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Experimental Investigation of Vibration Control

in the Husqvarna K760 Power Cutter

Master Thesis, Product Development

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Cover:
A K760 power cutter being used to cut a concrete water pipe (pp. 2-5).

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Abstract

This report describes the process and the result of a master thesis aimed to investigate the vibration control of a power cutter. The project was initiated by Husqvarna Construction AB and the K760 power cutter is the object that is investigated. The primary aim of the project was to investigate the transferring of vibrations from the vibrating part of the power cutter to the handle structure, across the three vibration isolating springs that separate these parts. By investigating the current situation understanding and knowledge was acquired and used as a foundation for attempting to predict how design changes in terms of the vibration control could affect the transferring in future designs.

Based on a pre-study a repeatable and reliable measuring method was established by designing a test rig to avoid distortion of results and using the measuring equipment provided by Husqvarna. Then data was collected by measuring the vibrations on both sides of the vibration isolating springs. These measurements were repeated for two additional spring sets, one weaker and one stiffer than the original, in order be able to relate the transfer to the spring stiffness. All data was then analysed to develop transfer functions describing the vibration control of the power cutter.

From these transfer functions "optimised" cases were developed for how the spring sets should be composed to most effectively isolate the vibrations. These were tested in the same manner as the previous sets and the results compared to the original.

When looking at the vibration isolators it turned out that the results from the "optimised" cases were not much different from the original, but a particular frequency was passed through to a greater extent in the "optimised" cases. This led to a significant increase in the amount of vibrations measured at the handles. The conclusion from this is that the attempted prediction approach is not sufficiently developed and that the total system of the power cutter is very complex and the handle structure also has to be investigated further. Conclusions could also be drawn regarding how the test rig should be designed and how to perform measurement.

Keywords: passive vibration control, experimental analysis, power cutter, concrete cutting, ergonomics

Preface

This report presents the results of a master thesis in the department of Product and Production Development at Chalmers University of Technology, Gothenburg, Sweden. The thesis is written as the final part of the masters' program Product Development. The project itself was requested by the Construction Products division at Husqvarna AB. The project consists primarily of experimental testing and analysis of one of the company's existing products, the K760 power cutter.

The master thesis has been written by two students, Andreas Börjesson and Martin Krogh, at the Product Development masters' program. Andreas has a bachelor in mechanical engineering and Martin has a bachelor in automation and mechatronics engineering.

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1
INTRODUCTION

1.1 Husqvarna

The company Husqvarna was founded in 1689 in the city of Huskvarna in Sweden, from which it also got its name. Originally Husqvarna was a weapon foundry but the company has since undergone several major shifts of the product line, having produced everything from sewing machines to motorcycles (see figure 1.1) in the past. While many of these past products are still being manufactured under license by other companies, Husqvarna is today a world leader in the garden and forestry equipment market, especially focusing on their line of chainsaws. (Husqvarna Group, 2013).

Over the past hundred years Husqvarna has acquired several competitors and added the acquired product lines to their own. One such acquisition was the company Partner in 1978, another Swedish chainsaw manufacturer, who at the time also manufactured so called power cutters for cutting in concrete. With this acquisition Husqvarna was able to enter into the construction product market, where they have now become one of the market leaders, specialising in products for cutting, surface finish and demolition, (Husqvarna Group, 2014).

Figure 1.1. An old Husqvarna motorcycle at display in the Husqvarna Industrial Museum (Husqvarna Fabriksmuseum, 2014)

In 2013 Husqvarna had a turnover of SEK 30.3 billions. Their primary market is Europe with about 45% of the sales, closely followed by Northern America with 42%, the remaining 13% is from the rest of the world. The construction products account for about 10% of the group's turnover globally. They state their main competitors overall to be Stihl and for the construction products market the competitors are primarily Stihl, Hilti and Tyrolit. Husqvarna has factories in 11 different countries around the globe, including China and the U.S. The company has about 14,000 employees and it is based in Huskvarna where the core manufacturing is located. (Husqvarna Group, 2013)

1.2 Background

Multi-storey apartment and office buildings are often built using concrete blocks. When building with concrete blocks it is common to cut holes for HVAC[1], doors, windows and staircases after the blocks are in place (see figure 1.2). This eases the process of placing the blocks and the installations can be performed with higher accuracy and speed. A common method for cutting in stone and concrete is to use a so called power cutter, which is a handheld machine with a circular blade having diamond coated teeth. These machines need to be high powered yet light and commonly use an air cooled single cylinder two-stroke gasoline engine.

There are also hydraulic and electric power cutters which then require an external power source. A challenge with the design of lightweight and high powered machines such as a power cutter is the emergence of vibrations, which are known to increase fatigue wear of structures and may ultimately lead to fractures (Buscarello, 2002). Long term exposure to vibrations also poses a threat to the operator's health.

Figure 1.2. A concrete cutting professional using a K760 to cut a water pipe
(Husqvarna Construction AB, 2014 A)

[1] Heating, ventilation and air conditioning

In a gasoline power cutter vibrations are created primarily due to:

- The rotating and translating mass movement in the two stroke engine
- The power train
- Unbalance in the rotating cutting blade

As the engine is accelerating the frequency and the amplitude of the vibrations in the machine increases. At certain frequencies the different parts of the machine will encounter their natural frequencies, specific frequencies at which they come into self-pulsation, or resonance. An object has one natural frequency for each degree of freedom and since a machine is an assembly of many parts there are many natural frequencies that may be necessary to take into consideration. If operating at a natural frequency the amplitude of the oscillation may become 5-10 times higher than normal (Buscarello, 2002) and cause critical damage to the machine and the operator. Therefore it is of great importance that the work, as well as the idle rpm does not coincide with a natural frequency of the structure. When sweeping from idle to work rpm it will very likely pass a natural frequency, but if passing through rapidly enough it will not start self-pulsating.

As with any professional tool there are legislation and norms regarding the acceptable levels of vibrations in a power cutter in combination with how long they are allowed to be used without breaks (Arbetsmiljöverket, 2005). These laws and norms are important as long continuous exposure of hand and arm vibrations are well known to cause health issues, both short and long term, such as carpal tunnel syndrome and white fingers (Akselsson et al. 2010). The vibrations are most dangerous for the human body between about 8-16 Hz and therefore a so called hand-arm filter is often used to weight the acceleration output from measurements for different frequencies to get a better picture of how exposed an operator of the product really is (Canadian Centre of Occupational Health and Safety, 2008).

Husqvarna AB's smallest gasoline powered cutter, the K760 (see figure 1.3), have reached the lowest vibration level according to Swedish law with 2.4 ms^{-2} (Husqvarna Construction AB, 2013), the limit being 2.5 ms^{-2} (Arbetsmiljöverket, 2005). This means that the machines are allowed to be used continuously throughout an eight hour workday. The legal tests are performed with the machine being handheld with a simulated load which results in measurements with high level of uncertainty. Because of this the effects of small design changes done to the power cutter may be hard to detect and there is even a risk that the vibration level is increased if performing small changes to the power cutter. There are however also problems with suspending the machine when measuring vibrations as there is a risk that the suspension mechanism adds too much mass and rigidity, dampening out or intensifying the vibrations.

Figure 1.3. A Husqvarna K760 power cutter
(Husqvarna Construction AB, 2014 B)

In order to reduce the vibrations that are transferred to the operator the cutter is split into two main parts (see figure 1.4); one containing the engine and cutting blade etc. and the other being the handles, tank and carburettor, with three springs acting as vibration isolators installed between the main parts.

This type of solution for vibration control is the most common within the industry, often used in combination with rubber buffers. To absorb the vibrations effectively the springs should be as weak as possible, but if they are made too weak it will be difficult for the operator to control the machine. If the springs are weak there is also an increased risk that they become fully compressed which would result in complete transfer of the vibrations to the handles.

Figure 1.4. The power cutter divided into its main parts.

Husqvarna is currently working to model the power cutter in a simulation program. Simulation offers a relatively cheap way to test and evaluate design changes without the need for expensive prototypes being manufactured. As the power cutter is a quite complex machine with many components and a lot of non-linear behaviour it is however a time consuming task to model it. It is also difficult to completely capture the behaviour of all parts and their interactions in the cutter in the simulation software and there is a great need for calibration between the model and the actual machine. (Fälth, 2014)

The low level of vibrations that have been achieved in the K760, in combination with the relatively large uncertainty of the measurement system, has caused Husqvarna to be interested in mapping the vibration levels and how the vibrations are transferred between the two main parts of the machine. This is intended to provide a better understanding and a broader knowledge base for if and how to modify the vibrations isolators as well as other parts of the power cutter.

1.3 Purpose

In this project the vibrations that occur in a power cutter will be investigated. Primarily the investigation will focus on how much of the vibrations that are transferred across the vibration isolating springs from the engine part to the handles.

The purpose of the project is to develop a better understanding of how the vibrations are transferred on today's cutters as well as developing knowledge that makes it possible to more accurately predict or more easily obtain data of the vibrations and the transferring in future designs.

1.4 Goals

- Find a reproducible way of performing the measurements of vibrations that closely resembles handheld testing
- Provide a data and knowledge base for future development work focused on the vibrations control in the K760 power cutter
- Develop and test suggestions for ways to reduce the handle vibrations by making predictable improvements to the vibrations isolators according to the gathered information
- Map the relationship between transferring of vibrations and the stiffness of the springs

1.5 Research Questions

- What is a reliable and sufficiently accurate way to measure the vibrations in a power cutter that allows the detection of effects of minor changes to the structure?
- What are the causal relations of the transferring of vibration from the cutting part of the cutter to the handles?
- Are there any natural frequencies of the power cutter that needs to be taken into consideration?
- How can the vibrations isolators be designed to further limit the felt vibrations of the power cutter without sacrificing the controllability of the machine?

1.6 Deliverables

- A methodology for reproducible and repeatable measuring of vibrations in the K760
- An analysis of the transferring of vibrations through the vibrations isolators
- A causal relationship describing the transfer of vibrations through the isolators
- An analysis of the origins of the vibrations in the K760
- Suggestions for changes to the vibration isolators

1.7 Delimitations

The project is only concerned with testing the power cutter in its current production variant with a variety of different types of vibrations isolators installed. The vibrations that are transferred across the vibration isolating springs are being the focus of the investigation rather than handle vibrations. Measuring handle vibrations may be used as a verification method for proposed changes but is not of primary concern. The measured vibrations will not be passed through a hand-arm filter as it is not specifically the human vibration exposure that is being researched.

The projected is not aimed to develop any actual prototype or perform changes to the current power cutter but rather the aim is to acquire data and knowledge to guide future development.

The project will not be focused on investigating the structural dynamics of the power cutter but will only focus on the vibrations isolators. Each vibrations isolating spring will be viewed as a separate and simplified system.

Two MSc students will work full time for approximately four and half months with the project. Both Chalmers and Husqvarna Construction Products in Jonsered provide supervision during this time as well as limited support in terms of knowledge and information. The tools and equipment supplied by Husqvarna Construction Products will primarily be used. Limited support will also be provided by the department of physical measurements of Husqvarna AB in Huskvarna.

1.8 Outline of Report

The report will start off by describing the state of the art of vibrations measurement and vibrations control in chapter 2, introducing measurement methods, ergonomics, general vibration control methods and the solutions used by Husqvarna and their competitors as well as other products where vibration control is necessary.

Then the research process is discussed in chapter 3, providing an overview of the steps taken to answer the research questions. Following this, in chapter 4, there is a more detailed description of how the measurement system has been designed in the specific case of this thesis work. This chapter includes descriptions of how the equipment was placed and used, how the tests were run and how the data was analysed.

The results are presented in chapter 5, after the measurement system design chapter. Here relevant graphs and data tables are shown and described. These results are then discussed in chapter 6, which also includes the answers to the research questions and a discussion on whether or not the thesis goals have been fulfilled. Possible sources of error in the measurements are also discussed.

Finally conclusions drawn from the results and the discussion will be presented in chapter 7, followed by recommendations in chapter 8. More in depth descriptions on what preparations were necessary will be presented in appendix A, B and C. The bulk of the result graphs are presented in appendix D and in appendix E the input vibrations are compared.

2
STATE OF THE ART

The following chapter will introduce theory that is related to the project. It will also discuss the test object, the K760 power cutter, as well as the field of vibration control, both in power cutters and other products.

2.1Vibrations

Mechanical vibrations can be described as a part or component in a mechanical system oscillating around an equilibrium point. In order for the vibrations to be initiated there must be some excitement of the system, usually from an engine or external forces such as a car moving over an uneven road. There are many factors that can produce or increase vibrations in a mechanical system when there is an exciting force such as reciprocating movement, imbalances in rotating parts, non-linear friction and gearing. (Brüel & Kjaer, 1982)

In all systems there is also some degree of inherent damping of vibrations. Damping means that the energy of the vibration is dissipated as heat or noise. Different materials provide different levels of damping, for example rubber and foams have high damping while spring steel has very low. Because all materials have some degree of damping the vibrations in a system will die out over time once the exciting forces are removed. (Rogers Corporation, 2012)

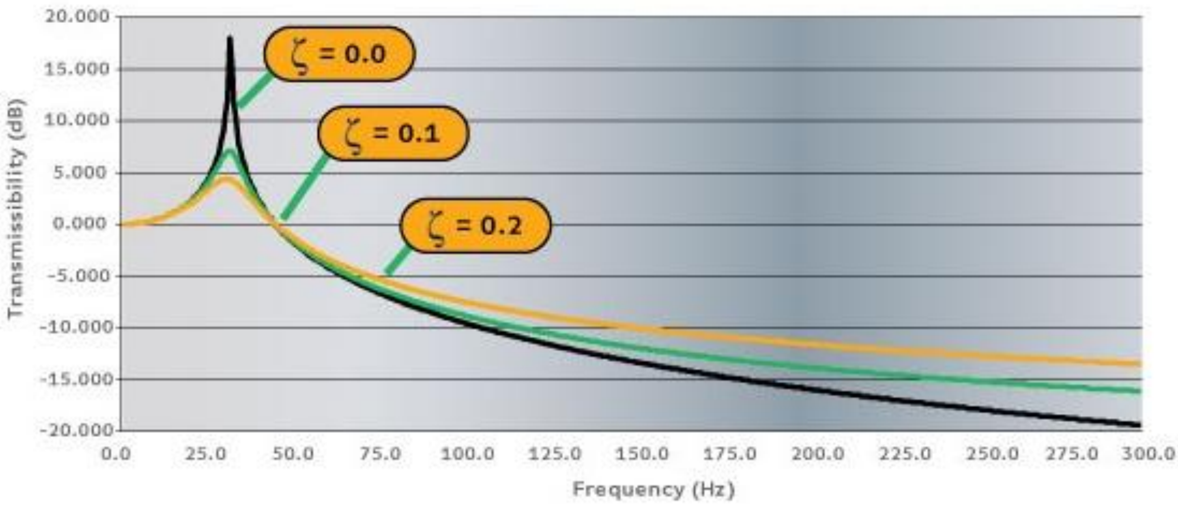

Figure 2.1a. Different damping and increased damping affect vibrations.
The decibel scale is converted according to dB = 10log (output vibration/input vibration).
(Rogers Corporation, 2014)

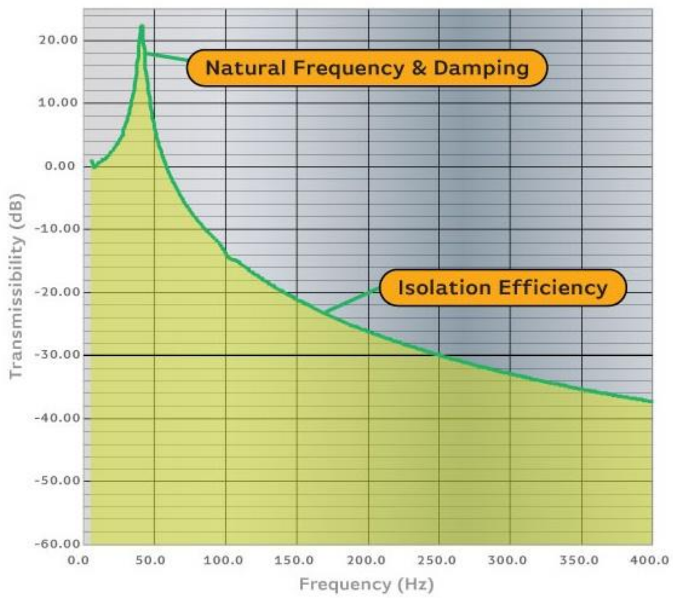

Figure 2.1b General material vibration behaviour.
See figure 2.1a for explanation for dB-scale.
(Rogers Corporation, 2014)

When a vibrating system is left to settle like this it will vibrate at its natural frequency. If the exciting force is added with a periodicity that matches the natural frequency of the system close enough, the vibrations will be amplified. This phenomenon is known as resonance and is important to consider in any system where vibrations are present as unwanted resonance can have dire consequences. (Buscarello, 2002)

In some cases vibrations are wanted in a system, such as in loudspeakers, but in a vast majority of cases they are a problem or at least not desirable due to their negative effect on longevity of parts and products. As with all cyclic loads mechanical vibrations causes fatigue wear and can lead to failures such as cracks or fractures (Buscarello, 2002). This is primarily a concern in systems with very high vibrations levels or vibrations in combination with high mechanical loads and of course with systems where a mechanical failure can have catastrophic consequences for people or property.

2.2 Measuring Vibrations

Vibrations are generally measured in acceleration, usually ms^{-2}, and are measured with a sensor known as an accelerometer (see figure 2.2) in combination with some form of data analysing equipment (Brüel & Kjaer, 1982). There are several different types of accelerometers with different characteristics, advantages and drawbacks, and they can be used for a variety of applications (National Instruments, 2013). One common type for measuring vibrations is a so called piezoelectric accelerometer, which has the best all around characteristics for this application with large frequency and amplitude range.

The core in the piezoelectric accelerometer consist of a piezoelectric material which implies that when the crystals in the material are deformed by tension or compression the mechanical work transforms into electricity. The generated charge is proportional to the force applied from which it is possible to obtain the acceleration (Brüel & Kjaer 1982).

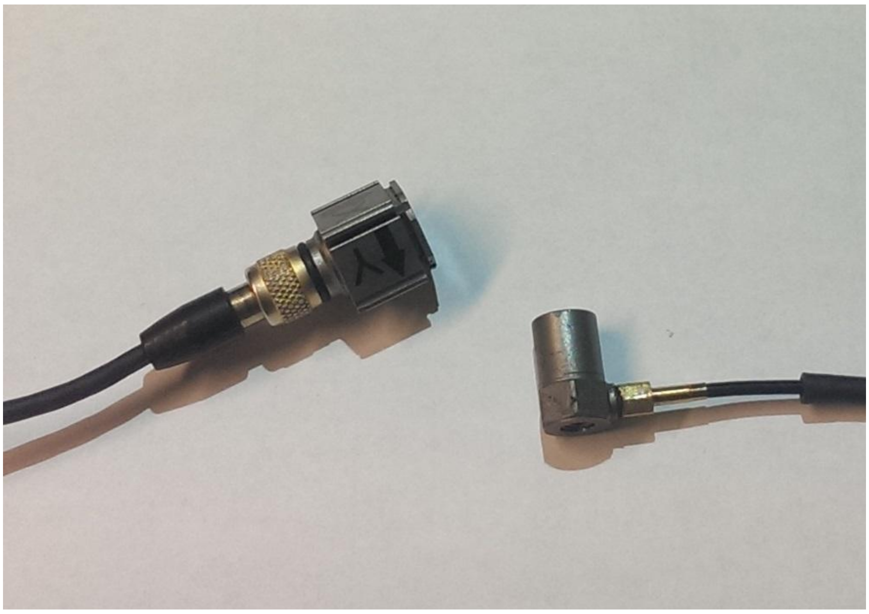

Figure 2.2. A three-axis accelerometer (left) and a single-axis accelerometer (right).

A data analyser is used to transform the measured acceleration over time into the frequency domain using an FFT algorithm in order to simplify the interpretation of the data, separating the different frequencies of vibrations present. A tachometer counts revolutions per unit time and is used to relate the measured vibrations to certain rpms of the working machine. This is done by sorting the vibrations according to their frequency multiple of the engine rpm, or so called order. The engine rpm frequency corresponds to the 1:st order, and in the case of the K760 the blade, which has a gear ratio of 2:1 from the engine, turns at half the rpm, the 0.5:th order. There are many different ways to measure revolutions in different situations. In the case of a two-stroke internal combustion engine a simple way to measure the engine rpm is to detect the cycles of the ignition magnet on the flywheel with an inductive sensor (see figure 2.3).

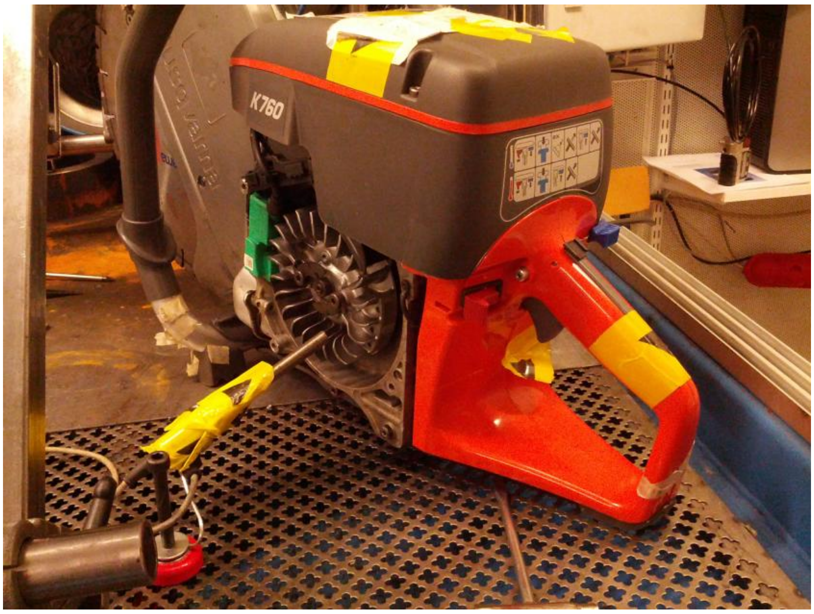

Figure 2.3. A tachometer placed close to the flywheel.

When measuring vibrations that affect an operator and considering the ergonomic and health aspects of vibrations there are certain rules and standards to use. For different types of machinery there are different standards for how to place the accelerometer and how the machine should be loaded. The results from the measurements are also usually passed through a special weighting filter that takes into account how dangerous different frequencies are to the human body, one such filter is the hand-arm filter (ISO 5349) which focuses on frequencies around 8-16 Hz (see figure 2.4). This filter assigns a gain of 1 to the most harmful frequencies and lower gain to the surrounding frequencies, with gain decreasing the further from the focus frequencies you get. (Canadian Centre of Occupational Health and Safety, 2008)

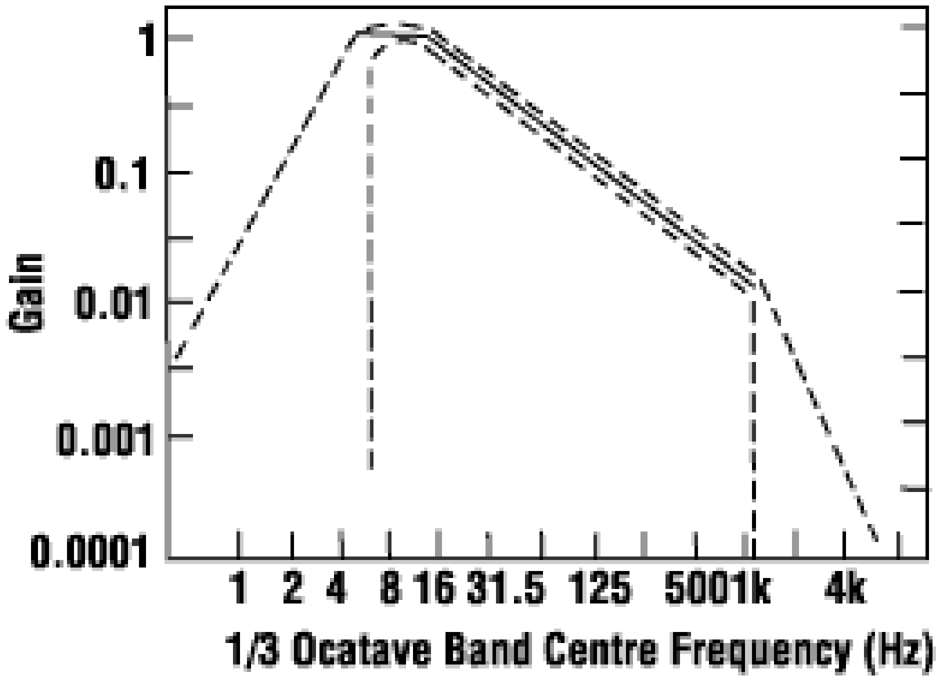

Figure 2.4. The ISO 5349 hand-arm frequency weighting filter.
(CCOHS, 2008)

2.3 Ergonomics

Exposure to vibrations is well known to cause both injuries and discomfort to man. Especially it is high vibration levels or prolonged exposure to vibrations that can be harmful, something that is often present in various workplaces and professions, such as working with a chainsaw or driving a truck. There are several injuries related to exposure of vibrations, some of the most common are white finger syndrome and carpal tunnel syndrome.

With the white finger syndrome the small and shallow blood vessels in the fingers are damaged and the circulation is hindered. This is often triggered by a combination of vibrations and cold, the symptoms include reduced touch sense in the fingers and fingers turning white. The symptoms can regress with time but is often re-triggered by exposure to cold. (Block & Sequeira, 2001)

Carpal tunnel syndrome is when the nerves leading through the wrist to the hand, through the so called carpal tunnel, are being squeezed. This causes numbness in the hand, muscle paralysis and a general discomfort or pain. It is usually triggered by prolonged and awkward positioning of the wrist or heavy manual work, often in combination with vibrations. The condition can be relieved by surgery, splinting of the wrist or through medication. (McCartan, 2012).

In order to limit the amount and the severity of vibrations related workplace injuries many countries have laws and regulations stating how large levels of vibration and how long exposure is acceptable, as well as required preventive and counteractive actions to be taken by the employer to ensure employee safety. (CCOHS, 2008), (Arbetsmiljöverket, 2005)

Usually the vibration hazard to the human body is divided into two different categories, hand and arm vibrations and full body vibrations, depending on how they affect our bodies. In the case of a power cutter it is only hand and arm vibrations that are considered. In Sweden there are two different threshold levels defined by law for hand and arm vibrations. The first corresponds to a total exposure of $2.5ms^{-2}$ in a full eight hour work day.

Anything below this level can be used without breaks or without any particular measures being taken. The second level corresponds to a daily exposure of $5ms^{-2}$ and anything above this level is not allowed at all. Anything in between the two different levels is acceptable but requires some special actions such as regular medical check-ups or workday planning that helps limit the exposure. (Arbetsmiljöverket, 2005)

Generally the most profitable situation is where an employer can have their employees work without limitations from this type of laws. Because of this it has become a competitive selling point for manufacturers of professional machinery to have the lowest possible levels of vibrations in their products.

2.4 Vibration Control

Because vibrations often have negative effects on both machinery and operators as previously mentioned there is often a need of some form of vibration control to be implemented. Mechanical failures can often be avoided by simply reinforcing structures but this tends to increase both the weight and the cost of the machine and is hence commonly not a desirable method. Instead various measures to reduce or manipulate the vibrations can be implemented. These measures are generally divided into two groups, active and passive vibration control.

Passive control includes all methods where devices manipulate the vibrations without any external energy being added. A common method of passive control is using rubber buffers (see figure 3.1) or other viscous damping to mitigate the vibration level by dissipating part of the vibration energy as heat. Another common method is using springs to isolate a vibrating part from its neighbouring parts. Springs achieve this isolation by shifting the natural frequency of the system lower or higher depending of their stiffness. (Rogers Corporation, 2012)

Active control means any measure that counteracts vibration by relying on external power and a series of actuators such as electric or hydraulic. These types of methods can achieve very low vibration levels but tend to be heavier and more expensive than passive methods. They are also more complex as they need some type of control system and hence they are more prone to breaking or failing and do require the presence of an external power source. (Sciulli, 1997)

A third approach to manipulate the vibrations in a system is to simply attack the source of the problem. In most cases this means considering changing the engine to one which produce less vibrations or operate in a different frequency range that is not as problematic, but this is not always possible to do due to other limitations in the product such as weight, size or fuel efficiency.

