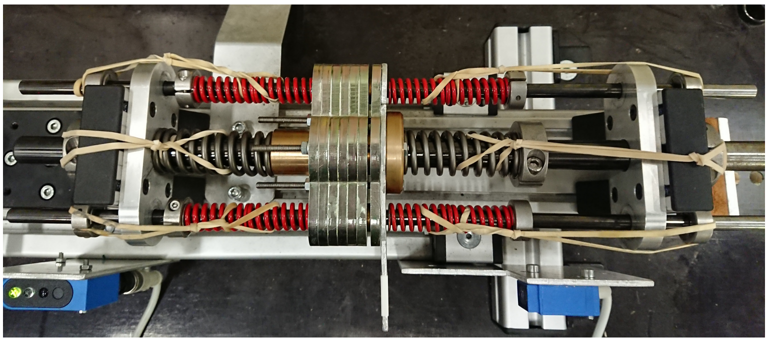


Figure A.3: *The ATVA on the test rig.*

Figure A.4: *The reciprocating saw.*

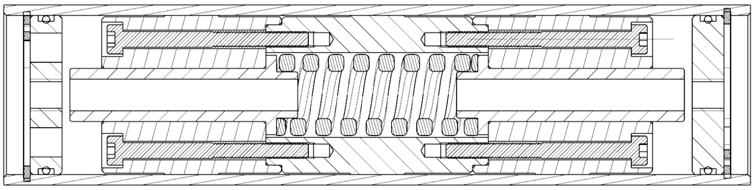


Figure A.5: *A drawing of the ATVA inside the reciprocating saw.*

B Complete graphs for the test rig

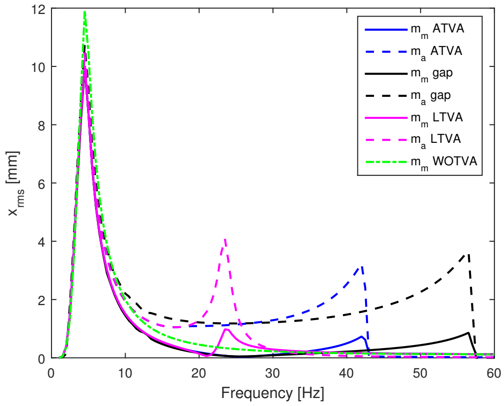


Figure B.1: *Comparison of the simulated results with increasing frequency for different TVA-systems.*

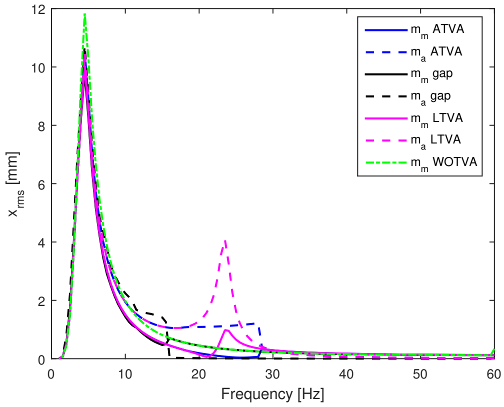


Figure B.2: *Comparison of the simulated results with decreasing frequency for different TVA-systems.*

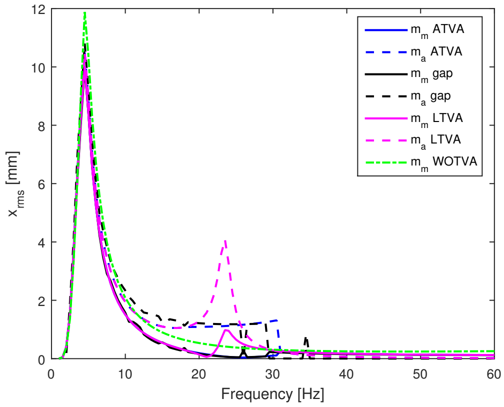


Figure B.3: *Comparison of the simulated results with restart at each frequency for different TVA-systems.*

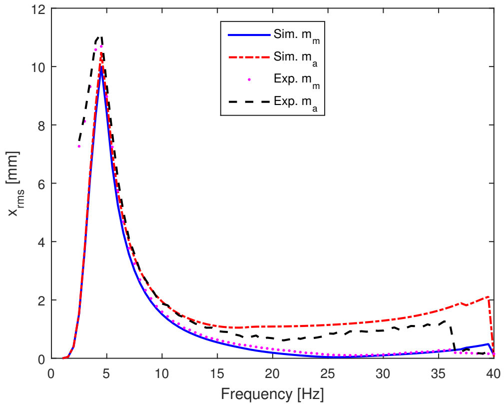


Figure B.4: *Comparison of the results from experiments and simulated results with increasing frequency.*

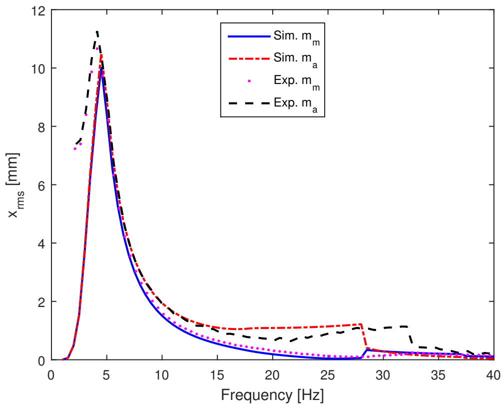


Figure B.5: *Comparison of the results from experiments and simulated results with decreasing frequency.*

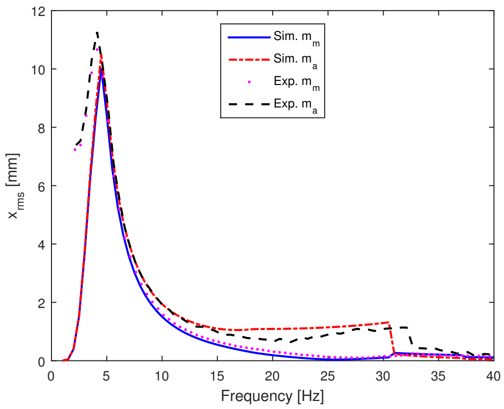


Figure B.6: *Comparison of the results from experiments with decreasing frequency and simulated results with restart at each frequency.*