Figure 2.5. One type of passive vibration control solution is rubber buffers. (Direct Industry, 2014)

2.5 The K760 Power Cutter

Husqvarna's smallest power cutter is called K760. It has a 73.5 cm^3 air cooled two stroke engine. It has a competitive power-to-weight ratio, where the power output is 3.7 kW and it weighs 9.8 kg. It can be fitted with either a 12" or a 14" blade. The tank holds 9 dl of gasoline (Husqvarna AB, 2014). The vibration level is at 2.4 ms^{-2} at both the front and rear handle which implies that it is legal to use the power cutter for an eight hour work day without any special restrictions (Arbetsmiljöverket, 2005).

The vibrations are dampened by three springs that are mounted between the engine part and the handle part. The tank, carburettor and air filtration system are mounted to the handle part while the engine, muffler, clutch, transmission, blade guard and blade make up the engine part. The power train consists of two pulleys with a rubber belt between and the cutting blade is rotating with half the engine rotation speed.

The cutter is used for cutting in concrete, asphalt, pipes and rails etc. The users of power cutters may be split into two different groups, the ones using it almost every day working as concrete cutting professionals and those who use a power cutter on more seldom occasions. The ones using it almost every day most often own the machine themselves and take good care of it, while those less frequently using the machine more often rent it and might not be as careful with it.

2.6 Competitors Solutions

When it comes to power cutters the solution for vibration control is largely the same among different manufacturers. Almost all manufacturers use springs and place these in largely the same places, although sometimes the number of springs differ between either three or four. A few manufacturers also use rubber buffers in combination with the springs and at least one manufacturer, Makita, use only rubber buffers in one of their machines, the EK8100 (Makita, 2014). Currently Husqvarna is the leading manufacturer when it comes to achieving low vibration levels in the machines.

Figure 2.6. Stihl (left) and Makita (right) power cutters.

The overall design of power cutters also tend to be similar, as well as the functionality. The design of handles and control interfaces varies a little bit from manufacturer to manufacturer, and machine to machine.

2.7 Vibration Control in other Products

In chainsaws the solutions for vibration control are quite similar to the ones in power cutters. The big difference is that rubber buffers are used to a larger extent and springs being rarer, although rubber buffers and springs are sometimes used in combination on power cutters as well. On power cutters rubber buffers are exposed to an alkaline environment, due to the concrete slurry that is created (The city of Edmonton, 2014), which implies that they wear out earlier than the springs.

In aircraft where there are often complex electronic systems and where reliability is of critical importance vibration control has developed quite far. Traditional passive technologies are implemented with springs and rubber buffers but this is often not enough. Instead, or in addition, systems like tuned mass dampers (AG&E, 2014) and active control are being used. Because the weight constraints are of high importance in the design of aircraft these systems have also been ingeniously incorporated in various ways. For example some helicopters utilise existing masses such as the batteries in tuned mass dampers. Another example is how modern helicopters may use their rotor blades in combination with their already existing advanced control systems to make minute blade angle adjustments at precisely calculated frequencies to counteract vibrations. (Pearson et al., 1994)

To avoid critical failures in tall buildings in earthquake and hurricane prone areas large scale tuned mass dampers are sometimes used. One example is the Taipei 101 skyscraper which has been built with three tuned mass dampers, the largest weighting 660 metric tons, supported on hydraulic dampers. (Taipei Financial Center Corp., 2009)

2.8 Conclusions

In the power cutter vibrations are primarily created due to the exciting force of the engine, the reciprocating movement of the piston and due to the imbalance in the rotating blade. It is important to know which parts of the power cutter that causes vibrations and at what rate respectively path the different parts moves in relation to each other. To be able to figure out from the measurements how much and which parts that contribute to the vibrations there is a need for an accurate and reliable measurement equipment. Piezoelectric accelerometers will be used to measure the amplitude and frequency of vibrations and a tachometer will be used to relate this to the engine rpm. This should give accurate results from the measurements.

At this point it is possible to partially answer the research questions and decide what is still necessary to investigate:

- *What is a reliable and sufficiently accurate way to measure the vibrations in a power cutter that allows the detection of effects of minor changes to the structure?*

  ⇒ It is known that the equipment provided by Husqvarna will be very accurate, but the question still remains if the human factor affects the ability to get repeatable measurements. The option of using a test rig will be investigated.

- *What are the causal relations of the transferring of vibrations from the cutting part of the cutter to the handles?*

  ⇒ It is known that the vibrations are dependent on the stiffness of a structure. Therefore the transmissibility of the vibrations will be related to the stiffness of the attached springs. This will be done by experimentally testing various spring stiffness in the K760.

- *Are there any natural frequencies of the power cutter that needs to be taken into consideration?*

  ⇒ It is well known that natural frequencies can provide dangerous amounts of magnification to the vibrations and hence they must be investigated thoroughly. The natural frequencies will be examined with the results from the measurements.

- *How can the vibrations isolators be designed to further limit the felt vibrations of the power cutter without sacrificing the controllability of the machine?*

  ⇒ Some knowledge of other methods to reduce the level of vibrations have been gained but this master thesis will not result in any advanced changes to the K760 and hence only handles springs as vibration isolators. How these springs should be altered to further reduce the vibrations will be examined experimentally.

3
METHOD

This chapter will discuss the overall research process. It will briefly touch upon the performed pre-study, the necessary preparations and the specific testing procedures.

3.1 Pre-Study

Before the testing was started a literature study was carried out. It focused on developing a fundamental understanding on the fields of vibration control and vibration measuring but also included a review of existing solutions for dealing with the problem of vibrations. Another necessary part of the pre-study was to develop familiarity with the specific equipment available for the thesis work; this included the K760 power cutter itself as well as the measuring equipment and testing facilities.

The pre-study was concluded with a series of study visits and interviews. Interviews were held with experts in related fields at both Husqvarna AB and Chalmers University of Technology. These provided an extension to the theoretical understanding of vibrations as well as hands-on tips for how to perform tests and measure vibrations. Two study visits were made, one at Husqvarna's department of physical measurements' vibration laboratory and one at Husqvarna's calculation and simulation department. These both helped increase the understanding of the specific case of the K760 and provided a better understanding of problems that could be encountered.

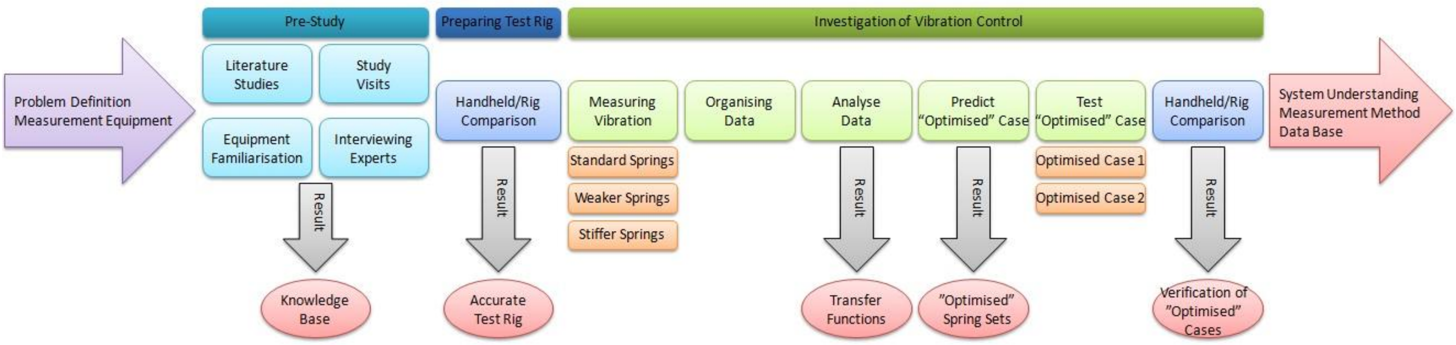

Figure 3.1. Flow chart describing the research process.

3.2 Preparing the Test Rig

Because the testing procedure had to be both reproducible and repeatable to facilitate reliable testing also outside the scope of this thesis using handheld testing, as is normal, had to be replaced with testing in a standardised rig. To achieve this the first step of the research process was to investigate the differences between measuring with the power cutter handheld and supporting it in a test rig. Based on the results of this investigation as well as theory on the field of vibration measuring, modifications were made to the rig and the procedure was then repeated until the results of the hand vibrations were as close to identical as seemingly possible.

3.3 Investigation of Vibration Control

Once a test rig was in place the actual testing could begin. First the power cutter was tested in its current configuration. Ideally the vibrations should be measured on both attachment points of each isolating spring in three dimensions simultaneously but due to limitations in the existing equipment these tests had to be divided into three segments where one direction, x, y or z, was measured at both ends of one spring at a time. Once all directions had been measured the test was repeated for the remaining two springs. After all data had been collected for the current production configuration of the power cutter the tests were repeated with a set of weaker springs installed followed by testing with springs stiffer than the current ones. The springs are shown in figure 3.2.

Figure 3.2. The different springs that were tested. From left to right: Weak, prototype, standard and stiff.

Following this first series of tests the data was organised and analysed. The analysis focused on finding a relationship between the stiffness of the vibration isolating springs and the transmissibility, the percentage of vibrations transferred across the spring. Based on the findings in this analysis it was attempted to predict the effect of making certain changes to the springs and two different perceived optimised spring sets were arranged.

These two sets of springs were then installed and tested in the same manner as the previous spring sets. Then the machine was removed from the test rig and was again tested while handheld to compare the results to the initial handheld tests and see if the predicted optimised case actually gave improved handle vibration results.

All the collected data and the done analysis of this data led to a series of conclusions and helped answer the stated research questions. This in turn also resulted in recommendations for future testing and research as well as a recommended measuring methodology and a test rig design.

4

MEASUREMENT SYSTEM DESIGN

This chapter discusses the test rig design and how the test rig was used. It also describes the modifications done to the test object to increase repeatability of measurements and defines the coordinate system and the overall view of the vibration control in the K760 as a series of systems. There is also a brief description of the necessary tests to ensure proper results later on, such as verifying that the maximum rated temperature of the accelerometers is never exceeded.

4.1 The Power Cutter as a System

There are a total of three springs for vibration control in the K760. One between the forward handle and the engine block, one between the tank and the air intake and one between the forward feet and the cutting arm. When analysing the vibrations while using the power cutter it is divided into three separate subsystems, one around each of these springs (see figure 4.1).

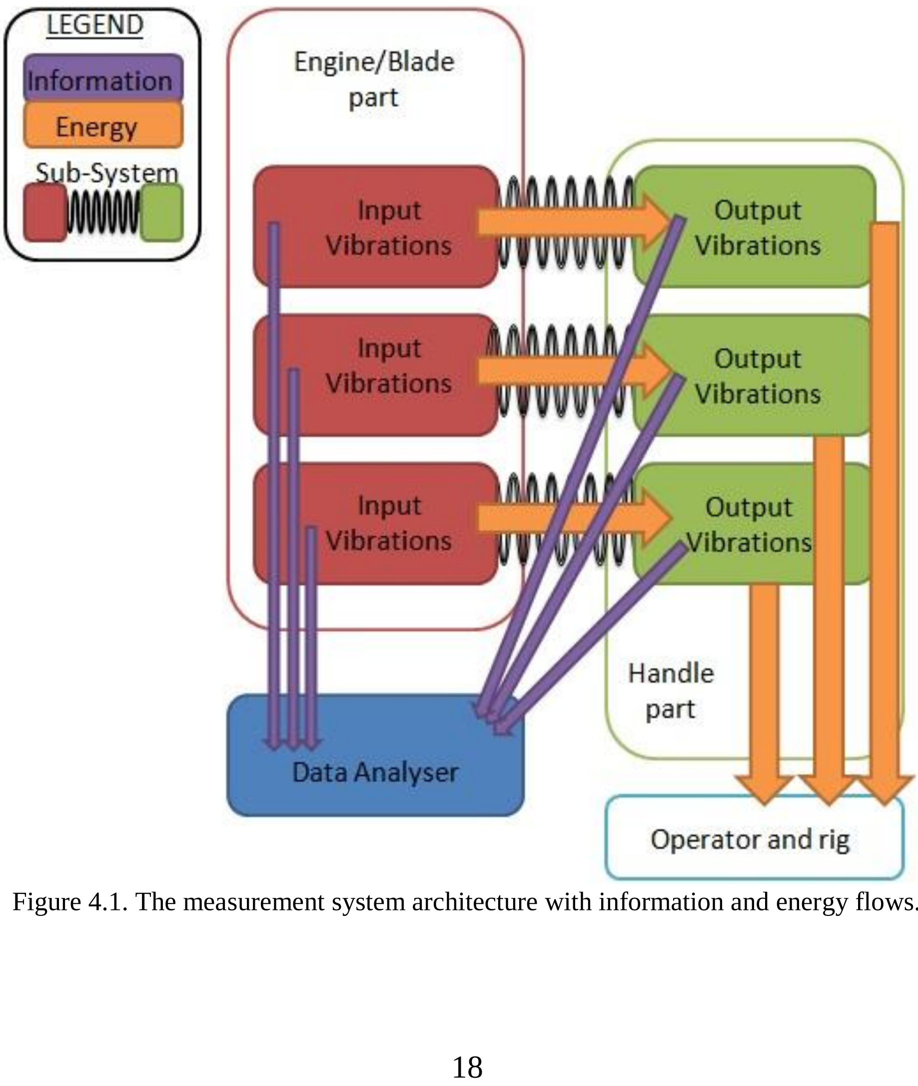

Figure 4.1. The measurement system architecture with information and energy flows.

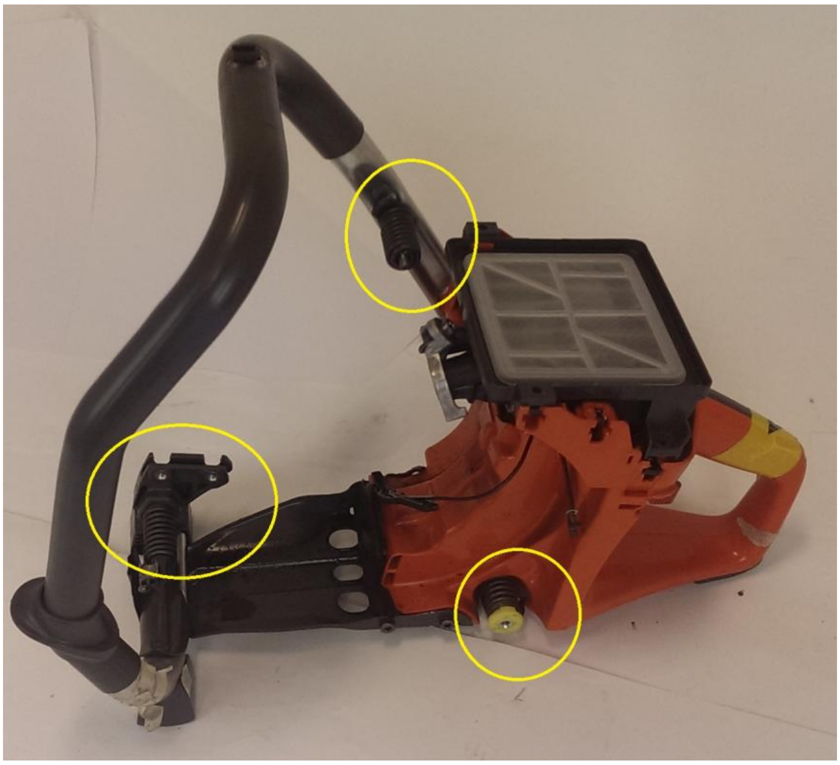
Figure 4.2. The three vibration isolating springs on the K760.

All three springs acting on the K760 have the same orientation, working in similar ways in the different directions (see figure 4.2). Therefore a three-axis coordinate system is applied for defining the directions of the power cutter, shown in figure 4.3. When placed on a flat surface, the upward direction coincides with the z-axis, the longitudinal direction of all three springs coincides with the y-axis and the forward direction when holding the power cutter coincides with the x-axis.

Figure 4.3. The defined coordinate system in relation to the K760.

4.2 Accelerometer Placement

The measurement equipment had the ability to measure one direction on both sides of the springs during the same measurement. It was then possible to see how much vibration that is transferred through them.

The stiffness are defined as axial and lateral stiffness (see figure 4.4), both defined from the centre point of the spring, the axial stiffness acting in the longitudinal direction and the lateral stiffness acting in the bending direction. Therefore the accelerometers are placed as close to the centre of the ends of the springs as possible to minimise the error of the measurement (see figure 4.5). To achieve this it became necessary to modify certain parts of the power cutter.

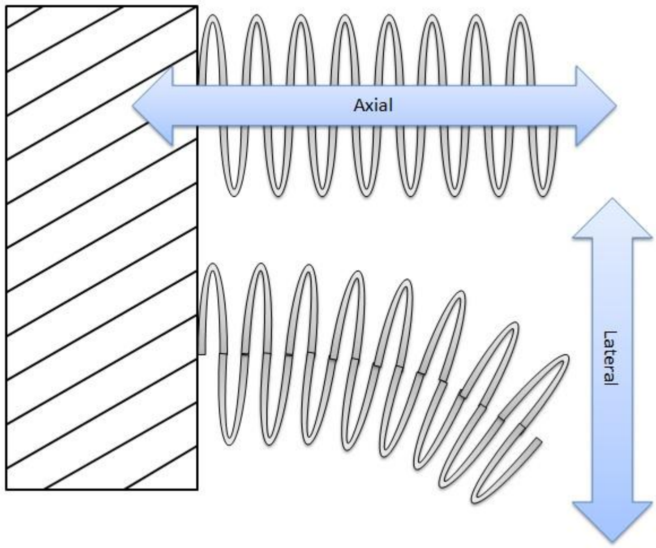

Figure 4.4. Defining the axial and lateral directions of the springs.

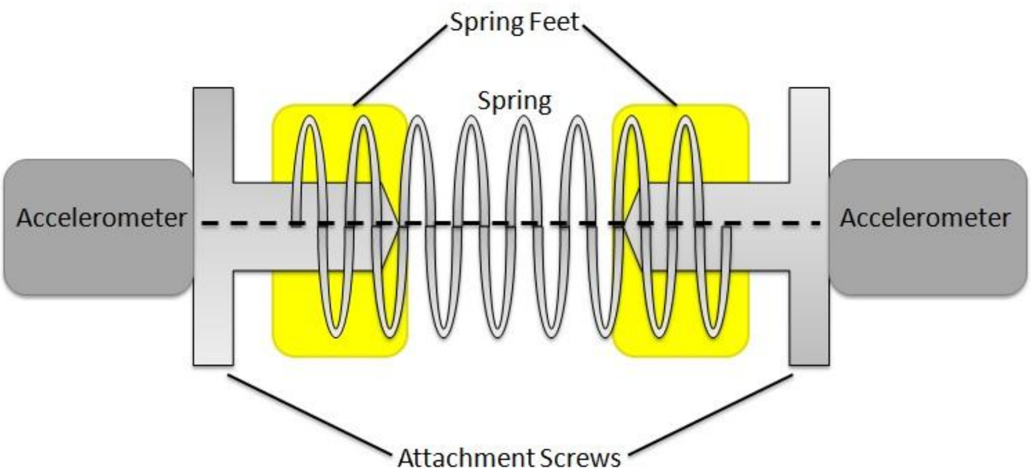

Figure 4.5. The accelerometer placement in relation to the spring.

It was also decided that in order to get a high degree of repeatability the accelerometers had to be possible to mount in precisely the same direction for all three spring sets and perfectly perpendicular for the different measurements, hence small mounting brackets had to be manufactured (see figure 4.6). The standard way of attaching the accelerometers is by bolting them to a small threaded mounting plate that can be semi-permanently glued to the structure. Since there is very limited room for mounting the accelerometers in some of the decided areas there is no room to use the thread. To be able to fasten the accelerometers in these confined places mounting magnets were used instead. To lock the sideway translations of the accelerometer and make sure the measurements were repeatable guiding pockets fixated the magnet. There was a need for six brackets, one for each end of the vibration isolators.

To be able to place an accelerometer near the end of the spring mounted into the fuel tank the tank had to be empty from gasoline. The gasoline was instead entered by a tube connected to an external fuel tank. By placing a sealed plastic bag filled with water inside the gasoline tank of the power cutter the properties of a half filled tank of gasoline was simulated. This also has the advantage that the tank volume remains constant throughout the test.

To be able to relate the measured accelerations to the engine rpm a tachometer was used to register the engine speed. It is placed such it is pointing towards the ignition magnet at the flywheel from which it registered one pulse per engine axle revolution.

4.3 Mounting Bracket Mass Tests

It was necessary to verify that the mass added by the mounting brackets did not affect the results. This was done by first attaching the mounting brackets and running a test, and then lighten the bracket to about 50% of its original mass. Then the test was redone and the process repeated until the results no longer changed with the reduction of the bracket’s mass, at which point the bracket was lightened slightly more for safety margin.

The tests showed that it was possible to use the lightened mounting brackets. When sufficiently lightened, with some margin, the brackets did not affect the results enough to be perceived through the normally present result variations while still maintaining sufficient mass and surface area to successfully hold the accelerometers.

Figure 4.6. The mounting brackets used to achieve perpendicular accelerometer placement.

4.4 Engine Temperature Investigation

The accelerometers are specified to operate within a certain temperature span with a maximum of 125°C. One of the vibration isolators on the K760 are attached directly to the engine block, therefore the temperature had to be measured to make sure that the equipment does not get damaged. The temperature was measured while running the engine with two different kinds of sensors to increase the reliability of the results.

The results from the performed measurements showed that the temperature stayed below the maximum rated temperature of the accelerometers at all times at the points interesting for accelerometer placement. The accelerometers are rated to work properly up to 125°C and the maximum measured temperature was about 112°C on the heat sink where the accelerometers should be placed (see figure 4.7a). The maximum temperature on the engine block itself was measured to 147°C (see figure 4.7b) which does exceed the rated temperature of the accelerometers. It was however believed that the accelerometers would never reach their maximum rated temperature at the point they were meant to be placed and that it was safe to proceed.

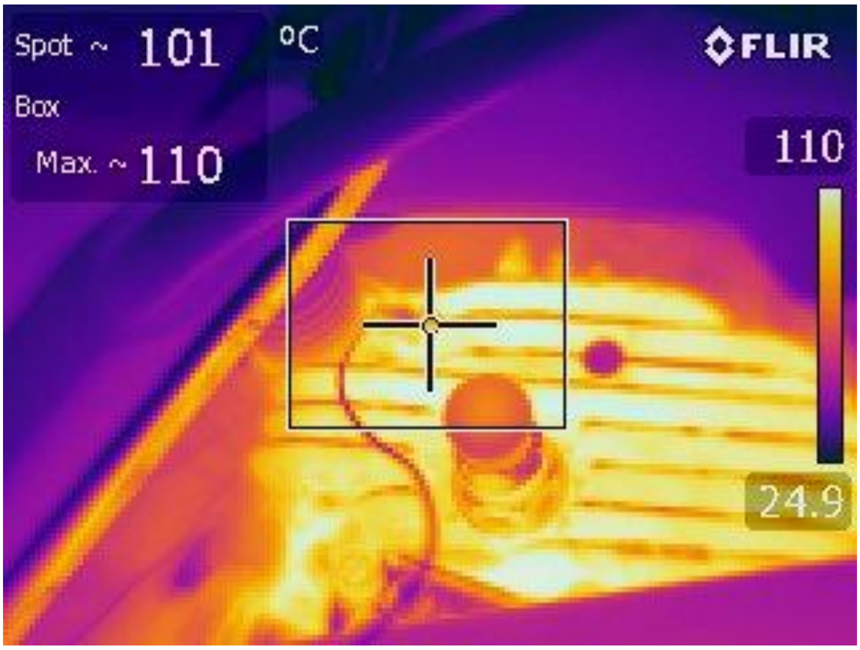

Figure 4.7a. An image from the FLIR-camera showing the temperature of the edge of the heat sink flanges.

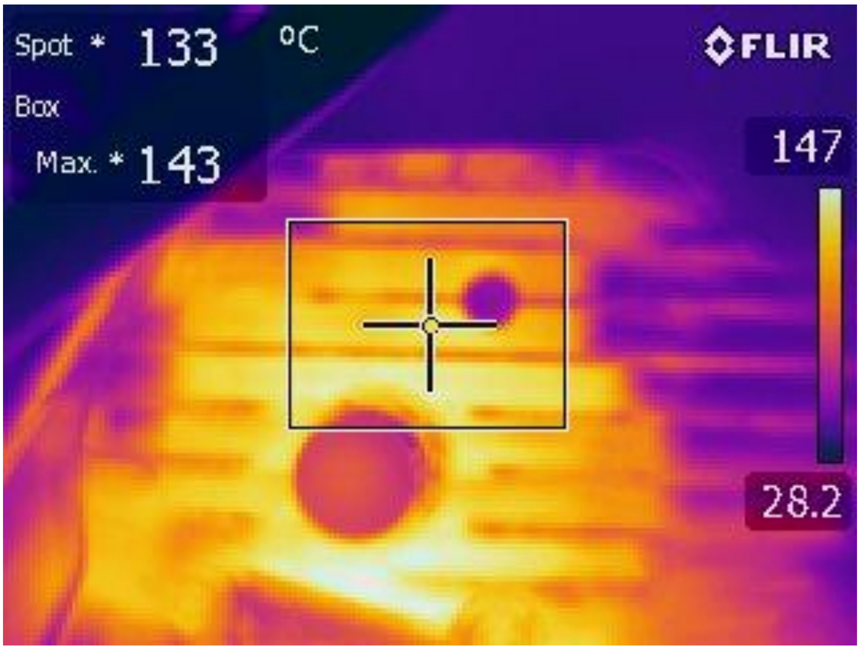

Figure 4.7b. An image from the FLIR-camera showing the temperature of the cylinder wall.

4.5 Data Collection

The vibrations are measured during rpm sweeps where the engine speed is steadily increasing at a slow rate to not miss any relevant peaks. The sweeps are performed in two different ranges, the first is a full-span-rpm-sweep which ranges from idle to maximum speed and the second is a short-span-rpm-sweep which is only very close to the idle respectively working speed. The full-span-rpm-sweep gives a great overview of which vibrations that is dampened and how much while the short-span-rpm sweep gives a more in detail description. In order to verify that the data collected is accurate each sweep was run several times and only used when it was possible to achieve the same result multiple times.

4.6 Data Analysis

All data collected was passed through a FFT-analyser which takes the measured time domain signal of the vibration and transforms it to the frequency domain using an FFT (Fast Fourier Transform) algorithm. The advantage of viewing the measured signal in the frequency domain is primarily that it offers a much clearer representation of the various frequency components of the signal. In cases where a gasoline engine is the primary cause of vibrations these frequency components are often represented as frequency multiples of the engine rpm. These multiples are called orders where the first order corresponds to the engine rpm. This representation is automatically done with the FFT-analyser by combining the accelerometer data with the data collected with the tachometer in real time.

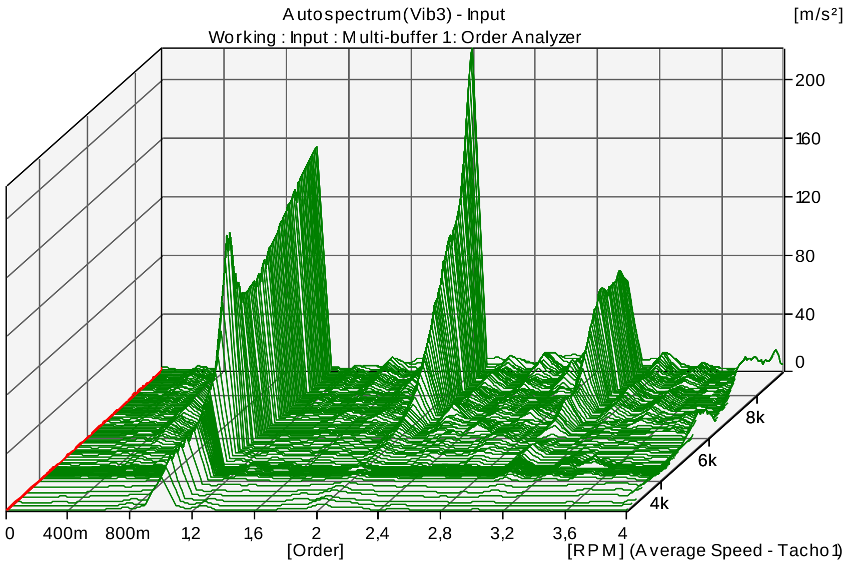

Figure 4.8. An example of a waterfall graph showing the vibrations according to order, the rpm and the amplitude.

The collected data was analysed through waterfall diagrams from the FFT-analyser, which plots the engine rpm, orders of vibrations and amplitude of the vibrations on the three axes. The data was collected during short-span-rpm-sweep which is slow sweeps around the idle rpm of 2700 and the working rpm of 9000. From the short-span-rpm-sweep the amplitude of the vibration at idle rpm and working rpm for the different orders are collected. The data from the measurements with the different springs were processed with MATLAB to produce a transfer function describing the transferring of vibrations in relation to the stiffness of the springs. The short-span-rpm-sweep is also examined visually to see if there is any strange behaviour of the power cutter close to the idle and working speed. The full-span-rpm-sweep is examined visually and helps giving a perception of how well the vibration isolators work and if it has any natural frequencies within the frequency range.

4.7 Comparing the Test Rig to Handheld Measurements

When comparing the measurements with the existing test rig at Husqvarna Construction Products with the results from the handheld measurements it appears that there is a strange amplification at around 32 Hz for the rig (see figure 4.9a and b). Because of this experiments with different stiffness springs in the rig were done. When applying stiffer springs this peak moved higher and with weaker springs it moved lower, indicating that the peak was due to a natural frequency in the rig. To completely remove this natural frequency behaviour from the frequency range of interest for the measurements two additional tests were made. One with an extremely stiff and rigid rig suspension trying to move the peak above the frequency range, and one with very weak springs in the suspension to move the natural frequency below the interesting range.

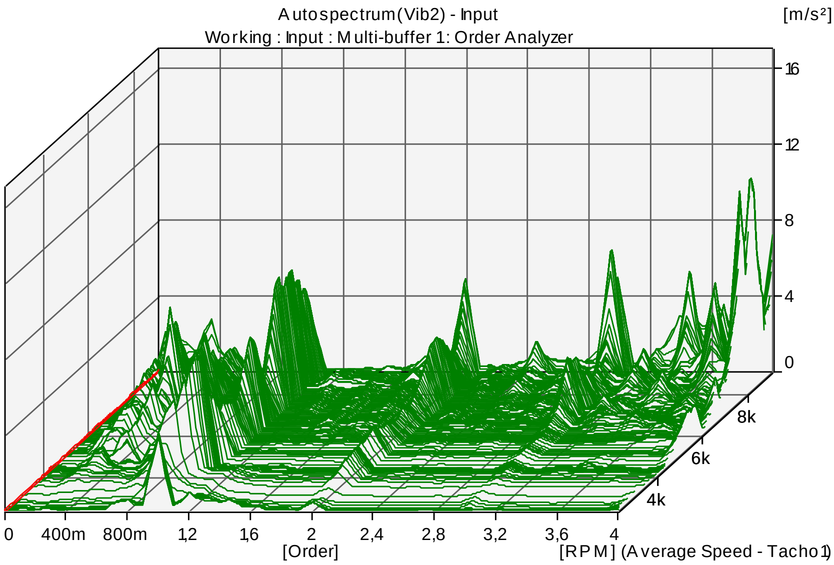

Figure 4.9a. The result from handheld measurements.

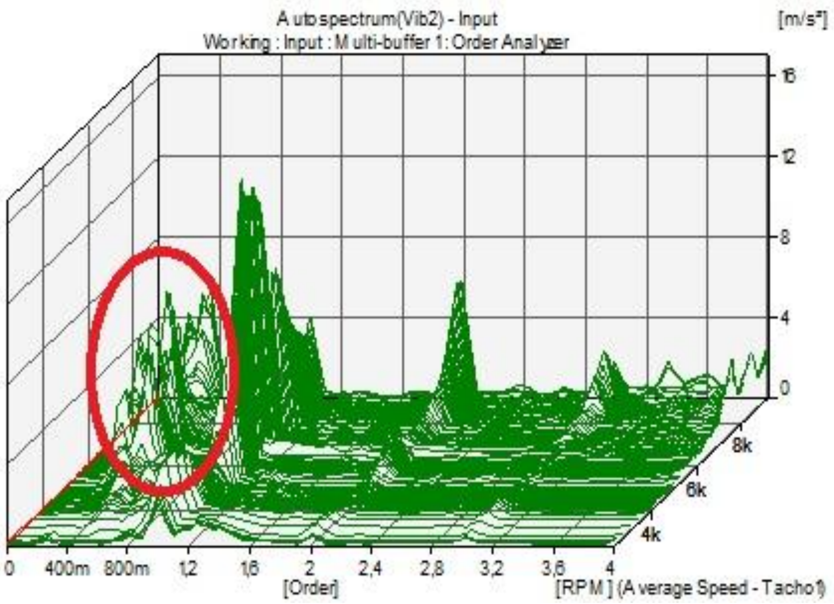

Figure 4.9b. The result from testing in the original test rig.
The diagonal ridge that is marked suggests a natural frequency.

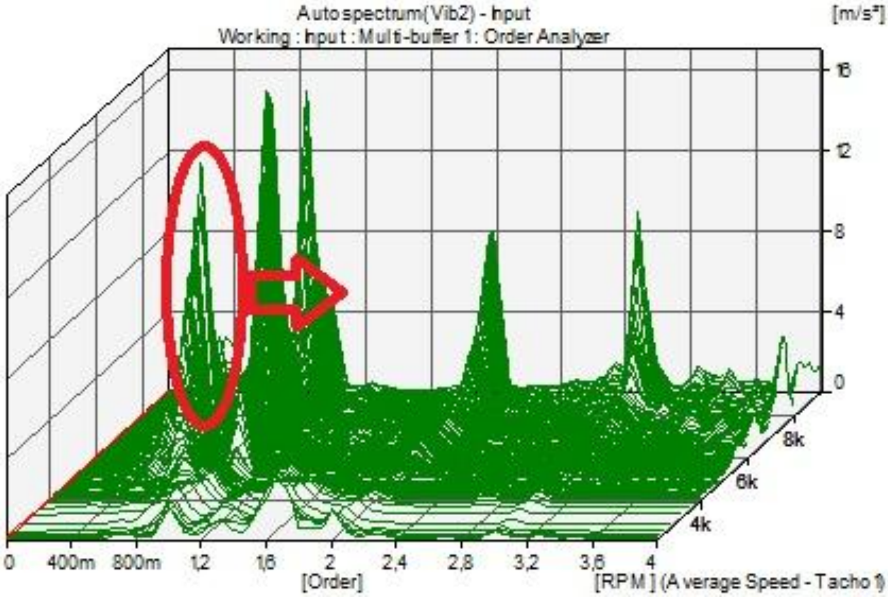

Figure 4.9c. With stiffer rig suspension the natural frequency moves higher.

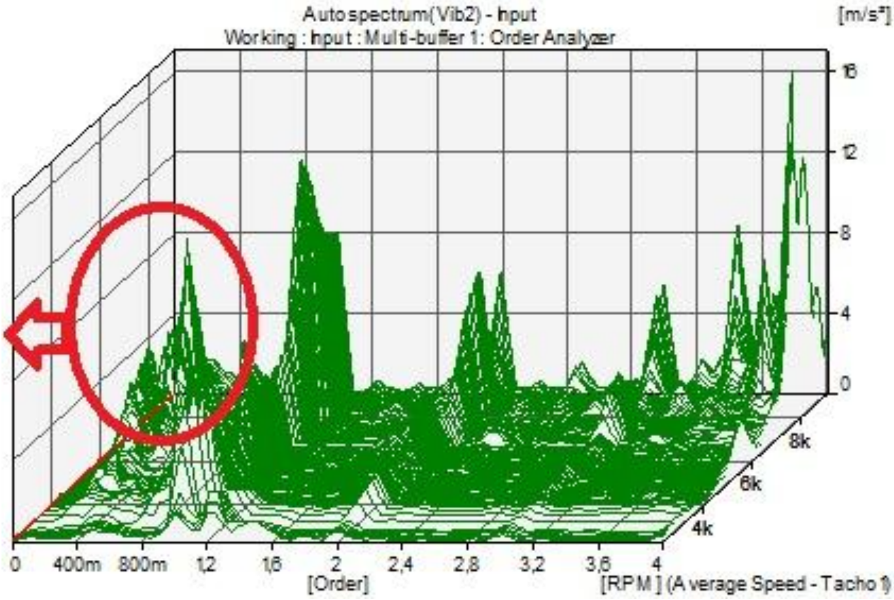

Figure 4.9d. With softer rig suspension the natural frequency moves lower.

The results from the rigid suspension successfully removed the natural frequency from the measured frequency range. It does however also show a very chaotic and random behaviour completely unlike that of handheld measurements and was therefore not deemed to be useable. The very weak spring suspension also was successful in moving the natural frequency outside of the interesting range and gave a result quite similar, although not identical, to that of handheld measurements, hence this approach was selected to proceed with. In order to adjust it more precisely to the handheld results experiments were done where the K760’s handles were loaded with additional mass in the form of rubber padding.

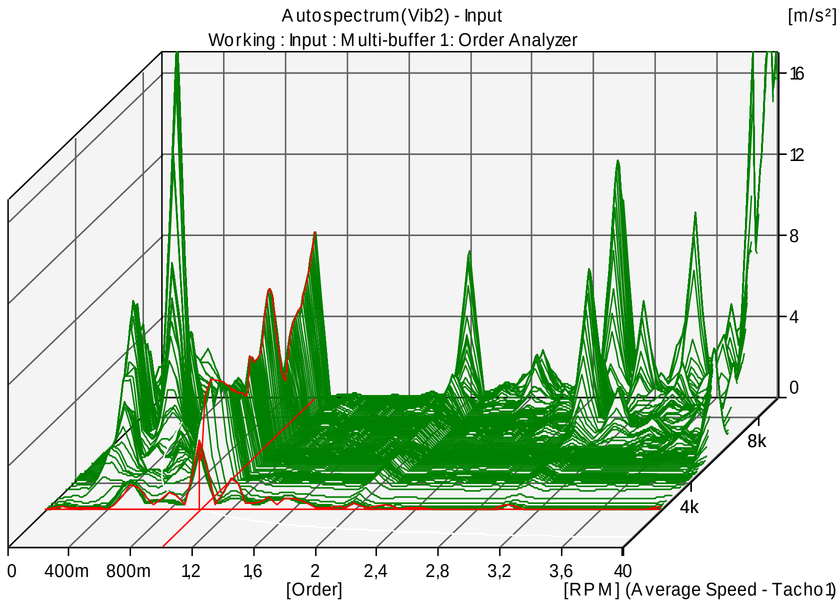

Figure 4.10. The results from mass loading the handles
This shows both similarities and major differences compared to figure 4.9a.

These tests showed that it was possible to achieve an overall better fit with the rig results to the handheld results but the rubber padding always provided some strange behaviour at various points across the measured frequency range (compare figures 4.9a and 4.10), depending on how stiffly it was attached to the handles. Because of this behaviour it was decided to abandon the rubber padding mass loading approach and instead use an operator to lightly hold the K760 during tests, providing some mass to the handles with his hands, while letting the weight of the machine itself be supported by the rig suspension (see figure 4.12). This was a compromise between accuracy in the tests and the convenience of running the tests as this method meant that it takes two people to perform the tests, one to mass load the handles and operate the machine and one to operate the measuring equipment. In combination with these tests the reproducibility was also tested by having the two operators switch roles. The results were extremely similar and could not be distinguished outside of the normal variations.

Figure 4.11a. The old test rig.

Figure 4.11b. The redesigned test rig.

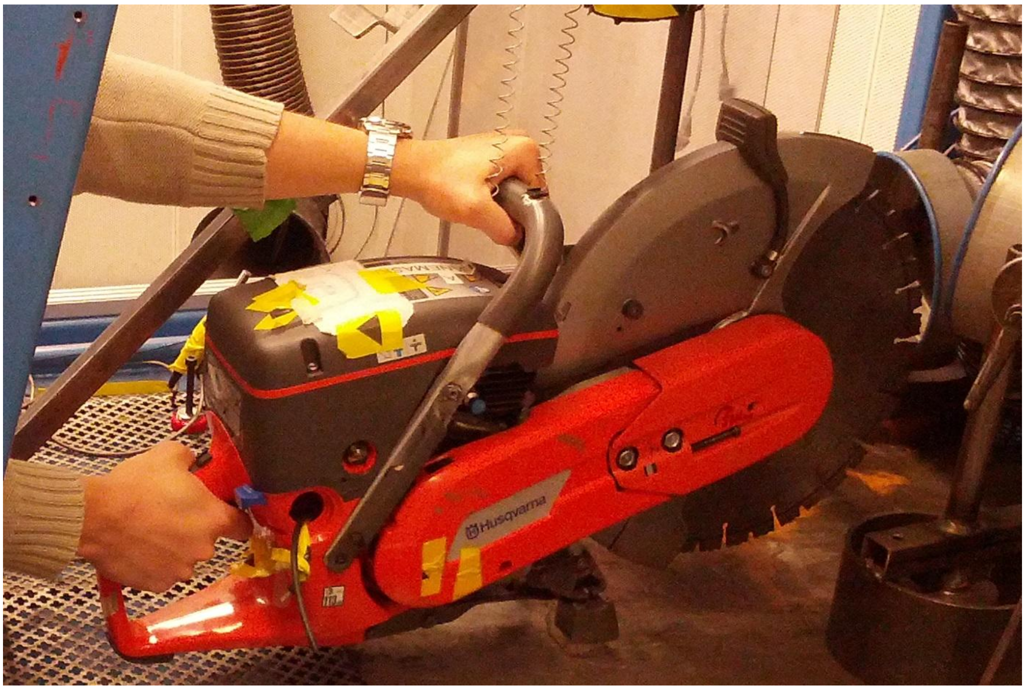

Figure 4.12. How the machine was held in the redesigned test rig.

Figure 4.13. How the machine is hold during handheld measurements.

4.8 Final Modifications to the Test Rig and K760

The following changes were ultimately made to the test rig and the K760 test object to facilitate accurate, repeatable and reproducible further testing:

- Very weak springs installed in the test rig suspension to remove the natural frequency of the rig from the investigated frequency range.
- An external gasoline tank added to eliminate the changing volume and mass of the onboard tank.
- Mounting brackets added to the K760 to enable accelerometer placement at perfectly perpendicular angles at the attachment points of the vibration isolators.
- Operator’s hands put on the K760’s handles to add mass simulating handheld measurement.

5
RESULTS

In the following chapter the final results from investigation of vibration transfer across the vibration isolators will be presented. Focus will be on the most important results, for a more extensive presentation of result graphs and data see appendix D. First the results from analysing the initial series of tests are presented with a focus on attempting to optimise the vibration control elements. After this the verification of the optimisation is presented along with the developed transfer functions.

5.1 Optimisation of Vibration Control

Based on the data in appendix D an assessment of all data collected in an initial series of tests with standard, weak and stiff spring sets were made. This was a somewhat rough assessment where both specific high and low rpm characteristics and general performance were taken into consideration. The purpose of the assessment was to find two "optimised" cases to verify whether or not the data and the transfer functions could be used to accurately predict the effects of changes to the vibration control.

Table 5.1a. The spring stiffness for the cutting arm-foot and air intake-tank points on the K760.

Spring	Axial Stiffness (N/mm) y-direction	Relative Standard	Lateral Stiffness (N/mm) x- and z-direction	Relative Standard
Weak	15.37	63%	13.56	65%
Standard	24.39	-	20.72	-
Stiff	46.82	192%	43.10	208%
Prototype	19.51	80%	20.20	98%

Table 5.1b. The spring stiffness for the engine-handle point on the K760.

Spring	Axial Stiffness (N/mm) y-direction	Relative Standard	Lateral Stiffness (N/mm) x- and z-direction	Relative Standard
Weak	13.57	52%	10.09	55%
Standard	25.88	-	18.23	-
Stiff	33.38	129%	26.16	143%

The assessment were performed using a [++ / + / 0 / - / --] rating system where minus corresponds to an amplification of the vibrations. The zero stands for poor isolation and a plus is given when the vibrations are isolated well. If two different springs both isolate vibrations well in the same direction but one performs much better, the better one is noted with two plus signs and similarly for poor isolation with minus signs. When assessing the level of vibration isolation the 0.5:th, first and second order of vibrations was of greater interest than the higher orders and therefore considered to a higher extent.

Table 5.2a. The assessment for the cutting arm – feet spring.

	Idle rpm (2700 rpm)			**Work rpm (9000 rpm)**		
Spring set	x	y	z	x	y	z
Stiff	0	0	-	-	0	-
Standard	0	-	+	+	0	+
Weak	-	+	+	+	+	+

Table 5.2b. The assessment for the air intake – tank spring.

	Idle rpm (2700 rpm)			**Work rpm (9000 rpm)**		
Spring set	x	y	z	x	y	z
Stiff	-	-	-	0	0	0
Standard	+	+	0	0	+	+
Weak	+	+	+	+	+	+

Table 5.2c. The assessment for the engine - handle spring.

	Idle rpm (2700 rpm)			**Work rpm (9000 rpm)**		
Spring set	x	y	z	x	y	z
Stiff	-	-	-	0	0	0
Standard	-	+	+	+	0	0
Weak	0	+	+	+	0	+

As can be seen in tables 5.2a-c the weaker springs are generally better at isolating vibrations but that there are cases where the standard springs appear to be the same as the weaker ones. There is even one case (table 5.2a) where the standard springs perform better than the weak ones in the x-direction for idle rpm. The stiff spring have poor vibration isolating properties and the vibrations are often not damped at all, sometimes even amplified. It mostly has bad isolation properties at low rpm. Because having too weak springs can cause other problems, such as some loss of controllability of the cutter, there are benefits with selecting the standard springs in these situations. To isolate the vibrations best the solution seem to be a combination of stiffness from the weak and the standard ones for different positions. There is also a possibility to use specially designed springs with for example the standard axial stiffness but the weak lateral stiffness. How the "optimised cases" were composed is shown in tables 5.3b-d.

When comparing the vibrations only on the vibrating side for the three different spring sets to one another it seems that the input vibrations are not significantly affected by the spring set that is being used (see figures 5.1 a and b). This suggests that it is possible to optimise the springs individually without them affecting the output vibrations for the other spring positions.

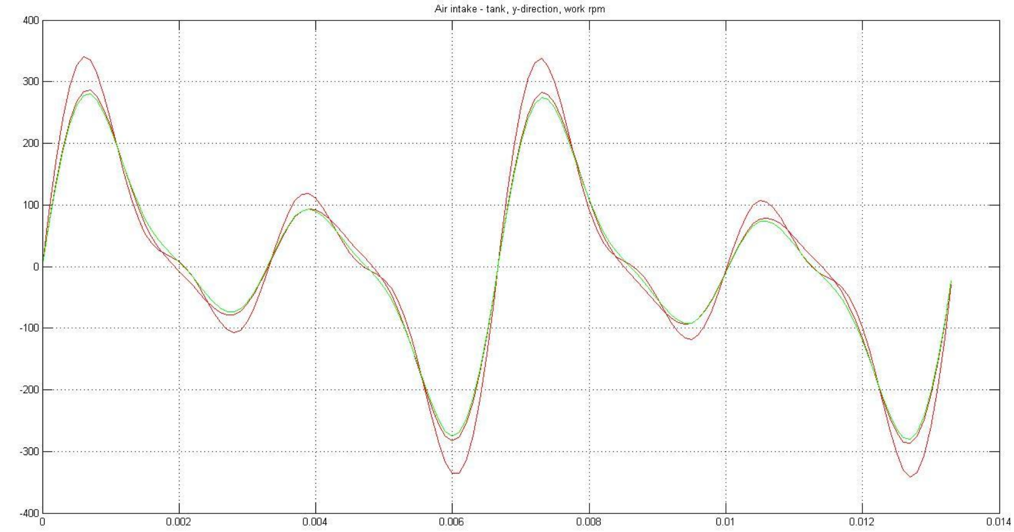

Figure 5.1a. An example of how similar the input vibrations were for the three different spring set. (The different curves represent the three different spring sets)

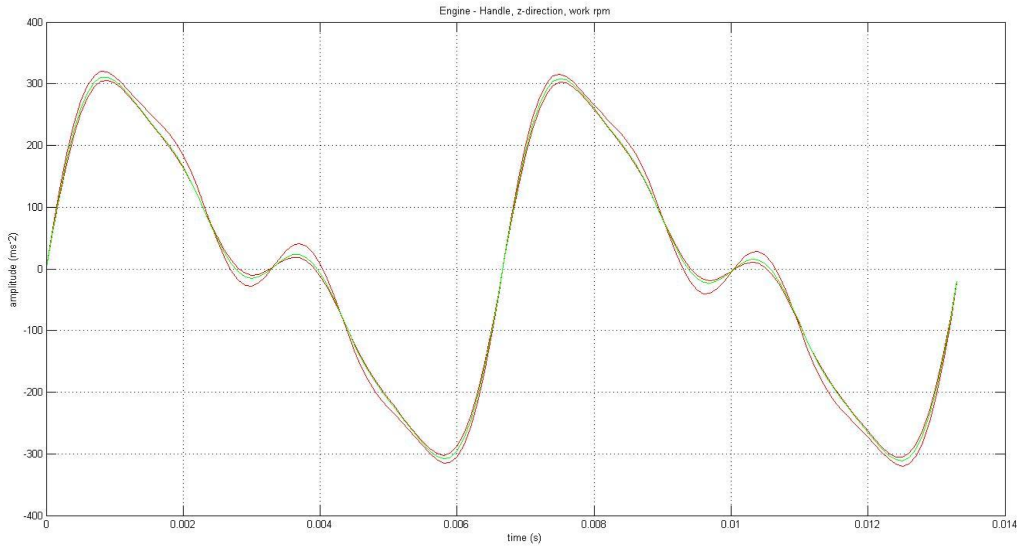

Figure 5.1b. An example of how similar the input vibrations were for the three different spring set. (The different curves represent the three different spring sets)

Table 5.3a. Colour coding for selection of “optimised” cases.

“Optimised” case 1	
“Optimised” case 2	

Table 5.3b. The selection for the cutting arm – feet spring.

	Idle rpm (2700 rpm)			Work rpm (9000 rpm)		
Spring set	x	y	z	x	y	z
Stiff	0	0	-	-	0	-
Standard	0	-	+	+	0	+
Weak	-	+	+	+	+	+

Table 5.3c The selection for the air intake – tank spring.

	Idle rpm (2700 rpm)			Work rpm (9000 rpm)		
Spring set	x	y	z	x	y	z
Stiff	-	-	-	0	0	0
Standard	+	+	0	0	+	+
Weak	+	+	+	+	+	+

Table 5.3d. The selection for the engine - handle spring.

	Idle rpm (2700 rpm)			Work rpm (9000 rpm)		
Spring set	x	y	z	x	y	z
Stiff	-	-	-	0	0	0
Standard	-	+	+	+	0	0
Weak	0	+	+	+	0	+

Further experiments were performed with springs having properties that correspond to these results. The two “optimised” cases sets were composed according to table 5.1, where “prototype” is a specially designed spring to better fit the wanted characteristics. All other terms refers to the previously used springs.

Table 5.4. The composition of the “optimised” cases sets.

Case	Cutting arm - feet	Air intake – tank	Engine - handle
“Optimised” case 1	Prototype	Weak	Weak
“Optimised” case 2	Prototype	Prototype	Standard

5.2 Transfer Functions

Below are the results from the measurement across the vibration isolators. These tests were initially done for three sets of springs; the standard production springs, a set of weaker springs and a set of stiffer springs and later done with two additional sets for the “optimised” cases. As can be seen in table 5.3, the “optimised” cases sometimes use springs from the other sets. The analysis was done with a focus on investigating transmissibility in relation to spring stiffness. The data was collected at 2700 and 9000 rpm for orders 0.5, 1, 2, 3 and 4 from waterfall diagrams for all directions for all springs and converted to transfer functions.

The transfer function describes the transmissibility in relation to the stiffness for a single direction and vibration isolating element on the power cutter. This results in a total of 18 different transfer functions describing the transmissibility behaviour for the entire K760 (see figures 5.2-5.4). The y-axis shows transmissibility, where 1 means 100% of the vibrations are transferred to the isolated end of the vibration isolating spring. The x-axis show the spring stiffness from lowest to highest and the different coloured lines show the different orders of vibrations.

Table 5.5. Colour coding of the orders in the transfer function graphs.

COLOR	ORDER
	0.5:th
	1:st
	2:nd
	3:rd
	4:th

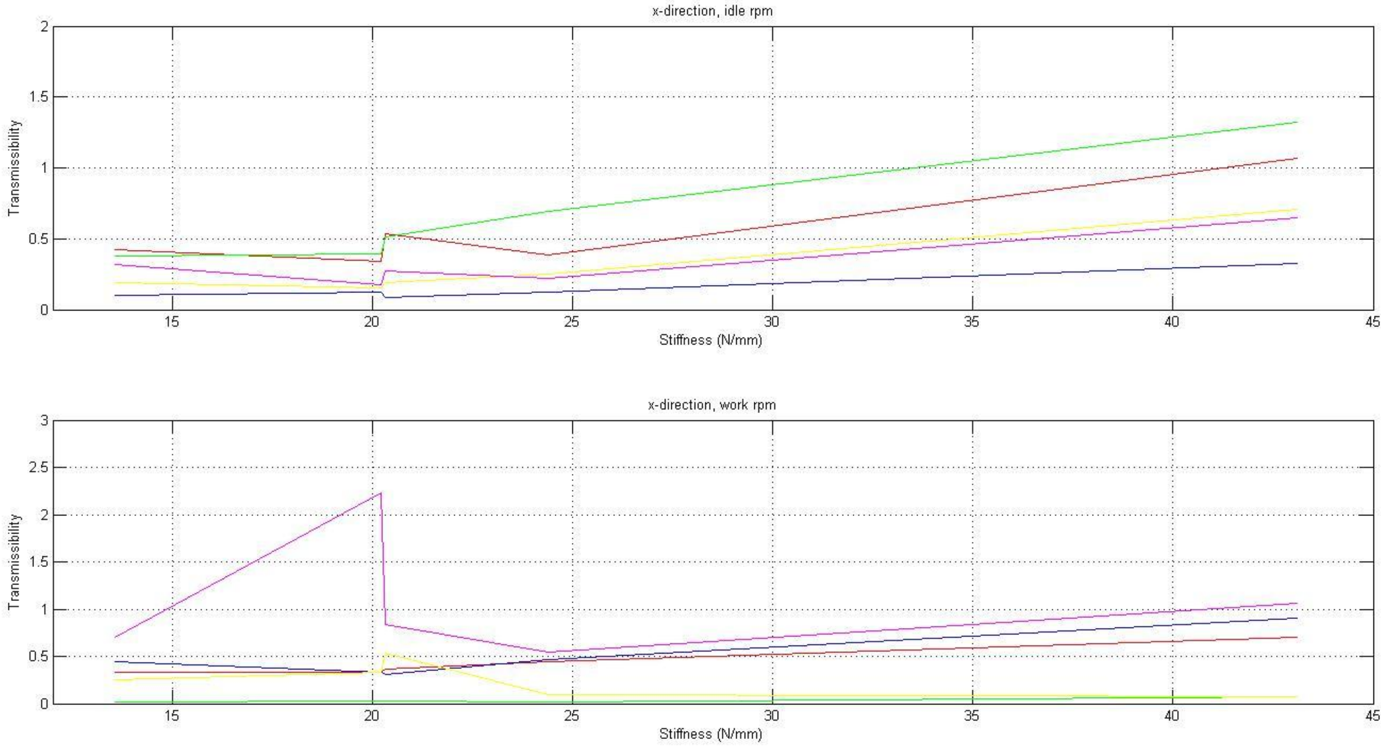

Figure 5.2a. Transfer function for the cutting arm – feet, x-direction.

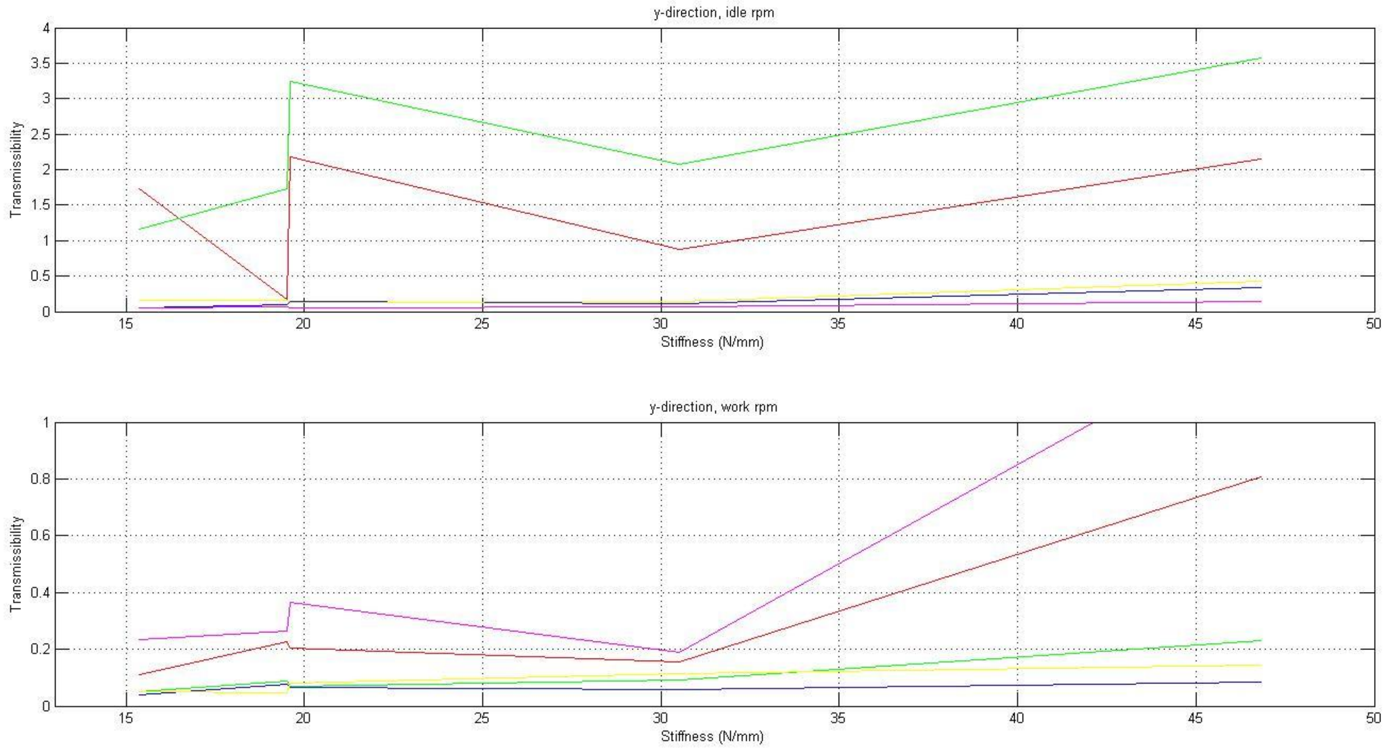

Figure 5.2b. Transfer function for the cutting arm – feet, y-direction.

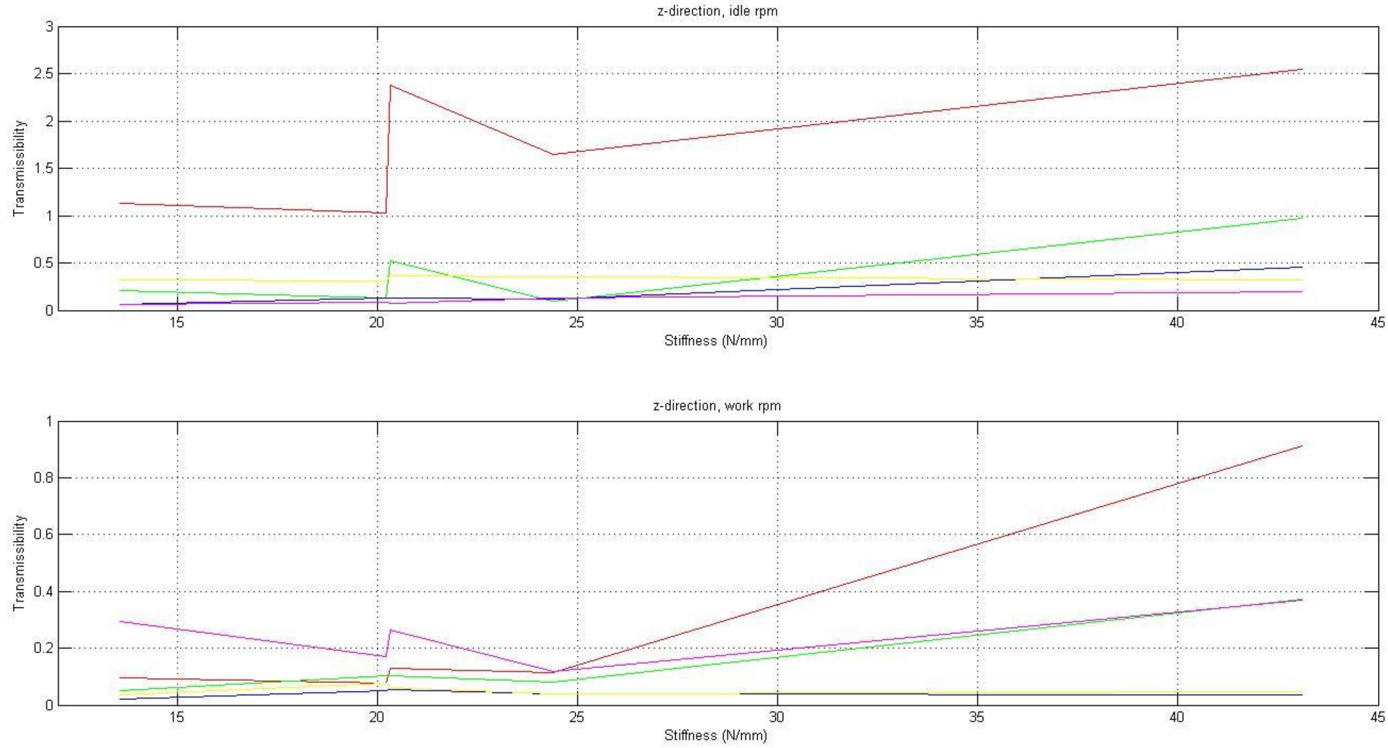

Figure 5.2c. Transfer function for the cutting arm – feet, z-direction.

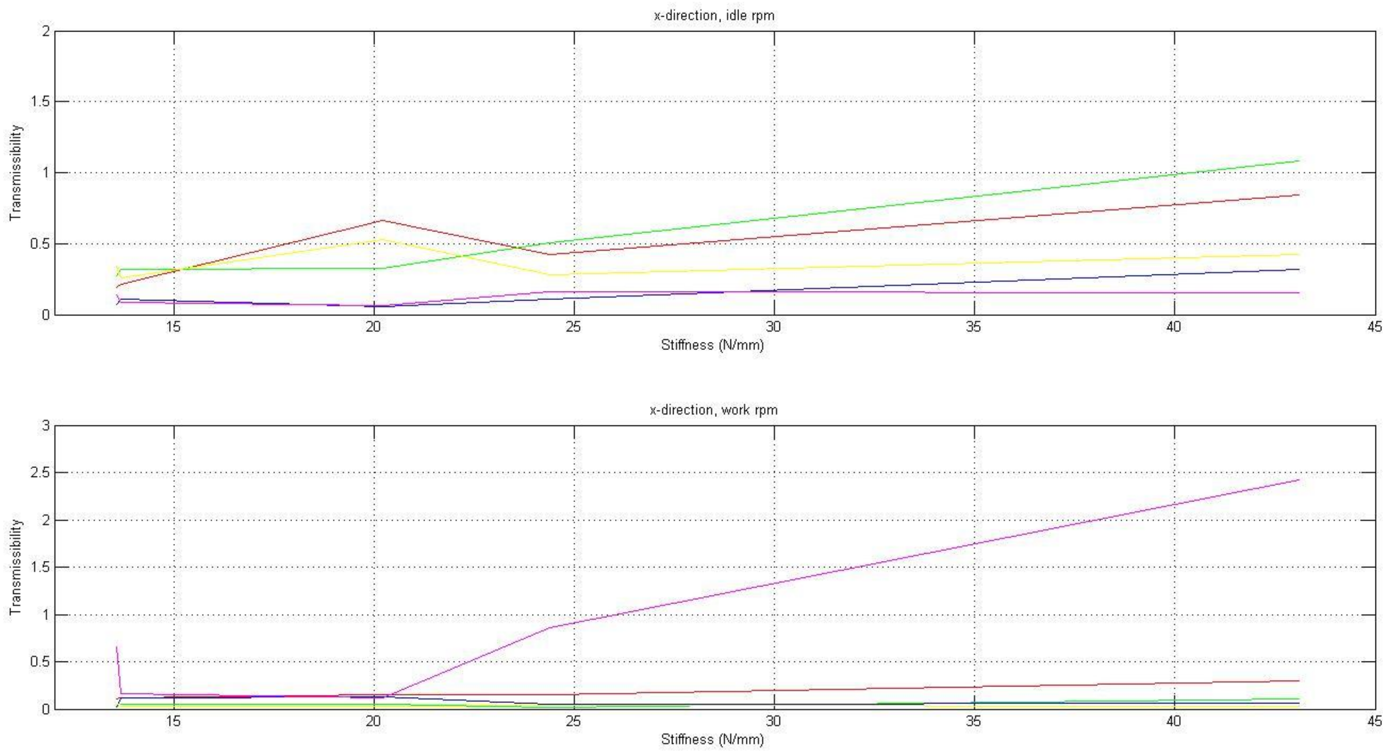

Figure 5.3a. Transfer function for the air intake – tank, x-direction.

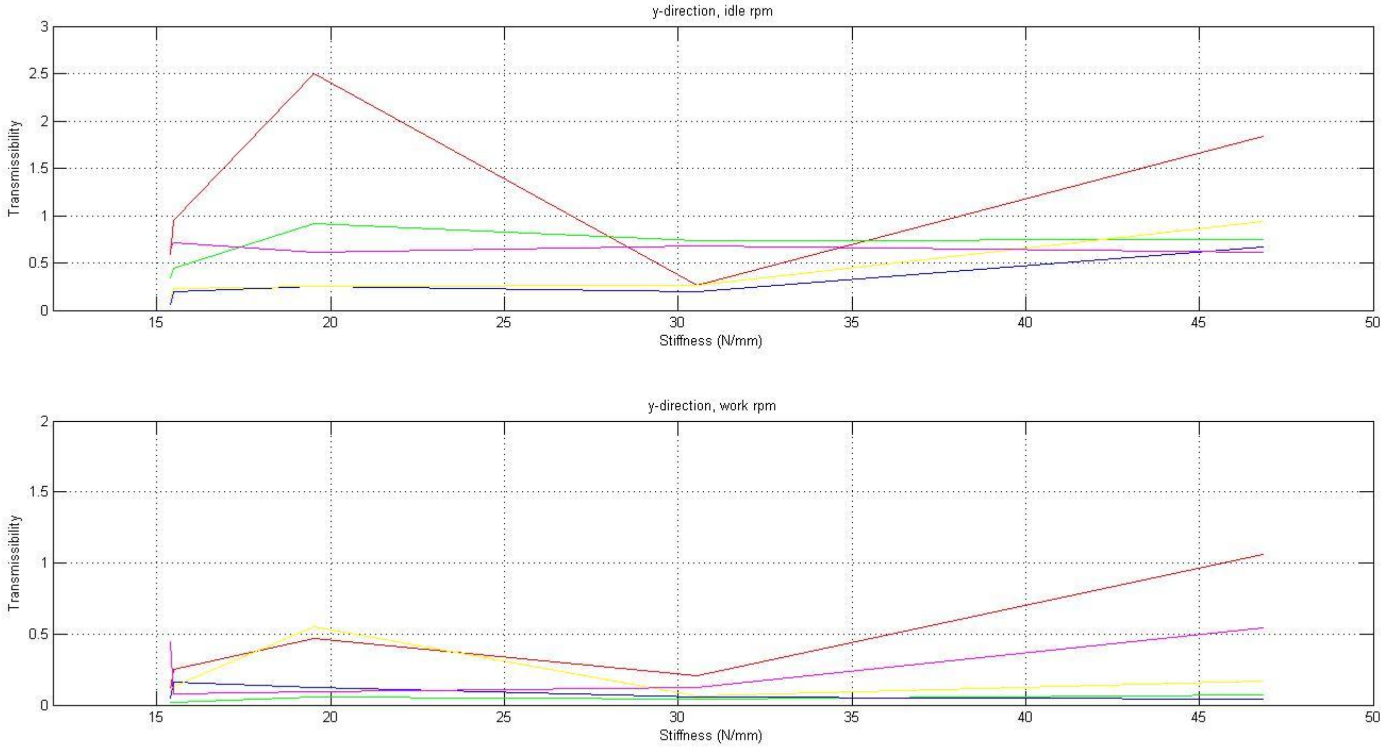

Figure 5.3b. Transfer function for the air intake – tank, y-direction.

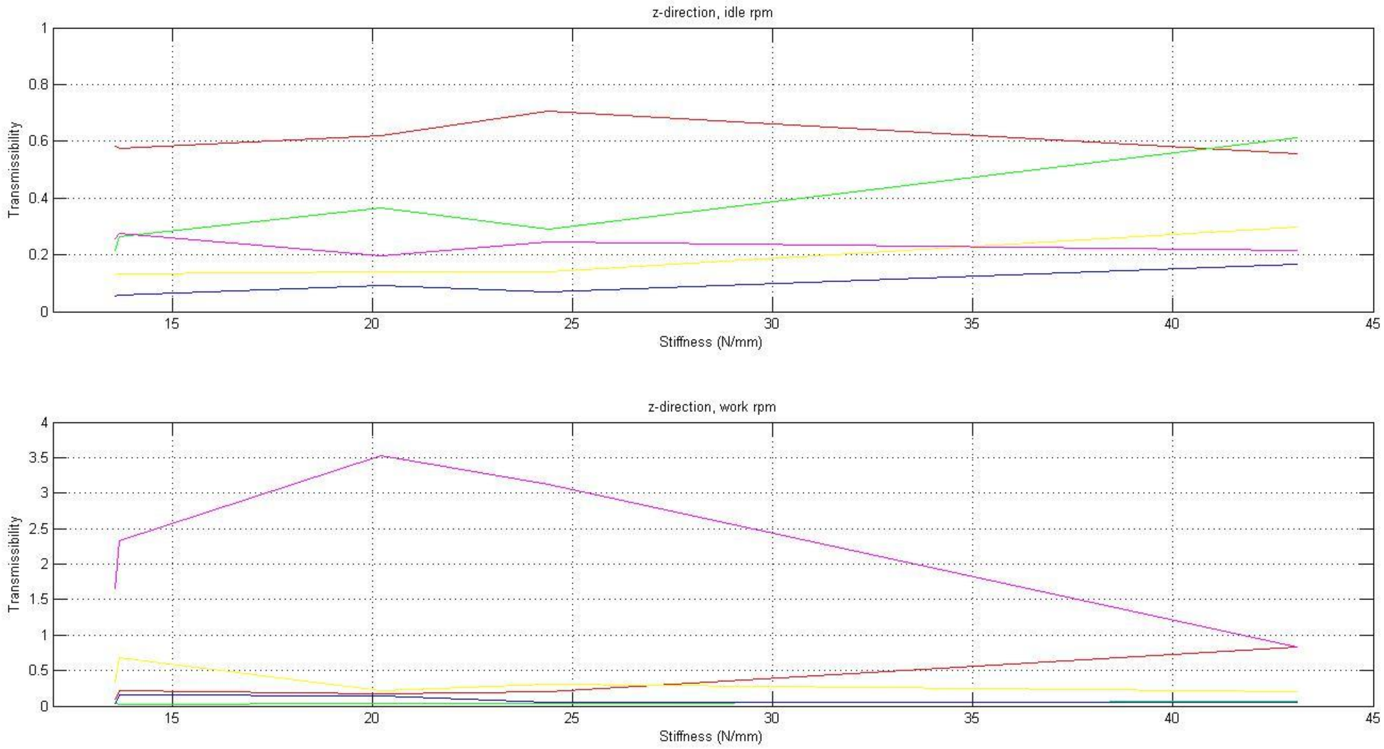

Figure 5.3c. Transfer function for the air intake – tank, z-direction.

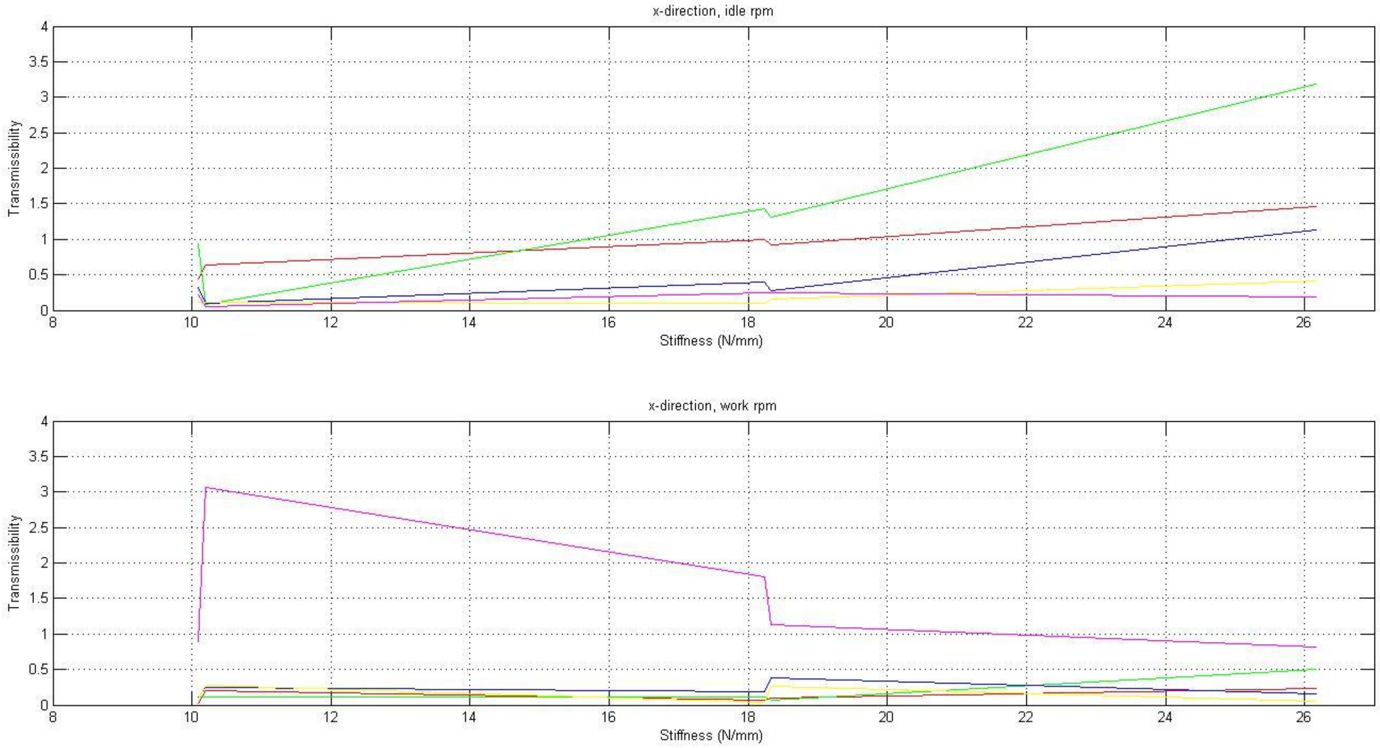

Figure 5.4a. Transfer function for the engine – handle, x-direction.

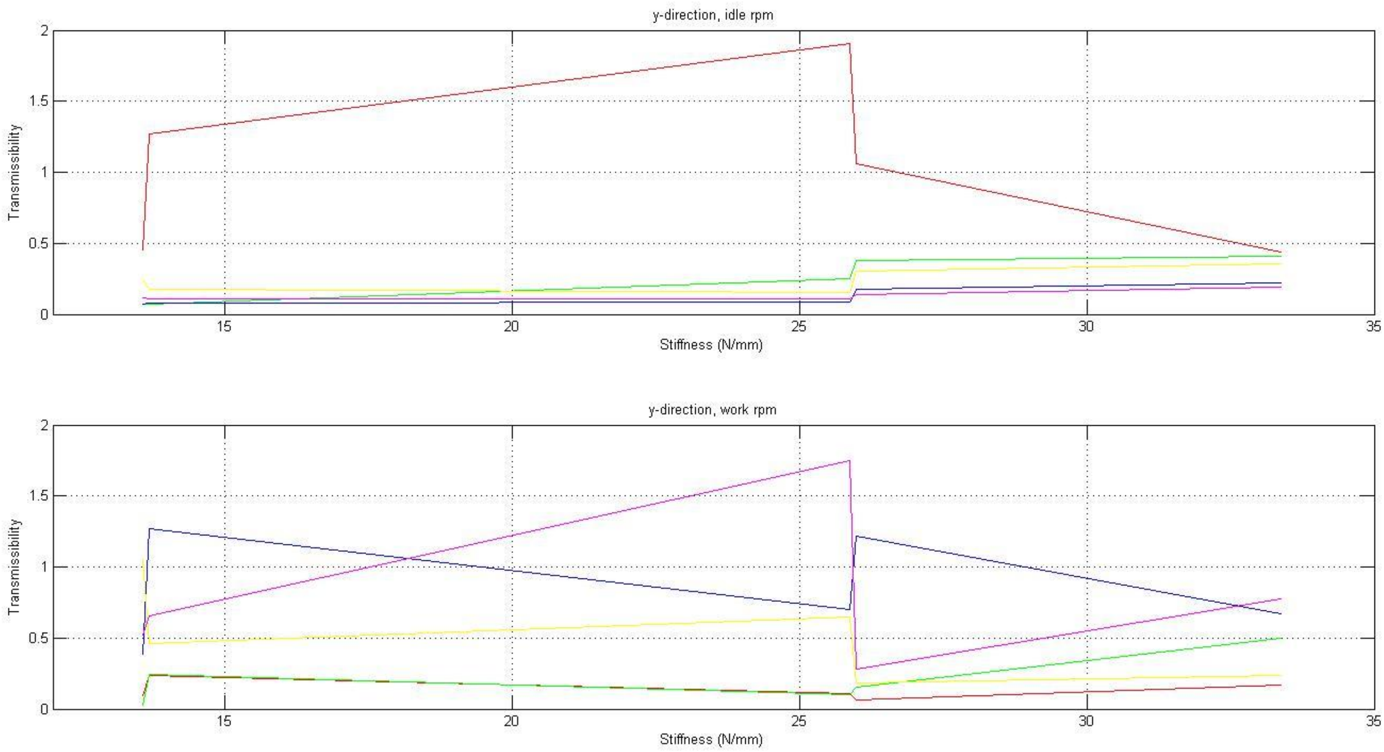

Figure 5.4b. Transfer function for the engine – handle, y-direction.

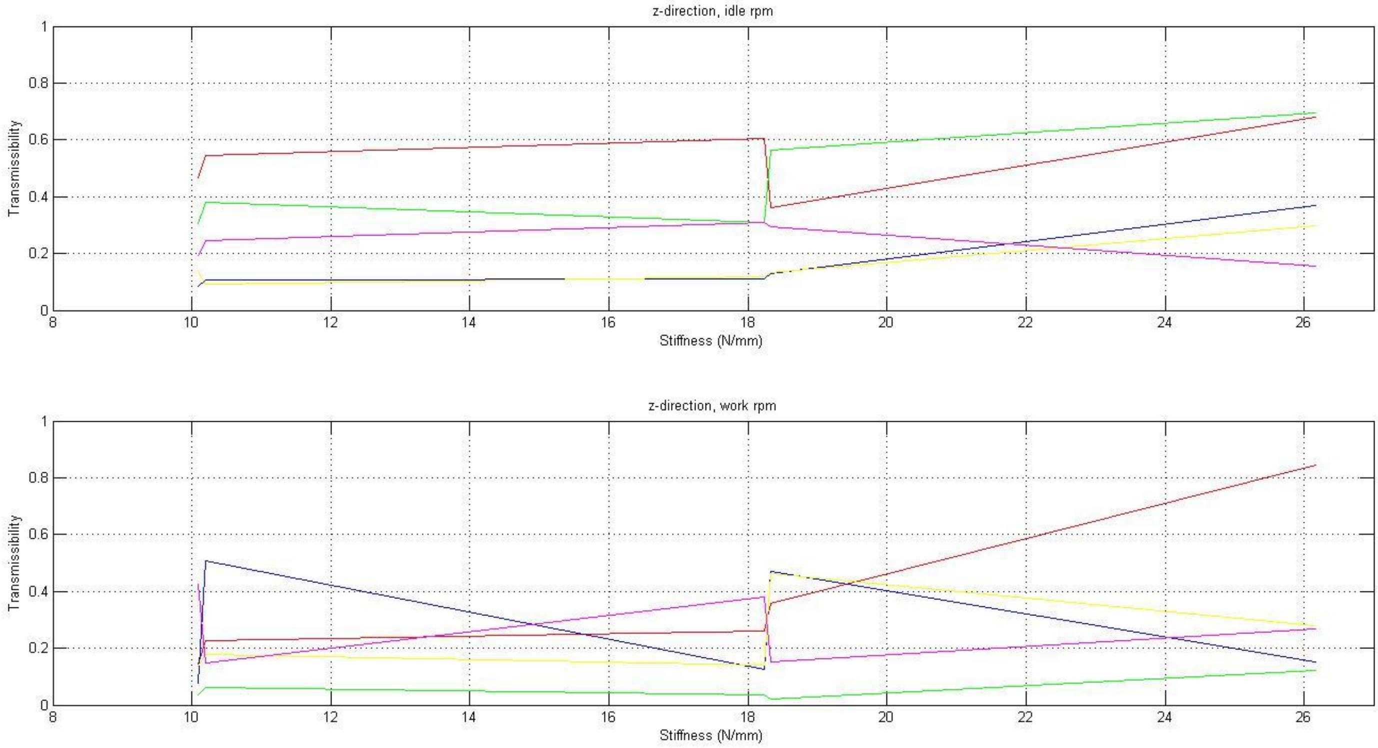

Figure 5.4c. Transfer function for the engine – handle, z-direction.

The different colours of the lines in the figures correspond to different frequency orders, according to table 5.5. The lines are made up from points which show the transmissibility at certain spring stiffness, the points come from measurements with different spring setups. The points from left to right show the sets in the following order for the cutting arm - feet and the air intake - tank: weak set, optimised case 1, optimised case 2, standard set and stiff set. For the engine - handle spring the order is: weak set, optimised case 1, standard set, optimised case 2 and stiff set

When two points on the same line are almost vertically placed relative one another it means that two measurements with the same spring stiffness has showed different results depending on the stiffness of the other springs in the set. These points are in reality perfectly vertical but have been purposely skewed slightly to avoid the many vertical lines blurring in to each other.

The transmissibility shows the percentage of vibrations that are transferred across the springs. A transmissibility higher than one means that the vibrations are amplified and is a clear indicator that something is bad, such as encountering a natural frequency. Transmissibility lower than one means that there is isolation. The closer to zero the better the isolation is.

5.3 Measuring Handle Vibrations

In order to relate the transfer of vibrations across the springs to the handle vibrations handheld measurements were performed for both the standard spring set and the new “optimised” cases. The results are shown in figure 5.5 to 5.7.

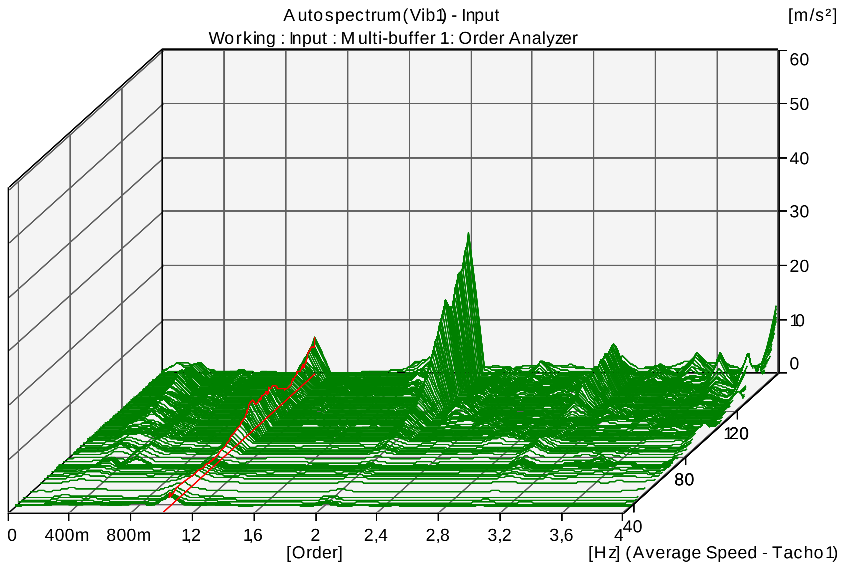

Figure 5.5a. Handle vibrations, forward handle with standard spring set, x-direction.

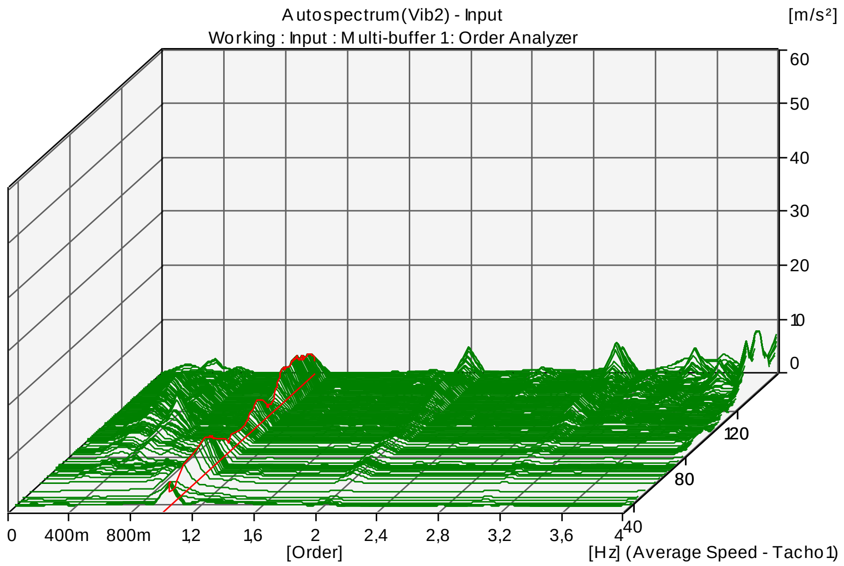

Figure 5.5b. Handle vibrations, forward handle with standard spring set, y-direction.

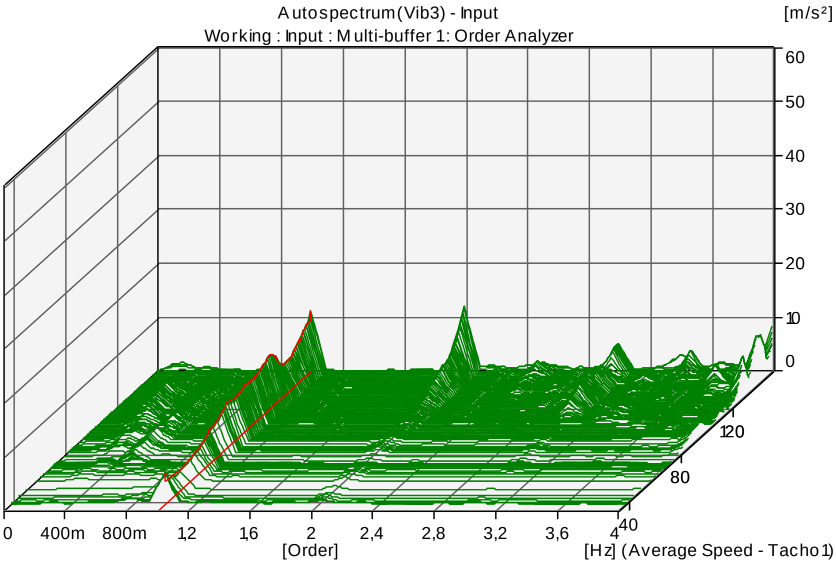

Figure 5.5c. Handle vibrations, forward handle with standard spring set, z-direction.

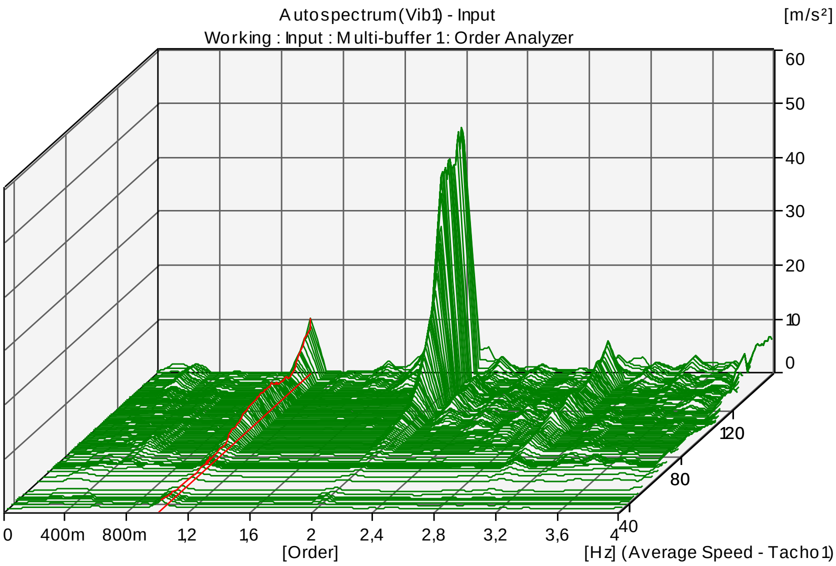

Figure 5.6a. Handle vibrations, forward handle with “optimised” case 1 set, x-direction.

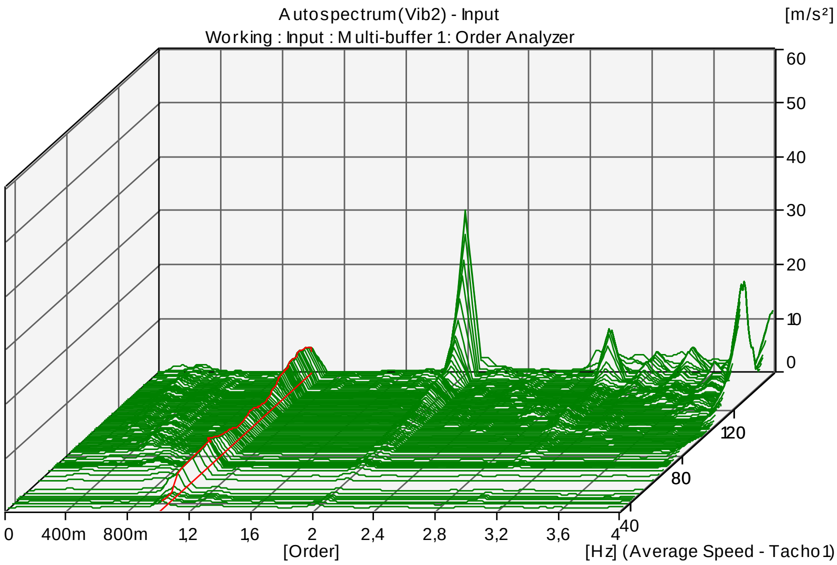

Figure 5.6b. Handle vibrations, forward handle with “optimised” case 1 set, y-direction.

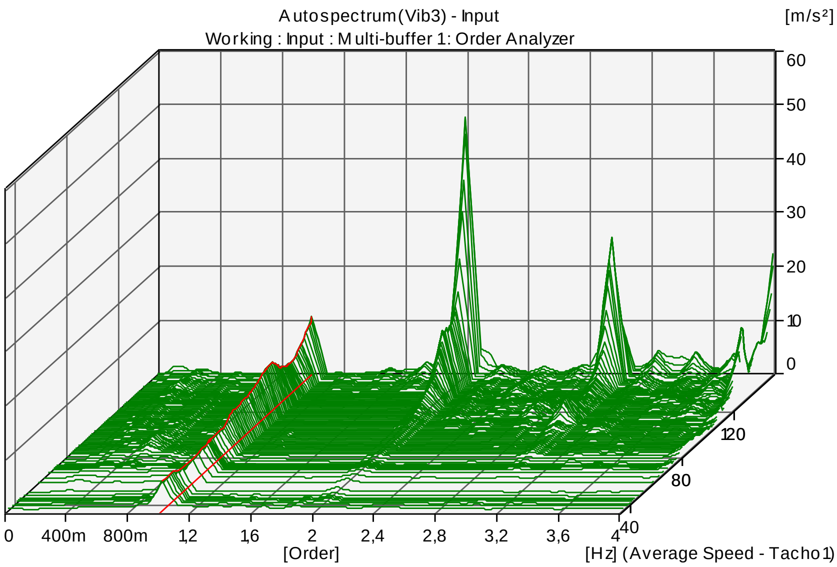

Figure 5.6c. Handle vibrations, forward handle with “optimised” case 1 set, z-direction.

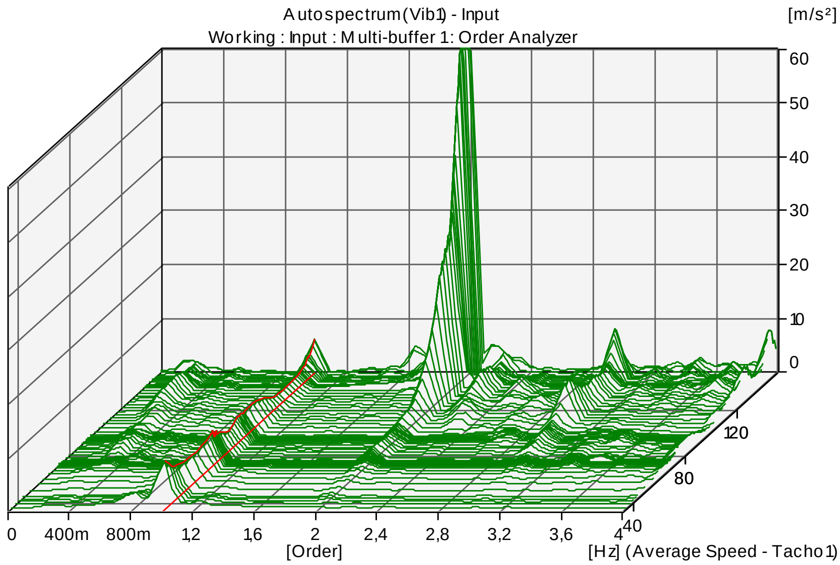

Figure 5.7a. Handle vibrations, forward handle with "optimised" case 2 set, x-direction.

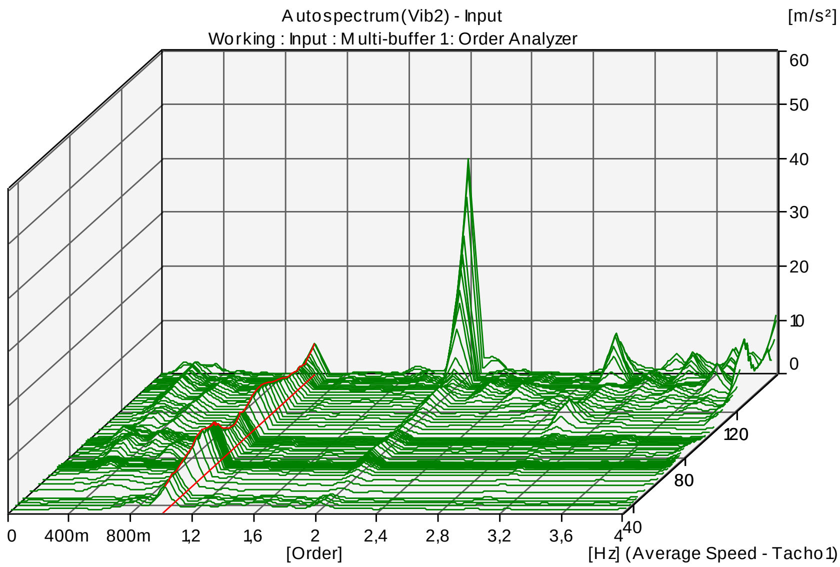

Figure 5.7b. Handle vibrations, forward handle with "optimised" case 2 set, y-direction.

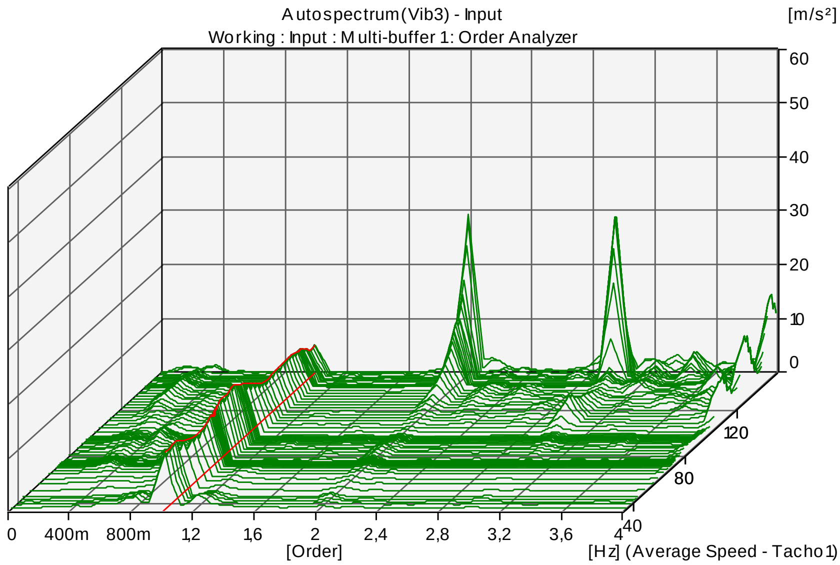

Figure 5.7c. Handle vibrations, forward handle with “optimised” case 2 set, z-direction.

The results show that the “optimised” cases have significantly higher handle vibrations at 9000 rpm than the standard set due to the second order of vibrations that seem to be transferred to a slightly higher degree across the springs at the engine - handle.

5.4 Unexpected Results

Throughout the analysis several strange and unexpected behaviours were detected. Most noticeable is when the transmissibility is close to or even above one, which can be clearly seen in the transfer function graphs. In many cases this is due to a natural frequency in the system, which can be observed by the peak moving with the changing of the springs. At the spring between the air intake and the tank a natural frequency of about 600 Hz is clearly visible in the fourth order in the z-direction around working speed (see figure 5.8). This peak is visible for all spring sets but moves slightly in frequency for the different springs.

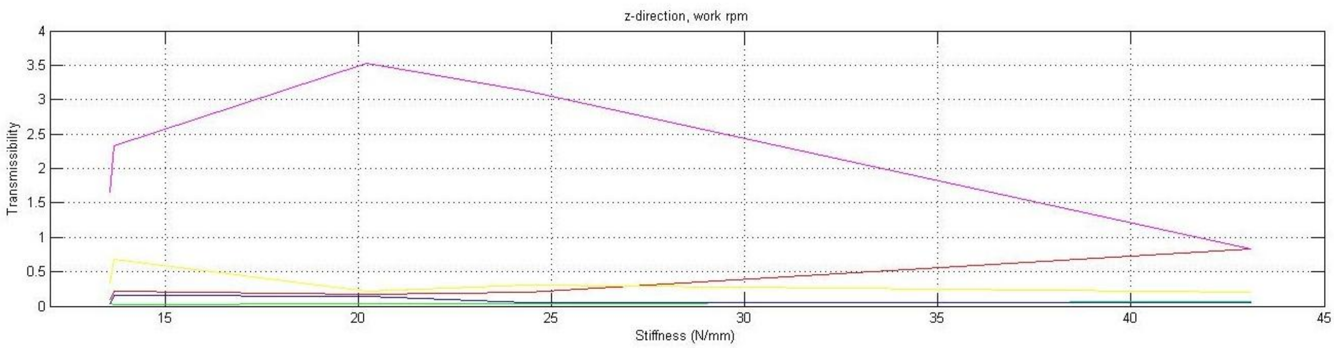

Figure 5.8. The fourth order (purple line) clearly has very high transmissibility for all springs.

It is important to note that all such behaviour is not necessarily visible in the transfer function graphs. At some measurement points the degree of transmissibility is exceptionally low but the amplitude of the vibrations that pass through are very high. With the standard spring set, running the K760 at work rpm it can be observed that only six percent of the second order vibrations in the y-direction are being transferred (see figure 5.9c) to the isolated part, but the amplitude of the vibrations on the isolating end of the spring is still very high, about 56 ms^{-2} (see figure 5.9b). This is simply because the amplitude of the vibrations acting on the non-isolated end of the spring is extremely high, about 982 ms^{-2} (see figure 5.9a).

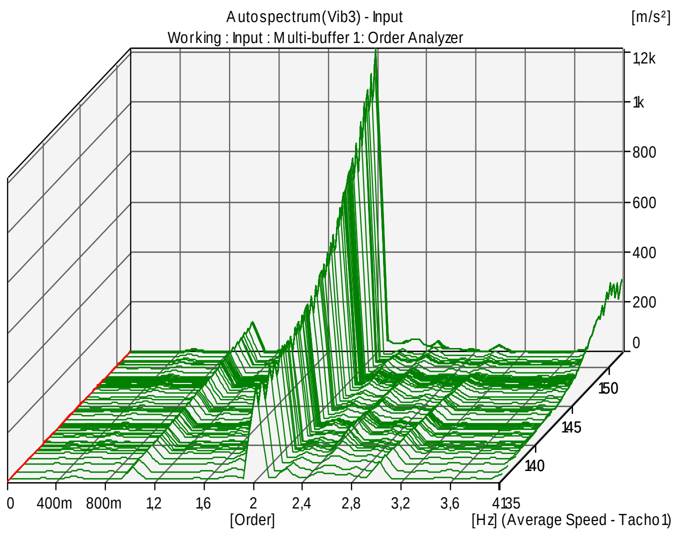

Figure 5.9a. Input vibrations.
The extremely high amplitude is clearly visible.

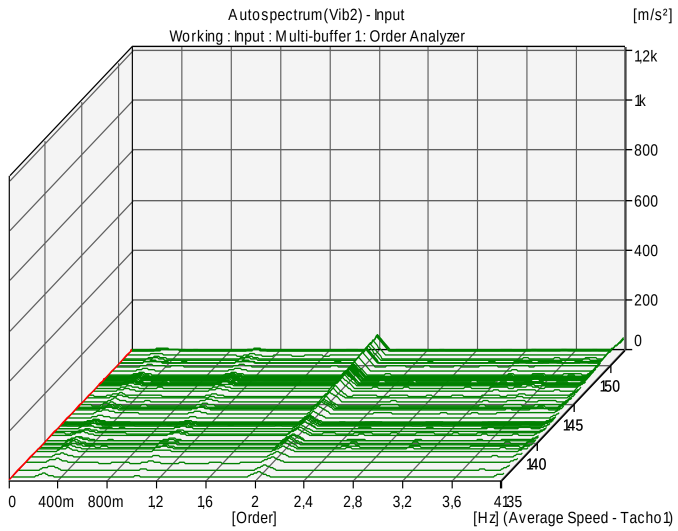

Figure 5.9b. Output vibrations.
The amplitude on the isolated side is 56 ms^{-2}.

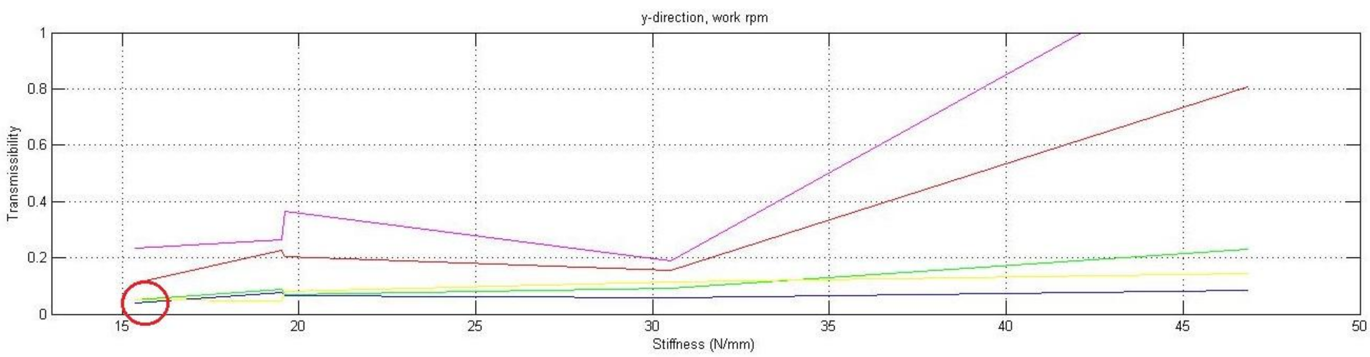

Figure 5.9c. The low transmissibility is apparent in the transfer function.

6
DISCUSSION

In this chapter the results presented in chapter 5 will be discussed further. If and how the results answer the research questions and how well the goals have been fulfilled will be the focus of this discussion, but it will also include possible sources of error in the measurements and unexpected results.

6.1 Answers to Research Questions

- *What is a reliable and sufficiently accurate way to measure the vibrations in a power cutter that allows the detection of effects of minor changes to the structure?*

  ⇒ It is possible to measure the vibrations in the power cutter with fairly high accuracy when using a test rig that replicates handheld measurements. Even with a properly set up test rig the accuracy will however be limited by the slight random and unpredictable variations of the vibrations produced by the machine and the operator. These variations are small relative the total measured vibrations but seem to produce an error margin of approximately 5-10% and are likely caused by a variety of mechanical deviations such as slight manufacturing imperfection or slightly differently torque applied to screws from each time the springs are changed.

  When comparing the graphs of the handheld measurement and the one performed in the new designed rig they coincide very similarly, see appendix A. For this method to work it is of outmost importance that the spring in the rig from which the power cutter is suspended is sufficiently weak relative the springs in the power cutter itself. A rough rule of thumb is to use a rig spring about 10% the stiffness of the machine's springs.

- *What are the causal relations of the transferring of vibrations from the cutting part of the cutter to the handles?*

  ⇒ The level of vibration isolation is dependent on the stiffness of the current spring as well as the stiffness of the total system. The relationship is complex but it is possible to use as a hint for predicting the amount of vibrations that will be transferred with different spring stiffness. It is however not possible to predict exactly how the handle vibrations will behave, most likely due to the natural frequencies of the handle part.

  The presented transfer functions describe this relationship between transmissibility and stiffness but this is only part of the necessary information. It is important to also look at the waterfall graphs presented in appendix D in order to get a holistic view and keep track of natural frequencies, unusually high amplitudes and noise in between the different orders.

- *Are there any natural frequencies of the power cutter that needs to be taken into consideration?*

  ⇒ The power cutter approaches an amplitude peak close to 51-52 Hz as can be seen in the graphs in appendix D . This is independent of which springs are used in the setup as this peaking phenomenon is visible in all short-span-rpm-sweep. There also seem to be a natural frequency in the handle part of the power cutter somewhere around 300 Hz which can be seen in the "optimised" case handle vibrations in figure 5.6 and 5.7. This is triggered when the second order vibrations are not sufficiently isolated at the higher rpms. Other than these frequencies it seems that most natural frequencies are at such high frequencies that they fall outside or far out in the periphery of the interesting frequency range.

- *How can the vibrations isolators be designed to further limit the felt vibrations of the power cutter without sacrificing the controllability of the machine?*

  ⇒ Weak springs have good isolation properties which should be utilised in a power cutter even though some rigidity has to be provided not to sacrifice the controllability of the machine. This requires balancing of spring stiffness. The acquired data and knowledge can be used as a hint and does provide some understanding on how the vibrations are transferred across the springs themselves. The data and knowledge developed during this thesis is not sufficient to accurately predict the precise effects of changes to the vibration isolators. The entire system is too complex and needs to be investigated further in order to fully understand the transfer of vibrations all the way to the handles.

6.2 Evaluation of Goal Fulfilment

- *Find a reproducible way of performing the measurements of vibrations that closely resembles handheld use*

  ⇒ By using the designed test rig and test method used reliable and reproducible results may be achieved. The rig is sufficiently accurate as it closely reproduces handheld measurements and does not seem to add any error. The rig yields both reproducible and repeatable measurement results in its current form, albeit difficult to use as it requires two operators to run tests. Further work could be done to the test rig to simplify the measurement process and eliminate the need for an operator to hold on to the machine during tests.

- *Provide a foundation of data and knowledge for future development work focused on the vibrations control in the K760 power cutter*

  ⇒ Data have been gathered and analysed to provide a basic knowledge foundation of how well the different springs isolate vibrations and what to consider when choosing springs. More work is however needed in order to properly relate the springs transfer functions to the resulting handle vibrations and ultimately measured hand-arm vibrations. The data gathered will serve as a solid base for calibrating the simulation model to resemble the real behaviour. In the long run this might be the most important contribution from the thesis as the simulation model will simplify future development and eliminate a large portion of the testing that is currently necessary.

- *Develop and test suggestions for ways to reduce the handle vibrations by making predictable improvements to the vibrations isolators according to the gathered information*

  ⇒ The results from the first series of tests converged and indicated what spring stiffness to use for further investigations. The "optimised cases" were tested but did not work as well as expected. Improvements could be seen in some cases, especially at idle rpm, but overall the transfer was very similar to the standard case. When measuring the handle vibrations however it was observed that the "optimised cases" produced significantly higher vibrations at work rpm. This shows that the system is coupled in a way not initially believed, where one spring is largely dependent on the other springs of the set. Conclusions and recommendations could be drawn from the analysis and even though the optimisation failed a lot was learned from the process and a great deal of useful data was gathered.

- *Map the relationship between transferring of vibrations and the stiffness of the springs*

  ⇒ The relationship between the vibration isolation and the stiffness of the springs have been mapped in transfer functions. Depending on the total stiffness of the set the springs isolate differently. For example in figure 5.1b the same stiffness spring transfers about 25% in a weaker set and 225% in a stiffer set. Therefore they are not exact but can be used as a hint of how much the springs will isolate.

6.3 Sources of Possible Errors

There are several possible causes of error in the results gathered through the vibration measurements. The most important of these will be discussed here.

Problems with weak springs

It is known that weak springs isolate vibrations well, but weak springs also have their drawbacks. It is more likely that they get fully compressed which makes them transfer all vibrations they are exposed to which makes the power cutter virtually inoperable. Weak springs may also mean that the operator does not have the same level of control of the power cutter due to the cutting arm moving too easily in relation to the handles. When analysing the result from the initial series of tests the weak springs proved to be good isolators but it also presents some unpredictable and random behaviour at some rpms which did not arise in measurements of stiffer springs. This indicates that something is not working properly, perhaps some spring might be fully compressed or it might enter a natural frequency. Therefore the choice of springs to use for the two optimised spring sets did not consist of only weak springs even though the weak set generally showed the best results.

Problems with stiff springs

Another phenomenon that can clearly be seen in the transfer function graphs is the inability of the stiff springs to isolate the lower frequencies (see figure 6.1 for an example). This can be seen for most measurements with stiff springs and at the spring between the air intake and the tank in the x-direction as much as 100% transmissibility can be observed for both the 0.5:th and 1:st order at idle rpm for the stiffest spring but not for the other springs. It is possible that this extremely high transmissibility also has other causes as it is unusually high, but the stiff springs certainly are part of the problem.

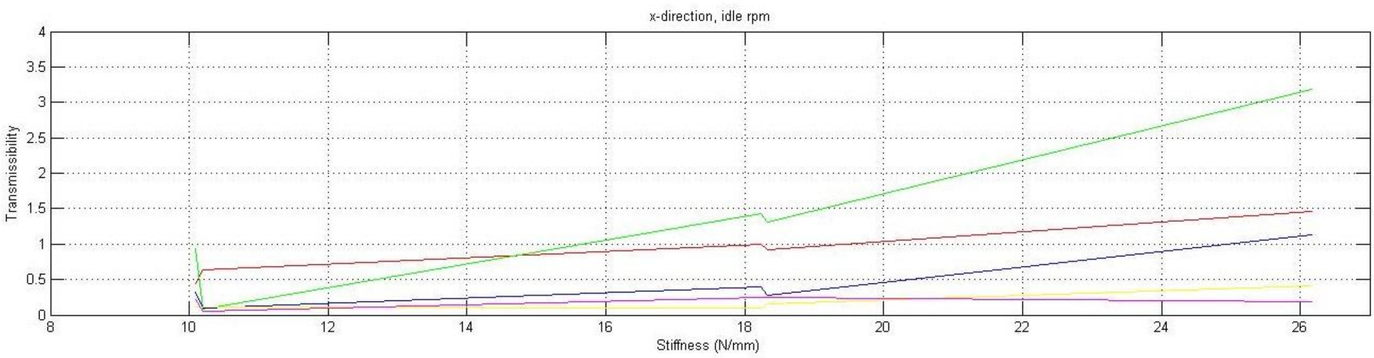

Figure 6.1. The stiff spring (rightmost points) clearly has poor low frequency characteristics.

Problems with the “optimised” cases

The handle structure seems to enter a natural frequency at work rpm while having very weak springs. This is shown when comparing the handheld measurements of the standard setup with the new weaker setups, see figure 5.6 and 5.7. When examining the transferring through the springs nothing indicates that there is an unusually high transmissibility of vibrations overall, but the second order vibrations seem to be transferred to a higher degree at work rpm. It is also possible that the total stiffness of the structure is lowered enough with these sets of springs for the handle to enter a natural frequency that coincides with the high rpm second order vibrations at about 300 Hz.

Lateral stiffness affects both x- and z-direction

If the spring is not deflected the y-direction are influenced by the axial stiffness while the x- and z-direction are both dependent on the lateral stiffness. The power cutter is suspended such that the lateral stiffness normally supports most of the weight, both when cutting in the wall and the floor. If only considering cutting in a wall the lateral stiffness also work in the x-direction where the weight influence is much lower but it still has the same stiffness. Due to the big difference in where the weight acts it is hard to optimise the isolation level in both z- and x-direction at the same time. If optimising in the z-direction it will lead to a high degree of transferring of vibrations in the x-direction and if optimising in x-direction the springs will be too weak to support the weight of the engine and it will probably be fully compressed which makes all vibrations in the z-direction pass through the spring.

To optimise the system for a certain cutting position it should be designed such that it has different stiffness for all directions since the weight and the amplitude of the vibrations acting in the different directions are different. But if it has to be able to work well in many cutting positions compromises has to be made and one has to consider that the most important cutting position has to be balanced towards how much vibrations are transferred in the different positions. The measurements have only been performed with the power cutter positioned horizontal so the results only hold true when considering the horizontal optimisation.

Coupled System

The tests on the “optimised” cases were executed with springs having stiffness correlating to the springs which performed best in all respective directions in first series of tests. The new set of springs isolated less of the vibrations than expected and resulted in higher handle vibrations. The different springs are acting as a complex system and the vibration isolators cannot be optimised with precision by only considering them as three independent systems as previously thought, though the individual optimisation gives some hint of what is actually transferred over the springs.

The relation between the stiffness and the directions it is supposed work in are not linear for this system. To be able to draw linear relationships the springs would have to be placed in the same directions and on the same axle. As soon as one spring deflects it influences the direction the stiffness act. The less stiff the setup is the more the springs deflect and the more complex the direction of the stiffness gets. The springs in the K760 are placed in the same direction but with an offset and when using less stiff springs the system becomes more clearly coupled. Changing the stiffness of one spring influences the transferring of vibrations over the rest of the springs in the setup. It is possible to measure how large part of the total mass of the cutter that acts on each spring, though this is changed when changing the stiffness of the springs and require a lot of work. This mass may also change with frequency adding even more complexity to the task of investigating it.

Loss of signal from the engine accelerometer

When measuring on the spring between the engine and the handles the signal from the accelerometer placed on the engine heat sink would be lost in the upper half of the rpm sweep. The exact same phenomenon was encountered on other measurements on a few occasions but was then proven to be due to a faulty signal wire. This did not seem to be the case with the measurements on the engine as changing the wire would not solve the issue. Another theory would be that the temperature is so high that the accelerometer fails, even though the temperature measurements suggested this would not occur. But this reason also appears unlikely as the accelerometer worked properly except for the higher rpms regardless of how long it was exposed to the heat.

Since the cause of the problem could not be found the results were accepted as they were, with a sparse amount of data for the higher rpms. The connection could be maintained better by tensioning the wire by hand while running the tests but it is probable that this adds some distortion the results. Therefore it is suggested that all results for rpms higher than about 6000 for this accelerometer are viewed critically and that a greater margin of error is applied should these results be used.

Variations in accelerometer placement

A possible source of error in the results is the slight variations that most likely have occurred in how the accelerometers were placed from test to test. With the help of the mounting brackets the placement was standardised and the variation minimised but because the brackets in many cases were glued to the heads of the spring attachment screws they had to be removed and reattached whenever the spring set was changed. It is likely that they were not attached in exactly the same way each and every time and these slight displacements and angle differences would produce an error in the results.

In the case of the spring between the cutting arm and the feet the mounting brackets on both ends could not be attached directly in line with the spring axis due to physical limitations from the machines structure. These offsets produce a systematic error meaning that the results can still be properly compared to one another but perhaps not with the results from the other spring positions.

It was also found that when testing the two “optimised” cases it was not possible to mount the accelerometer on the same position on the feet of the K760 due to a larger diameter of the spring used in this position. Instead the accelerometer had to be attached on the other side of the mounting bracket, resulting in a measurement error.

The human factor

All the measurements were performed with the same operator holding and triggering the power cutter in order to reduce variation and the sweeps were always performed until two similar graphs were achieved. The operator also attempted to press the trigger slowly and steadily when moving over sensitive areas but despite these efforts it is likely that the human factor might have influenced the results since it is hard to perform the sweeps slow enough and to know when to be more cautious. It is also difficult to guarantee that the sweeps were run in the same exact manner each time. It is believed that, together with mechanical imperfections in the machine, this provides the bulk of the error and is a much larger factor than the other error sources discussed.

7
CONCLUSIONS

When measuring the handle vibration in the developed test rig the results closely resemble those from the handheld measurements. The test rig makes it possible to perform repeatable and reproducible measurements with reliable results and may be used for more accurately calibrating the simulation model to the real behaviour of the power cutter. It is of great importance to use the test rig in its new design in future tests since the previous design had a natural frequency completely distorting the results with random behaviour that is hard or even impossible to interpret. The current drawback of the new test rig is that it requires two operators, but this is something that can be solved with either more work on mass loading the handles of the power cutter or by implementing for example a start/stop pedal control for the measurement equipment.

The transfer functions may be used as a hint of how well the different springs will isolate vibrations and what stiffness to use for the different directions and spring positions. The transfer functions should only be used when considering stiffness that is close to stiffness already performed measurements on. The transferring mostly depends on what springs the setup consists of as well as how large the amplitude of the vibrations the springs are exposed to are at the different spring positions. If the transfer functions are going to work on a more general level there is a need to perform more measurements since it consists of too few measuring points as in its current form. The transfer functions are also limited since they only consider the amplitude of the vibrations at the engine rpms of 2700 and 9000. To be able to ensure that no natural frequencies or strange behaviours are occurring the measurements should be examined visually in the form of the waterfall graphs as well.

The two “optimised” cases tests showed good vibration isolation through the springs, although not much different from the standard spring set. But when looking at the handle vibration from the handheld measurement both “optimised” cases exhibit a peak in the second order at 9000 rpm. This vibration peak is not present in the result of the spring measurements, although the second order vibrations are transferred slightly more at the engine - handle spring, and therefore has to be due to a natural frequency of the handle structure. This may be explained since the total stiffness of the power cutter decreases when using less stiff setups of springs.

The springs at different positions are not possible to consider as three independent systems. They influence each other and act as a complex system. Each spring correspond to a subsystem that changes when changing the stiffness of the springs as well as when tilting the power cutter in different angles and directions. The springs often perform very differently at idle and work rpm. When trying to optimise the vibration isolation compromises have to be done considering cutting position, what directions to isolate from vibrations and if optimising for idle or work rpm.

The new designed test rig and the method used for measuring vibrations can be used to assess how changes to the power cutter have influenced the transmissibility. However the results may only be used for assessing changes already made and as a hint for performing further changes but not for accurately predicting what changes should be performed to get a certain result.

8
RECOMMENDATIONS

In this chapter recommendations for how to use the results in this thesis and for future work that is interesting to look into.

8.1 Measurement Methodology

Perhaps the most important recommendation that this thesis results in is regarding the measurement methodology and the test rig. For the test rig it is recommended to use a weak spring to suspend the power cutter. The spring of the test rig should have a maximum stiffness corresponding to 1/10 of the total stiffness of the power cutters suspension.

The weight and stiffness of the arm and hand must be included in the system to simulate handheld measurements. The easiest and most straight forward way to simulate this is by holding the power cutter in place using the hands and arms of an operator while supporting the majority of the machines weight using the mentioned weak spring suspension. It is also recommended that the power cutter is supplied with gasoline from an external tank and that the space of the tank is filled with a closed bag filled with water that can simulate the same constant gasoline content in the tank for all measurements.

8.2 Results Analysis

When analysing the data from measurements it is important to maintain a holistic view so that worrying natural frequencies or unusual behaviours are not missed. All different graphs such as the waterfall and transfer function graphs should be used in combination in order to extract as much useful information from tests as possible.

It is also important to keep in mind that the system is complex and that springs can not necessarily be considered on their own, at least in terms of optimising the vibration control. It is hence recommended that the results presented in this report are only used as a hint to guide future research and development and as a foundation for building further knowledge.

8.3 Changes to the Vibration Isolators

Based on the findings of this thesis it is not possible to recommend specific design changes for the vibration control. A more general recommendation would be to decide what to optimise the vibration control for as it seems compromises will have to be made, for example between low and high frequency characteristics or different orientations of the power cutter. These limitations may be possible to avoid by changing the vibration control system to a more advanced one, but this would also have its drawbacks and would require a lot of extra work.

8.4 Future Work

It is recommended that further effort is put into developing the test rig so that tests can be run with only one operator. To achieve this it could be of interest to investigate some system of mass loading of the handles in such a way that the stiffness and mass of an operator’s hands are simulated accurately. The rubber padding investigated in this thesis did not work well for this but it is possible that other material or even just other attachment methods will work better.

It is also recommended that the most important measurements from previous test series are redone using the new test rig design in order to guarantee that the distortion from the old test rig do not affect them in any way. It is especially important that this is done for all measurements used to calibrate the simulation model so that future simulation results can be guaranteed a high degree of reliability.

It is recommended to perform more measurements with different spring stiffness and setups to make the transfer functions more usable and general for a wider range of stiffness.

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A

Reference Measurements

Before the actual testing was started a series of reference measurements had to be carried out. These tests were performed using a three-axis accelerometer mounted to the forward handle of the power cutter. The goal with these measurements were to compare the test rig to handheld measurement and redesign and fine tune the rig until it gave very similar results to the handheld ones. Ideally the tests should be done handheld but this is simply too cumbersome considering the amount of tests needed. Furthermore, the existing ventilation equipment is not as effective for handheld measurements as it is for testing in the rig, increasing the risk of carbon monoxide poisoning and other hazardous effects of exposure to the exhaust gases.

Initially the existing test rig was compared to handheld measurements. This showed a natural frequency affecting the results in the rig, distorting them so that they were not similar to handheld results, compare figure A1 and A4. Following this the test rig was first made less stiff and then extremely rigid in order to investigate how to best remove the natural frequency from the frequency range that is interesting for this thesis, about 22.5 Hz to 600 Hz.

The rigid test rig did move the natural frequency high enough but also added a lot of noise in between the orders, compare figure A1 and A2. The weak rig suspension moved the natural frequency lower and it was decided that a weak spring was the best approach in order to avoid the noise of the rigid suspension. The results from using a very weak spring can be seen in figure A3.

When comparing figure A1 and A3 it is clear that the results from the weak spring still does not resemble handheld measuring good enough. This led to experimenting with mass loading the handles. Different materials and masses were tested and the results can be seen in figure A6-A8. Unfortunately none of the tested masses or materials accurately resembles the mass and stiffness that a hand would add and it was ultimately decided to have an operator hold the handles for the actual tests. Both gripping the handles firmly and relaxed was tested, see figure A9 and A10, and the relaxed method was selected to be used. This way the machines weight is supported in the rig by the spring suspension and the operator only provides mass to the handles.

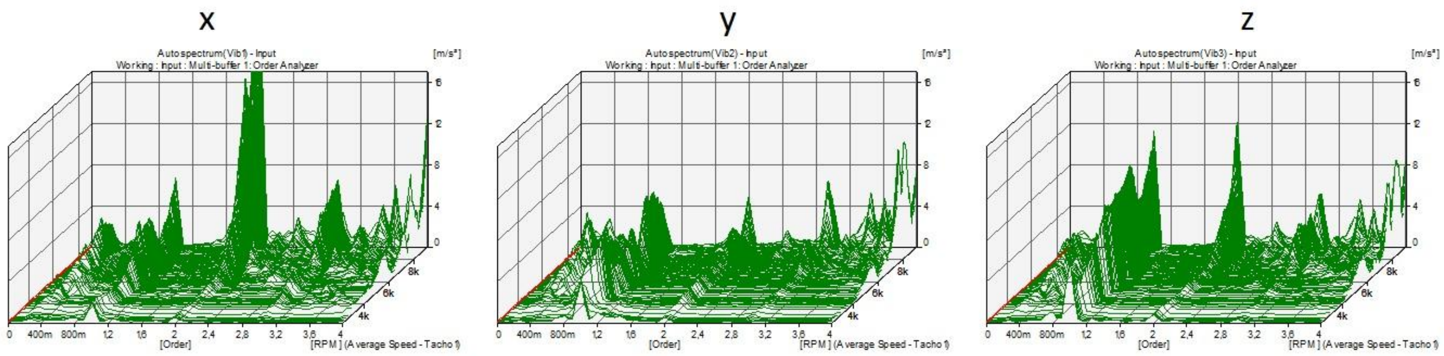

Figure A1. Handheld measurements.

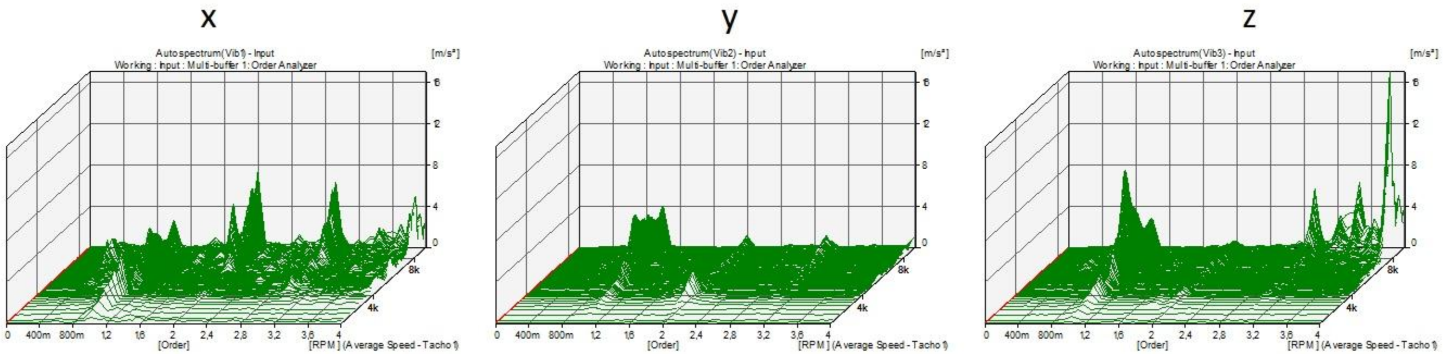

Figure A2. Rigid test rig suspension.

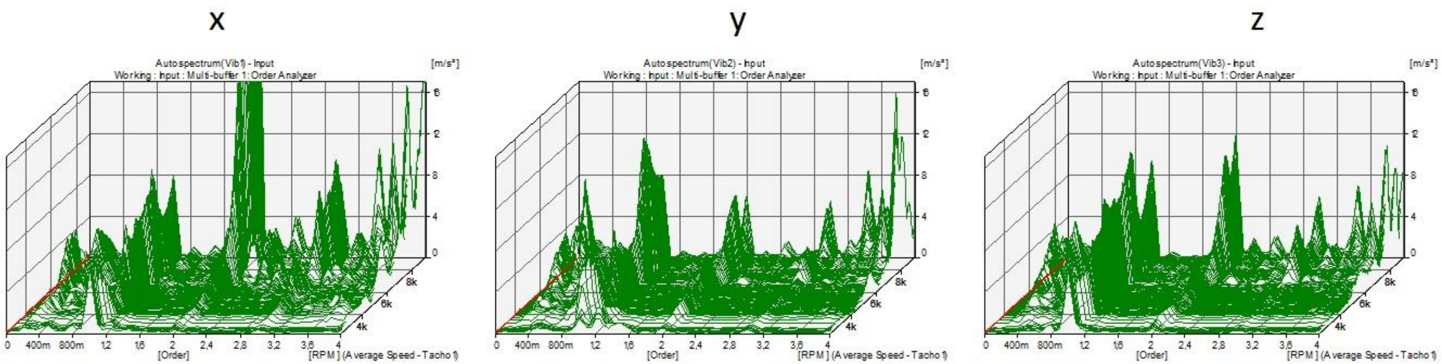

Figure A3. Weak spring suspension.

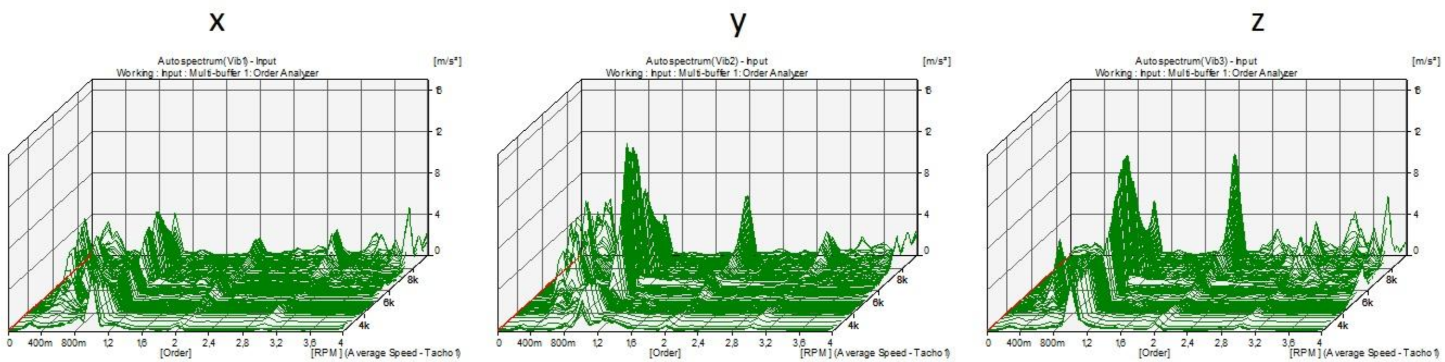

Figure A4. Original test rig.

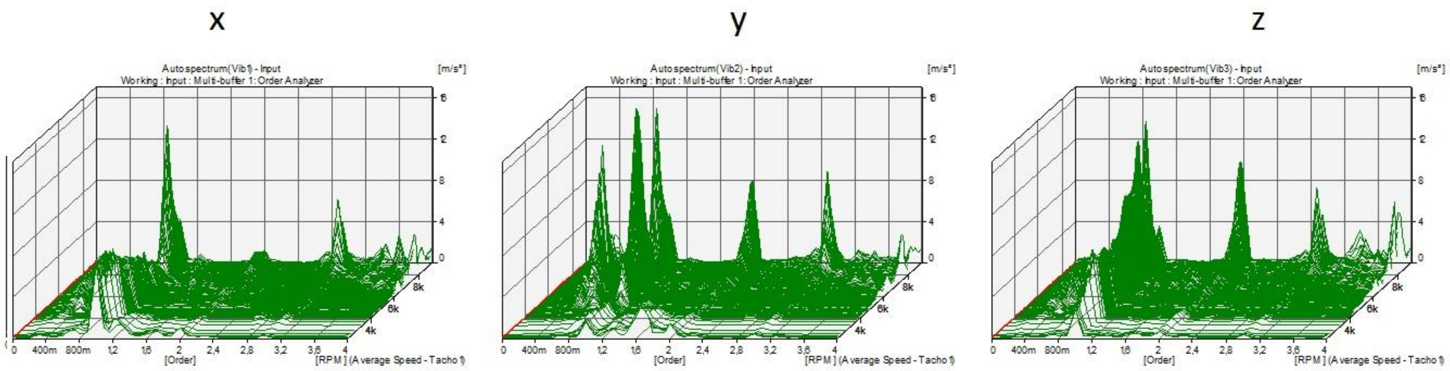

Figure A5. Stiff rig suspension.

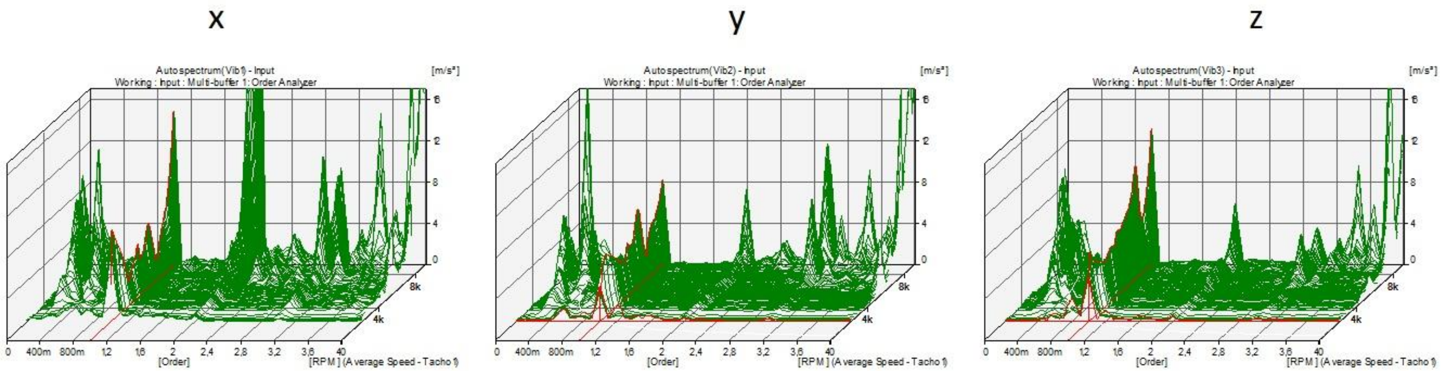

Figure A6. Mass loading the handles using 200 g rubber padding.

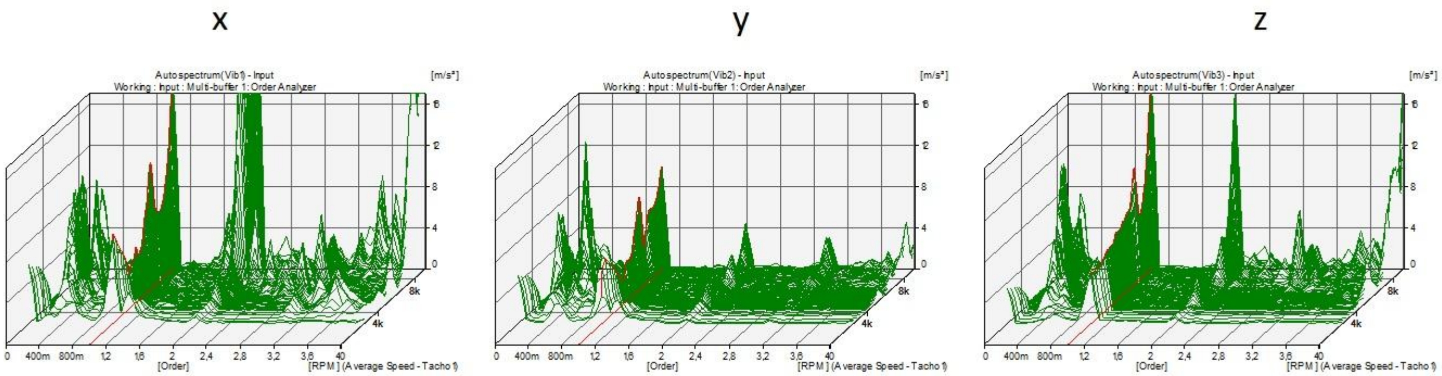

Figure A7. Mass loading the handles using 200 g steel weights.

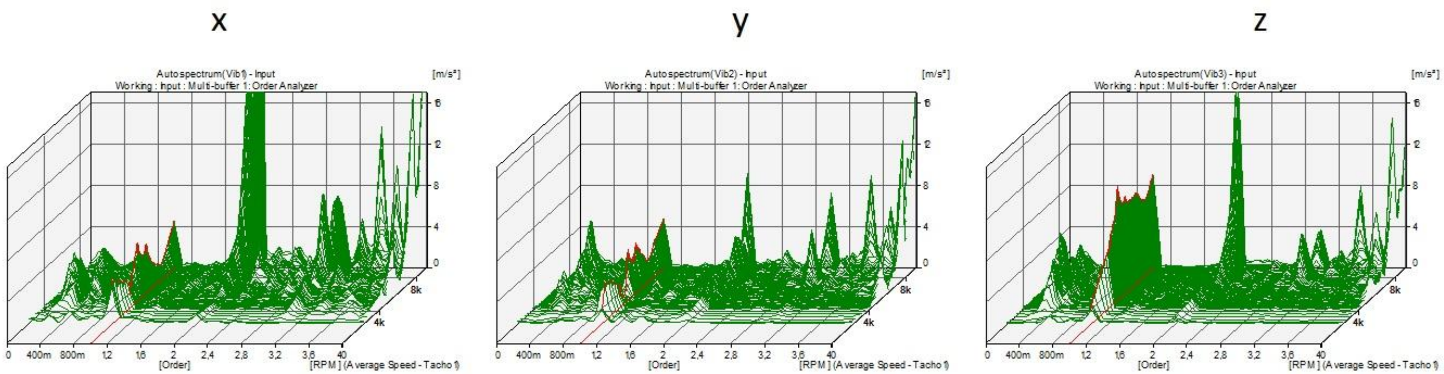

Figure A8. Mass loading the handles using 862 g rubber padding.

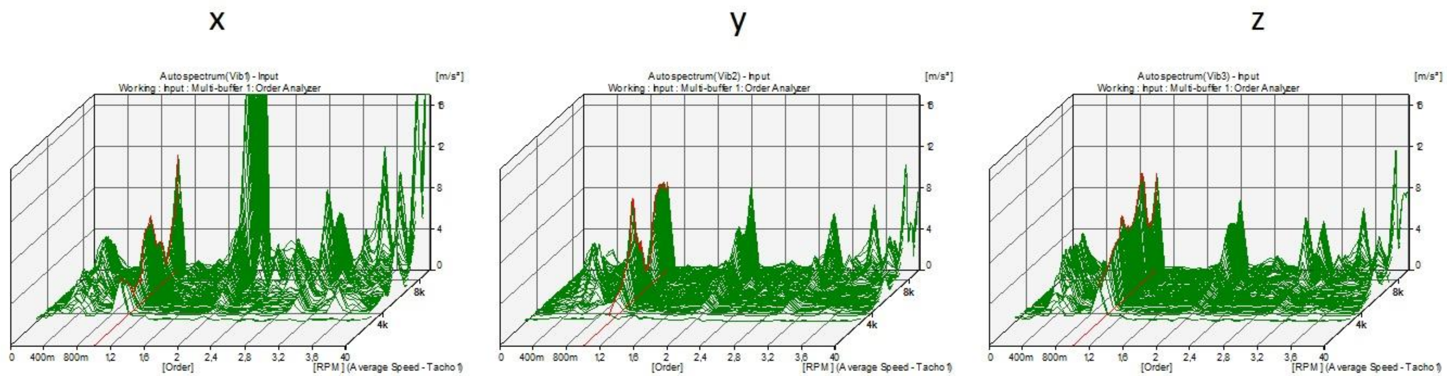

Figure A9. Operator firmly holding the handles of the power cutter in the test rig

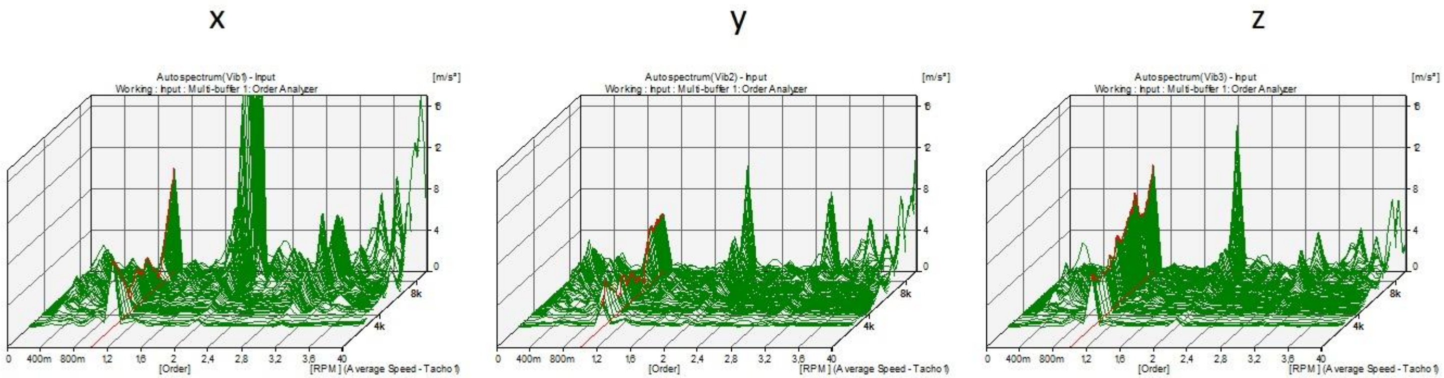

Figure A10. Operator holding the handles of the power cutter with a relaxed grip in the test rig.

B

Accelerometer Placement

The accelerometers are placed as close to the centre of both ends of the springs as possible. It is important that they are placed as close to the centre as possible to be able to measure the transferring of vibrations with high accuracy. To keep track of all data a 3-axis coordinate system was defined relative the power cutter handle. Measurements are performed in the three directions of the coordinate system and to be able to place the accelerometer such that it measures accurate in all three directions at all springs small mounting brackets were manufactured to hold the accelerometer. The mounting brackets are small cubes with perpendicular sides which is fastened at the ends of the springs and oriented in the same manner for all springs. The mounting brackets are glued to the structure to hold its position. To be able to access the vicinity at the ends of the springs some modifications had to be performed to the power cutter.

At the spring connected by the air intake and the tank a hole was made in the tank to easily access the end of the spring. The gasoline was then entered by a hose connected to an external tank. To be able to fasten the mounting bracket in the right orientation and close to one end of the spring at the “cutting arm - feet” - spring the foot had to be milled down. The mounting cube fastened at the motor had to be special manufactured to be able to be fastened with the orientation of the 3-axis coordinate system. The accelerometers are fastened into the mounting cubes either by using magnets or by screwing it onto a mounting plate with a threaded pin. In this appendix pictures of the mounting brackets are displayed when fastened at the end of all springs.

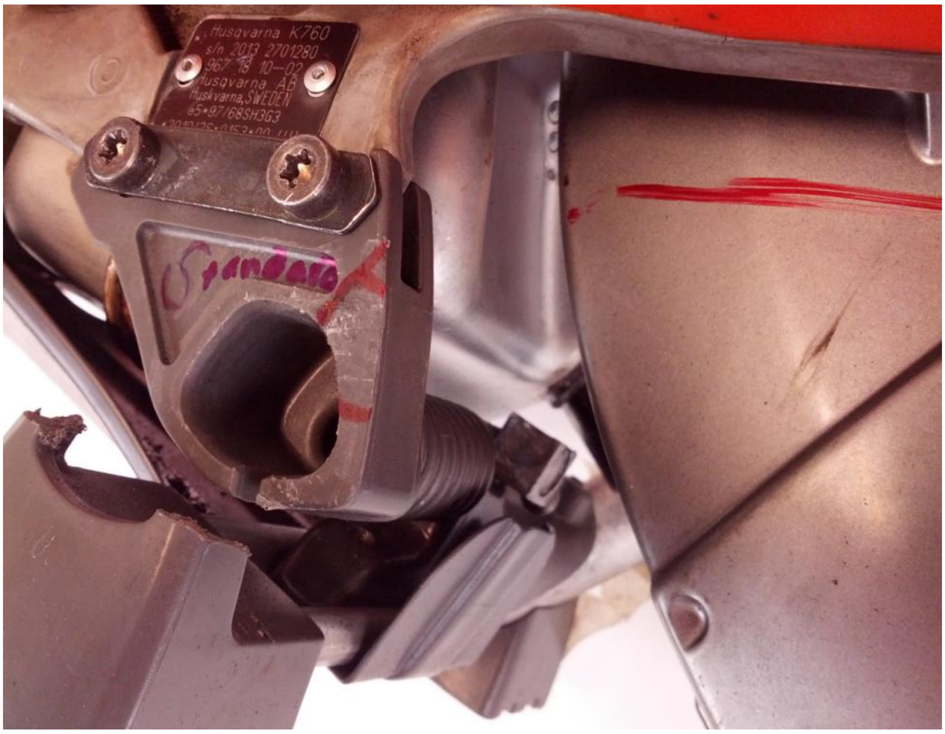

Figure B1. The mounting bracket at the feet of the power cutter. The red marking in the foreground is an indication for where to glue the mounting bracket to the cutting arm.

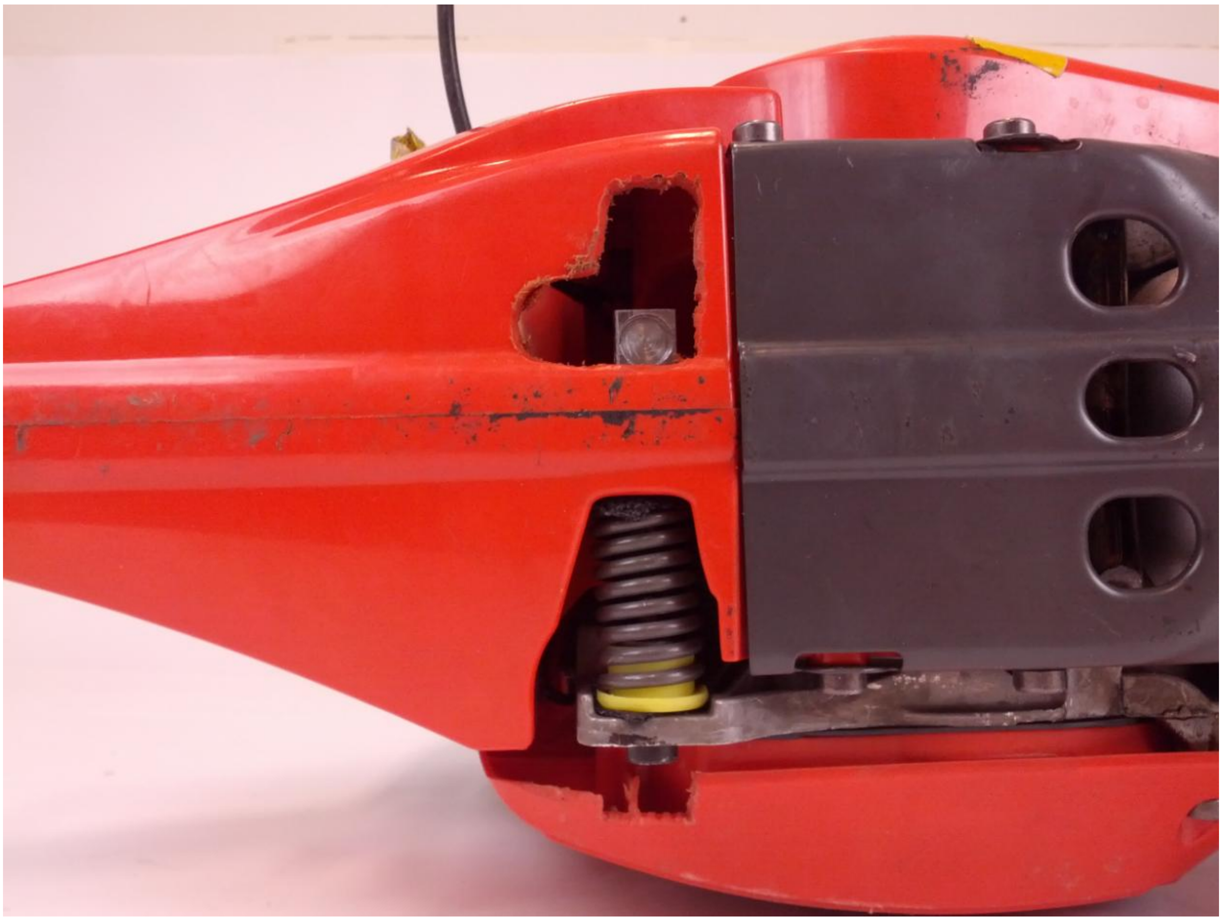

Figure B2. The mounting bracket inside the tank. At the bottom of the picture plastic has been removed to accommodate for the mounting bracket on the air intake.

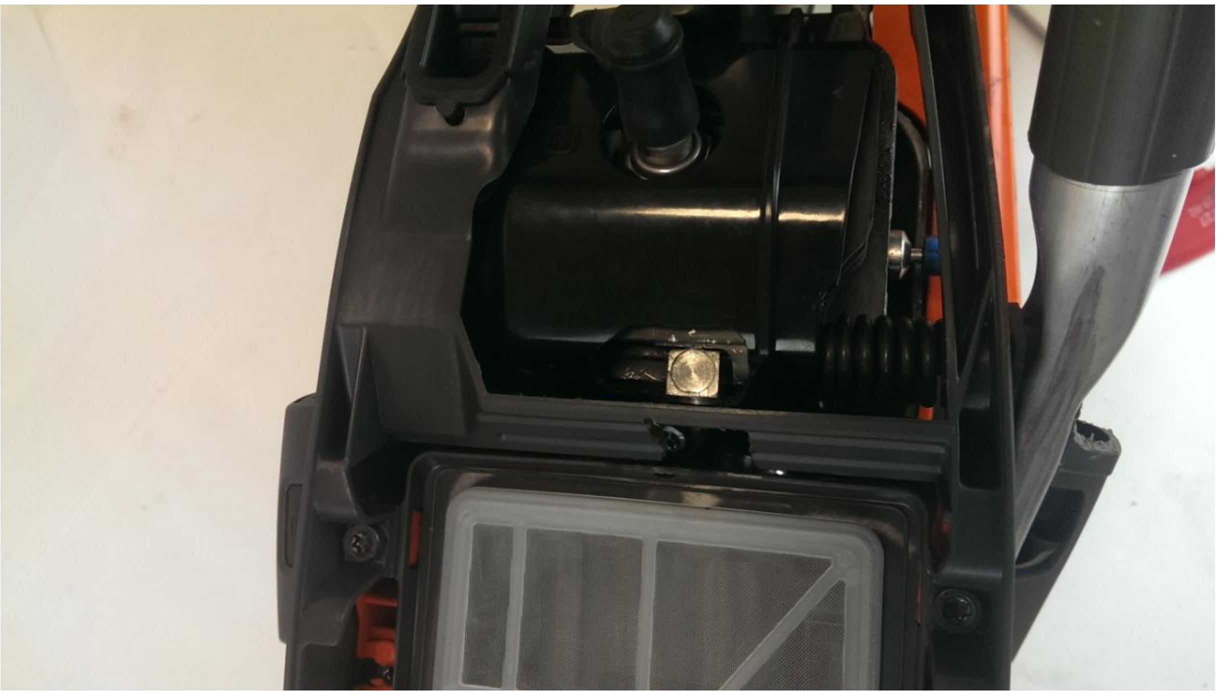

Figure B3. The mounting bracket on the engine.

Figure B4. The mounting bracket for the three-axis accelerometer on the forward handle.

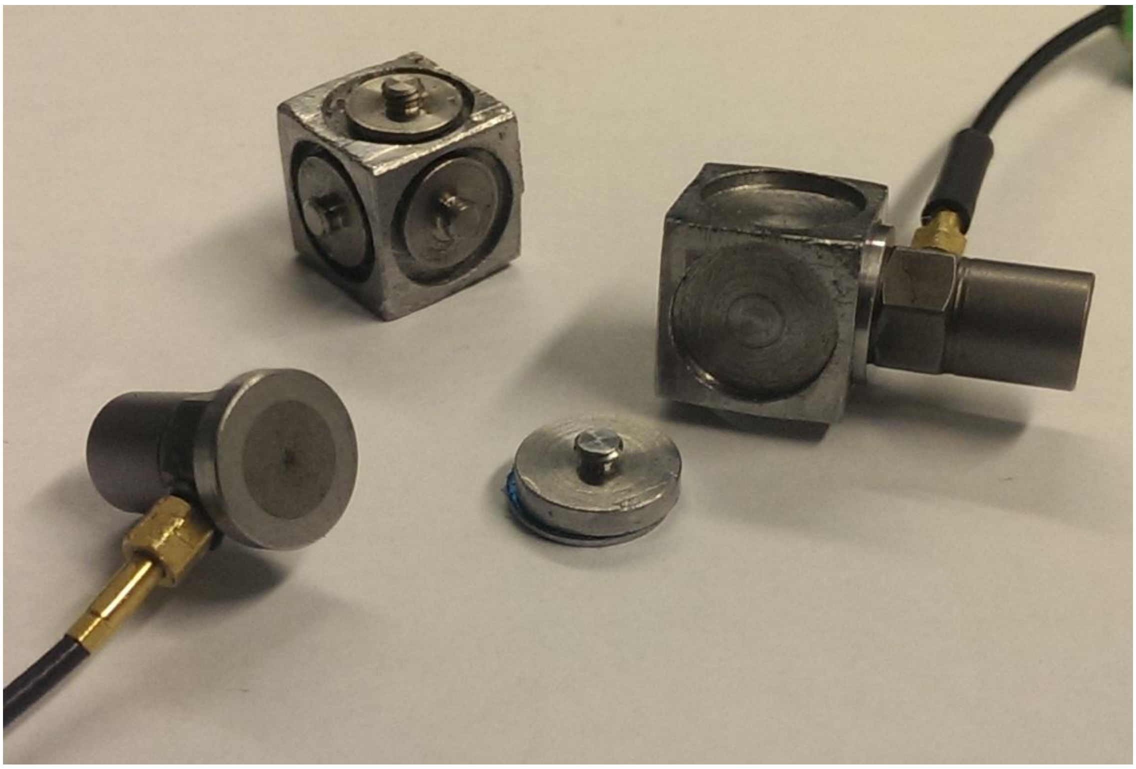
Figure B5. The different types of mounting brackets and the mounting magnets along with single-axis accelerometers.

C
Temperature Measurements

The accelerometers are specified to work up to a maximum of 125°C, beyond that the accuracy of the measurement can no longer be guaranteed. For most of the measurements of this investigation this is far above the temperature range the accelerometers will be exposed to, but because one of the vibration isolators are directly attached to the engine block heat sink it was necessary to measure the maximum temperature of that region prior to mounting any accelerometers.

The temperature was measured with two different methods, using a FLIR thermal camera (FLIR, 2014) and using a thermocouple (Omega, 2014), simultaneously to be able to guarantee reliable results. The FLIR camera is an infrared imaging system that can provide quick and easy, yet accurate, temperature measurement over an area. The image created includes a temperature colour scale as well as specific information about focus point and local maximum temperature. A problem with the FLIR system is the cameras size which makes it difficult to get measurements in crowded spaces where no direct line of sight can be achieved.

The thermocouple on the other hand can only measure the temperature at a single spot. They work by measuring the resistance where two wires of different material meet. The resistance in this intersection increases proportional to the temperature. For the sensor to work properly it is important that the intersection is in contact with the spot where the measurement should be taken. It is also important that there is a sturdy connection between the two wires. The advantage of this sensor type is its small size, making it possible to reach virtually anywhere.

When measuring the temperature the engine was first run at maximum rpm for about five minutes, at which point no further increase in temperature could be perceived, and then turned off. The temperature was continuously measured first during the engine run and then after it was turned off until the temperature started to drop again. The reason for continuing to measure after the engine is turned off is that the engine tends to increase slightly in temperature immediately after due to the lack of cooling air flow. In fact it was at this point that the maximum temperature was found.

In order to find a safe placement for the accelerometer two different spots of interest, both an ideal and a backup, were measured (see figure C1). Unfortunately the thermocouple placed at the backup spot was impossible to get into solid contact with the engine heat sink so this spot had to be measured only using the FLIR. The opposite case applies to the ideal spot, where the FLIR cannot see so only the thermocouple measured there. The results from the measurements are presented in table C1. The images from the FLIR can be seen in figure C2 and C3.

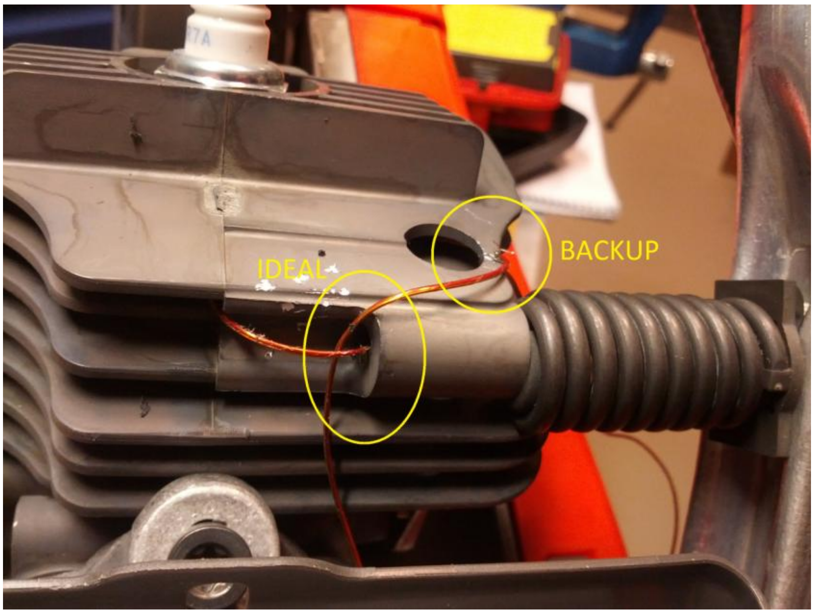

Figure C1. The two considered spots for mounting the accelerometer.

Table C1

Spot	FLIR (°C)	Thermocouple (°C)
Ideal	-	112
Backup	110	-

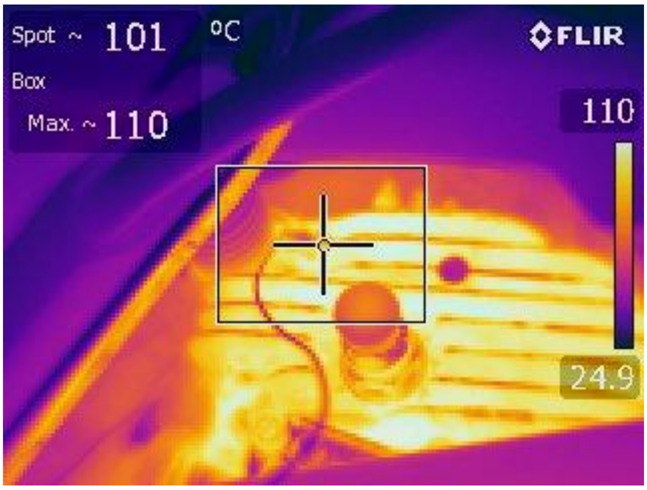

Figure C2. FLIR ir-image angled to see only the heat sink, focused on the backup spot.

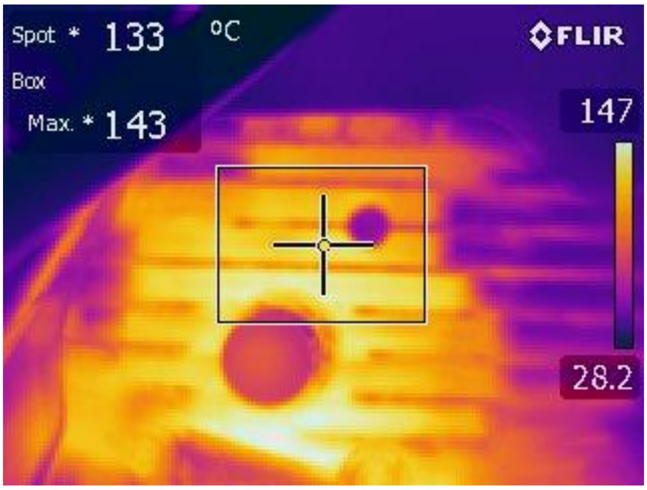

Figure C3. FLIR ir-image of the cylinder and heat sink focused on the cylinder wall itself.

Because the maximum temperature at the ideal spot is still below the maximum rated temperature of the accelerometer with some margin, it was decided that this placement would be safe enough. Although the specific spot could not be verified also by the FLIR, the fact that the highest temperature measured anywhere on the actual heat sink was only 110°C (see figure C2) supports the validity of thermocouple result. The maximum temperature measured on the external of the cylinder was 147°C which is still not that far above the maximum rated for the accelerometer and measured quite far from the intended placement. Hence it seems unlikely that the accelerometer will ever be subjected to unsafe temperature levels.

References:

FLIR (2014), FLIR E-Series, http://www.flir.com/cs/emea/en/view/?id=41372 (2014-02-10)

Omega (2014), Introduction to thermocouples, http://www.omega.com/prodinfo/thermocouples.html (2014-02-10)

D

Initial Series of Tests

The waterfall graphs from the initial tests are presented in this appendix. The graphs show short spans of rpm sweeps. The sweeps were performed close to idle and work rpm. The sweep close to the idle rpm goes approximately from 2300 to 3300 rpm and the sweep close to the work rpm goes approximately from 8100 to 9100 rpm. In the graphs the amplitude of the vibrations is on the z-axis, the rpm of the machine when performing the measurement is on the y-axis and the order of the operating speed that the vibrations are vibrating in is on the x-axis. The graphs show the measured vibrations on the vibrating as well as the vibration isolated side in x-, y-, and z-direction at all three spring positions. The graphs show the results from measurements with weak, standard and stiff spring setups.

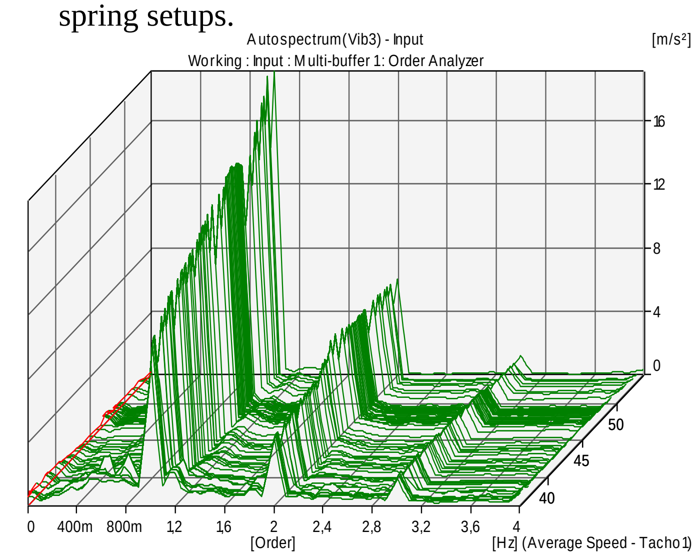

Figure D1a. Input vibrations, cutting arm – feet, idle rpm, x-direction, standard springs.

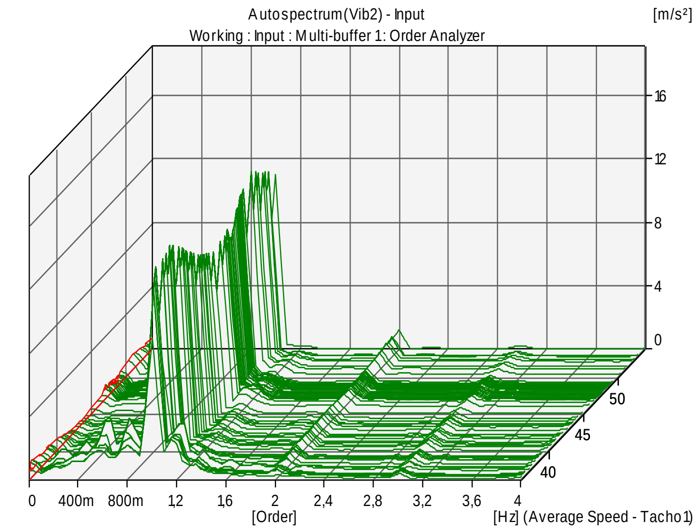

Figure D1b. Output vibrations, cutting arm – feet, idle rpm, x-direction, standard springs.

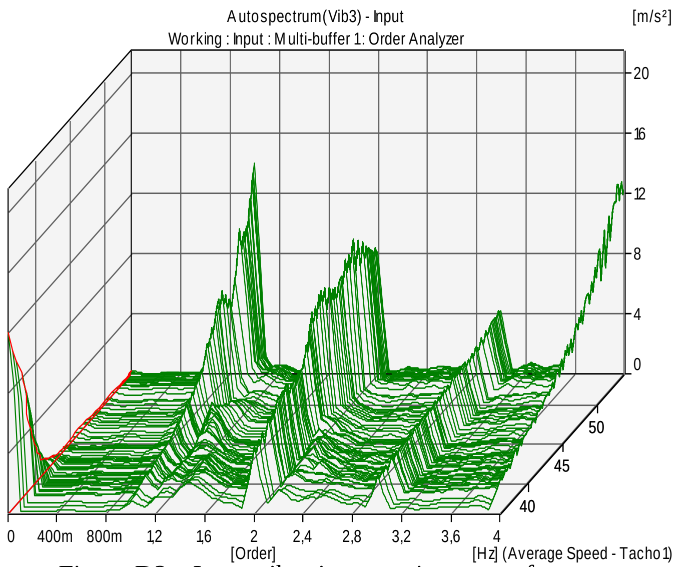

Figure D2a. Input vibrations, cutting arm – feet, idle rpm, y-direction, standard springs.

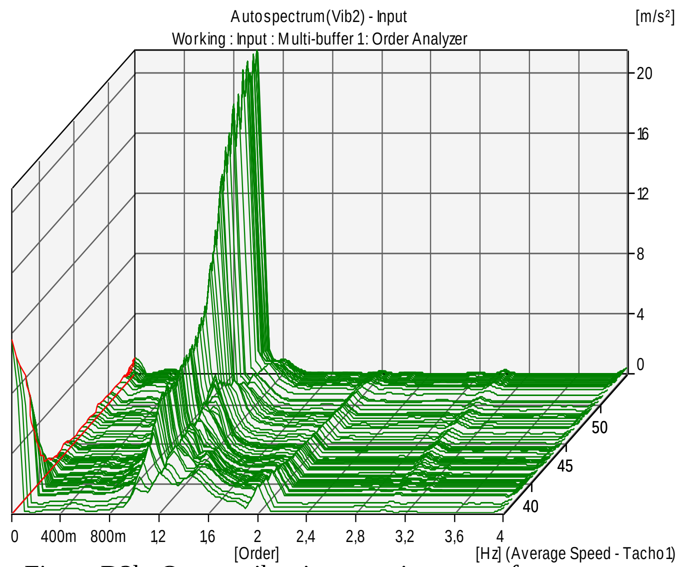

Figure D2b. Output vibrations, cutting arm – feet, idle rpm, y-direction, standard springs.

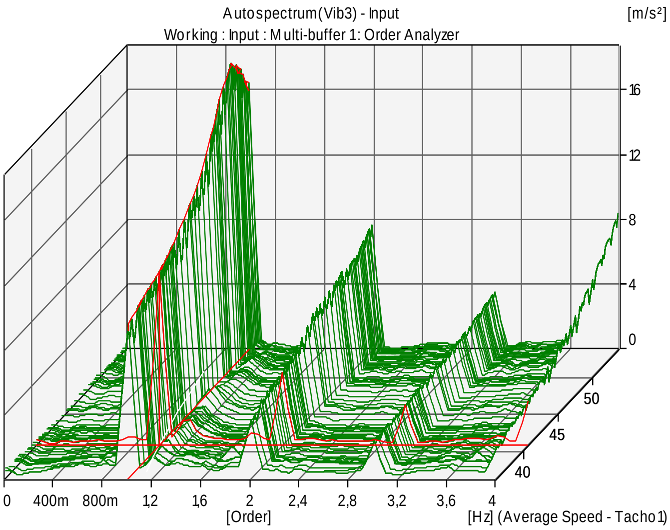

Figure D3a. Input vibrations, cutting arm – feet, idle rpm, z-direction, standard springs.

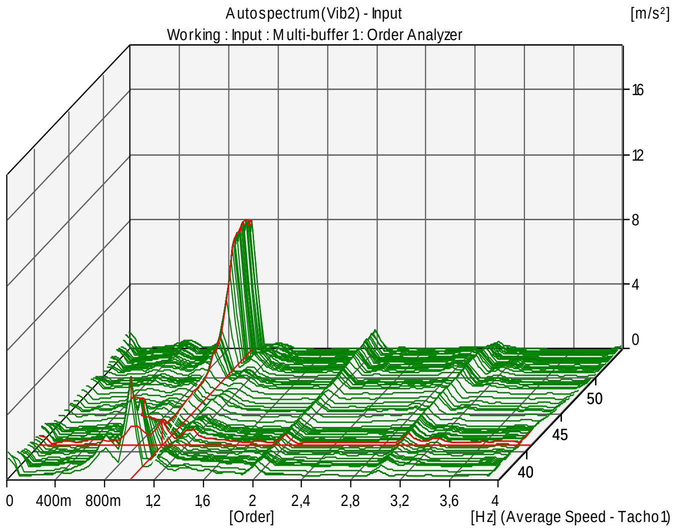

Figure D3b. Output vibrations, cutting arm – feet, idle rpm, z-direction, standard springs.

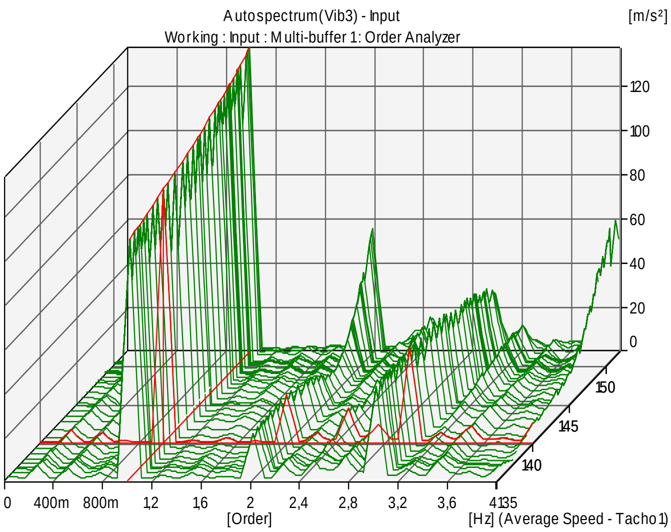

Figure D4a. Input vibrations, cutting arm – feet, work rpm, x-direction, standard springs.

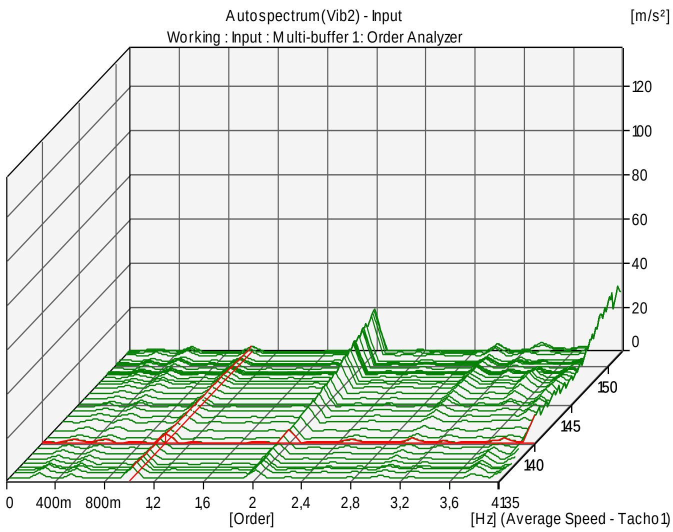

Figure D4b. Output vibrations, cutting arm – feet, work rpm, x-direction, standard springs.

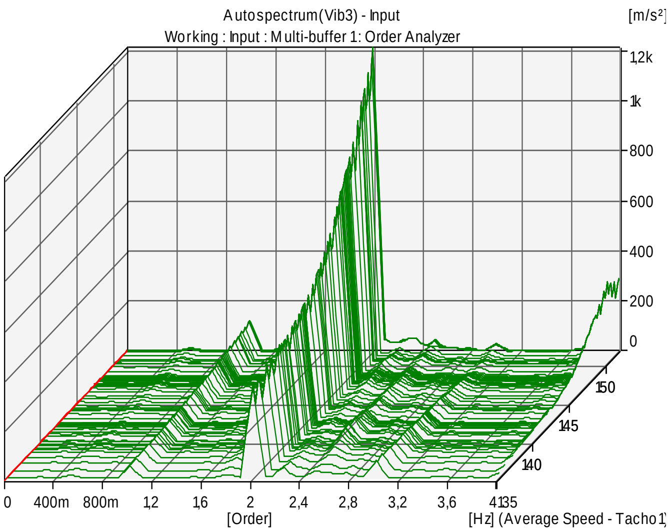

Figure D5a. Input vibrations, cutting arm – feet, work rpm, y-direction, standard springs.

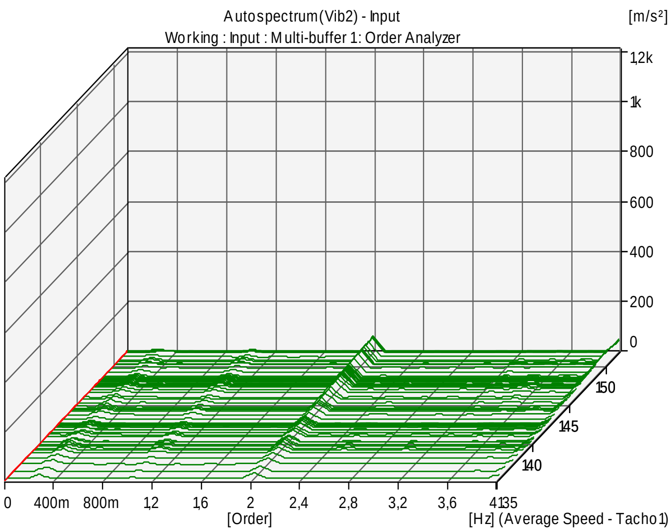

Figure D5b. Output vibrations, cutting arm – feet, work rpm, y-direction, standard springs.

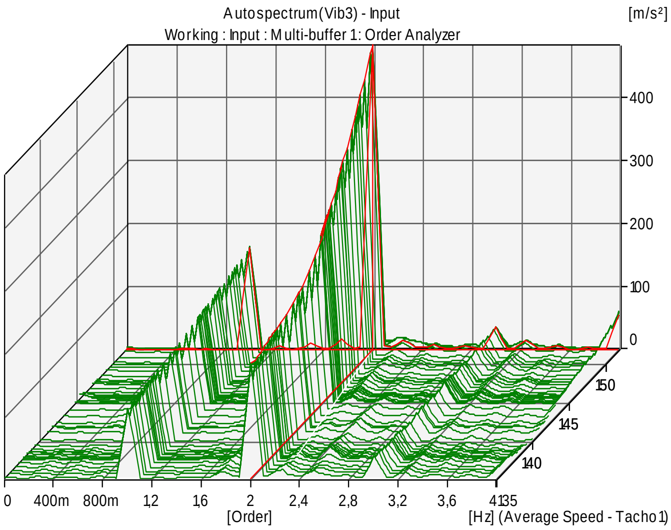

Figure D6a. Input vibrations, cutting arm – feet, work rpm, z-direction, standard springs.

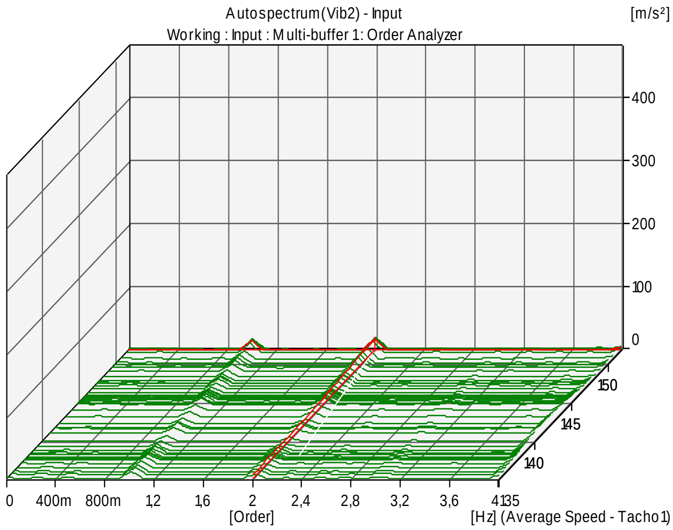

Figure D6b. Output vibrations, cutting arm – feet, work rpm, z-direction, standard springs.

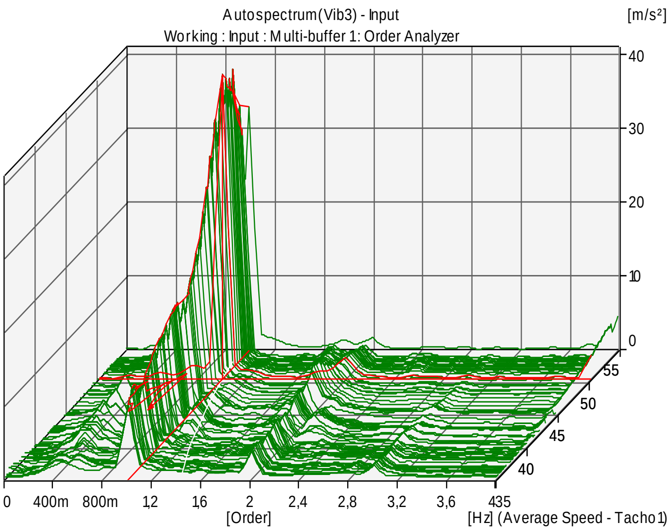

Figure D7a. Input vibrations, air intake - tank, idle rpm, x-direction, standard springs.

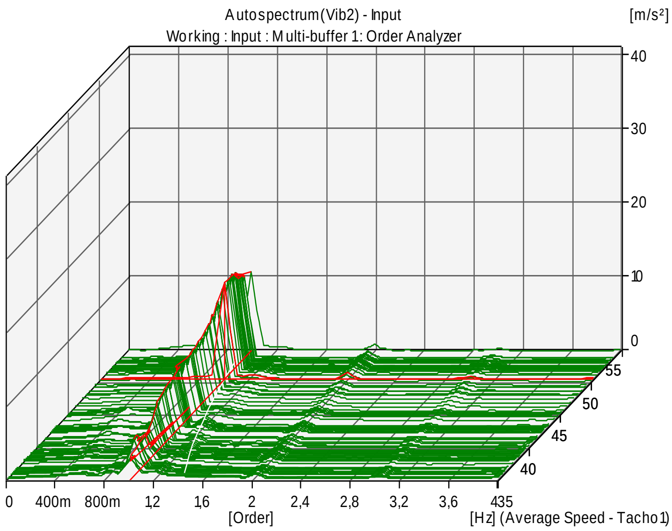

Figure D7b. Output vibrations air intake - tank, idle rpm, x-direction, standard springs.

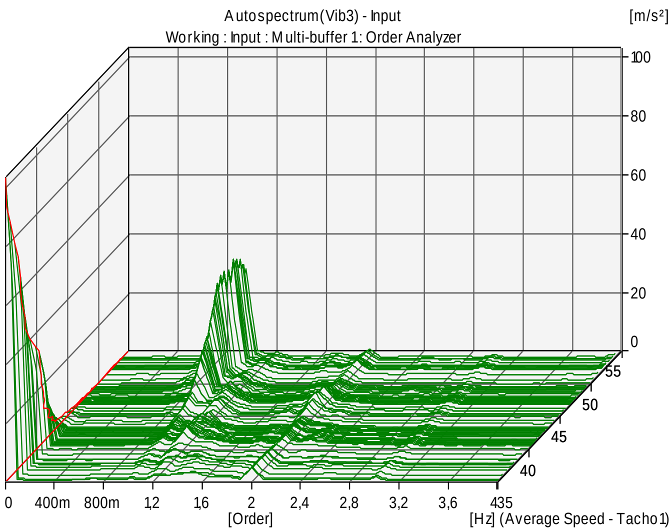

Figure D8a. Input vibrations, air intake - tank, idle rpm, y-direction, standard springs.

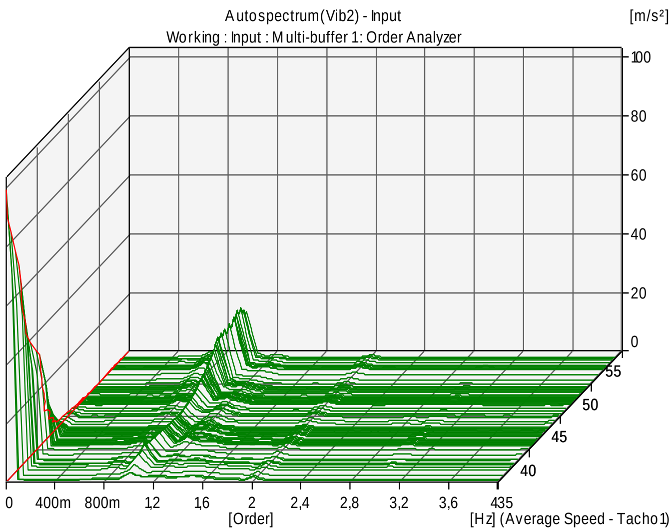

Figure D8b. Output vibrations air intake - tank, idle rpm, y-direction, standard springs.

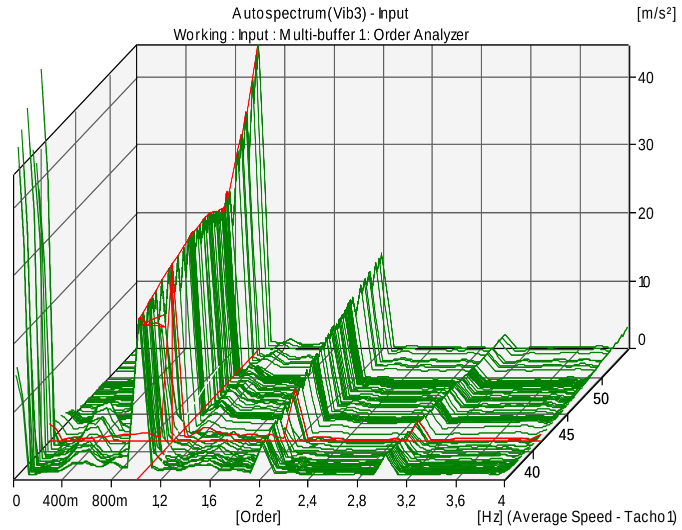

Figure D9a. Input vibrations, air intake - tank, idle rpm, z-direction, standard springs.

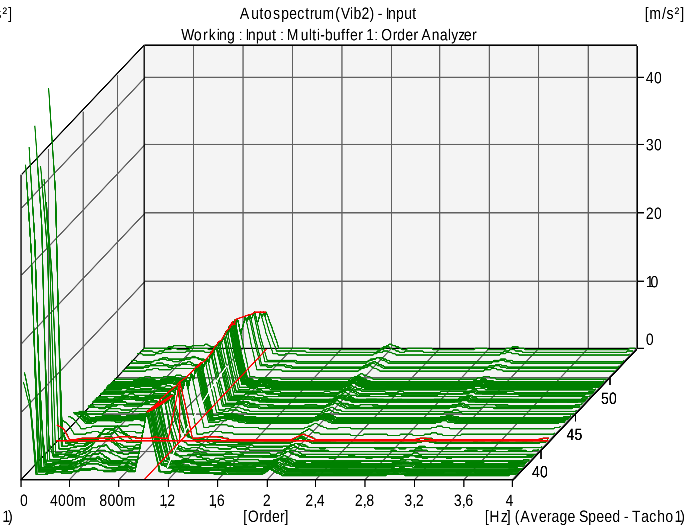

Figure D9b. Output vibrations air intake - tank, idle rpm, z-direction, standard springs.

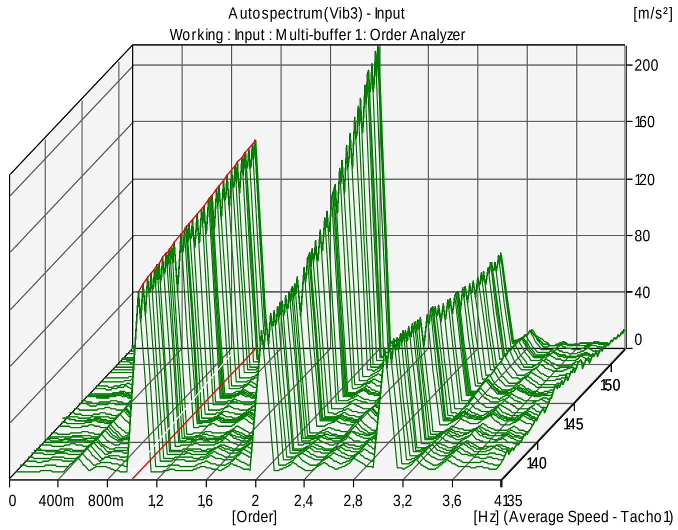

Figure D10a. Input vibrations, air intake - tank, work rpm, x-direction, standard springs.

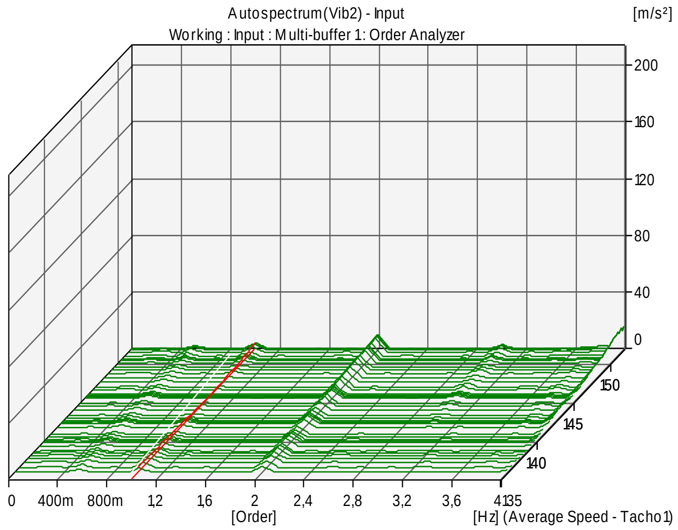

Figure D10b. Output vibrations air intake - tank, work rpm, x-direction, standard springs.

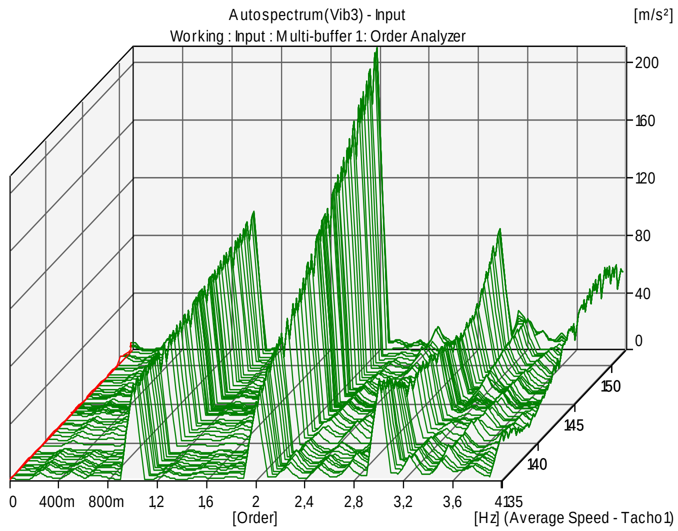

Figure D11a. Input vibrations, air intake - tank, work rpm, y-direction, standard springs.

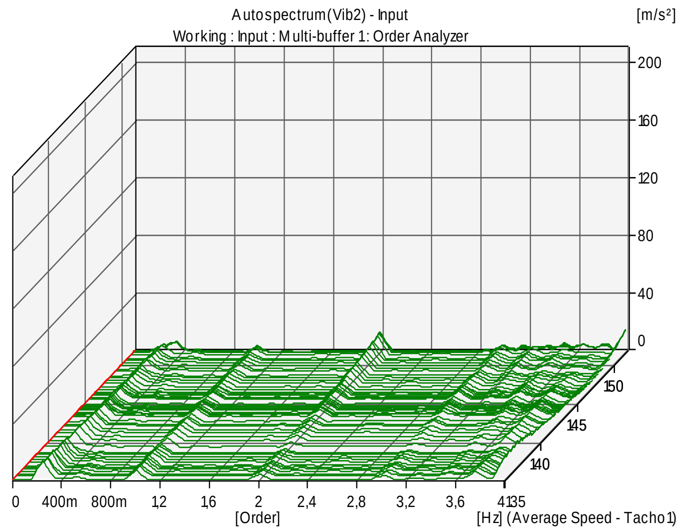

Figure D11b. Output vibrations air intake - tank, work rpm, y-direction, standard springs.

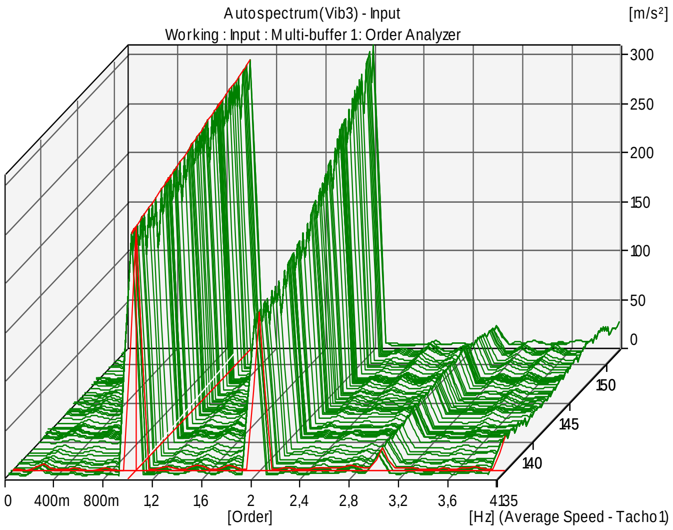

Figure D12a. Input vibrations, air intake - tank, work rpm, z-direction, standard springs.

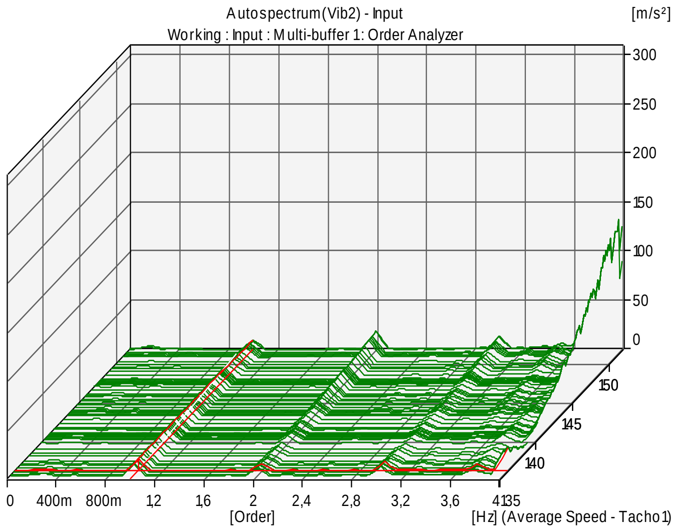

Figure D12b. Output vibrations air intake - tank, work rpm, z-direction, standard springs.

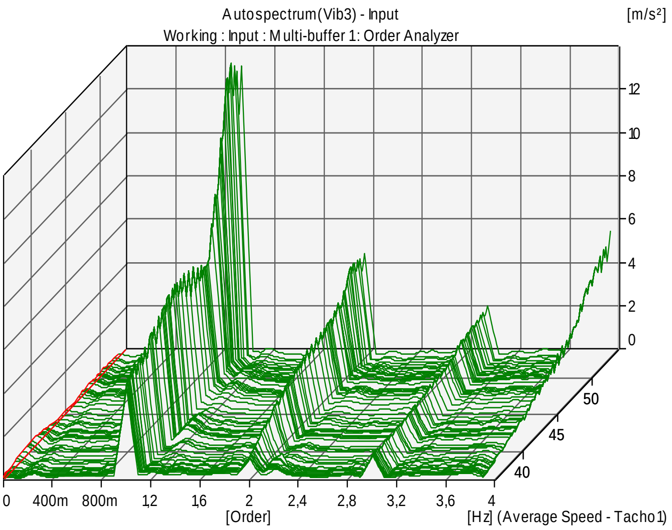

Figure D13a. Input vibrations, engine - handle, idle rpm, x-direction, standard springs.

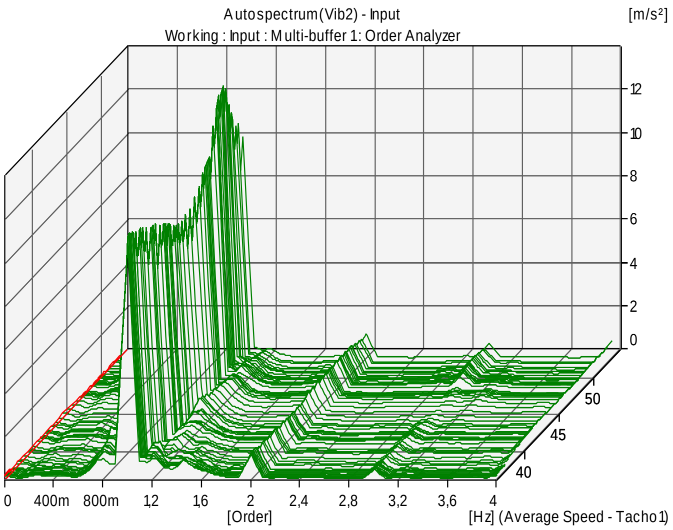

Figure D13b. Output vibrations, engine - handle, idle rpm, x-direction, standard springs.

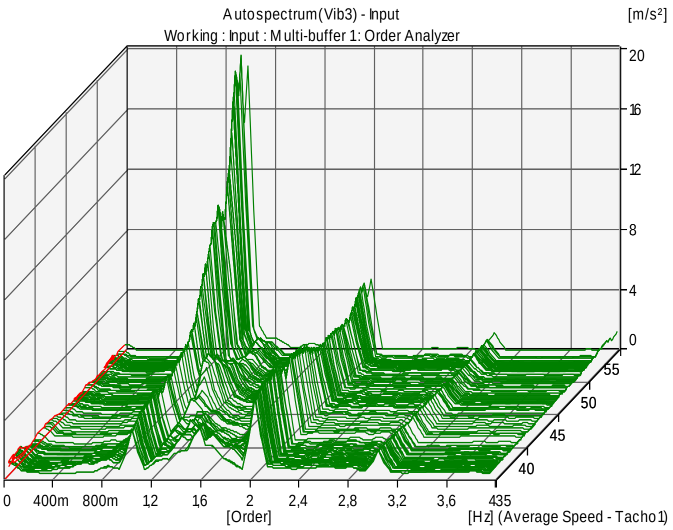

Figure D14a. Input vibrations, engine - handle, idle rpm, y-direction, standard springs.

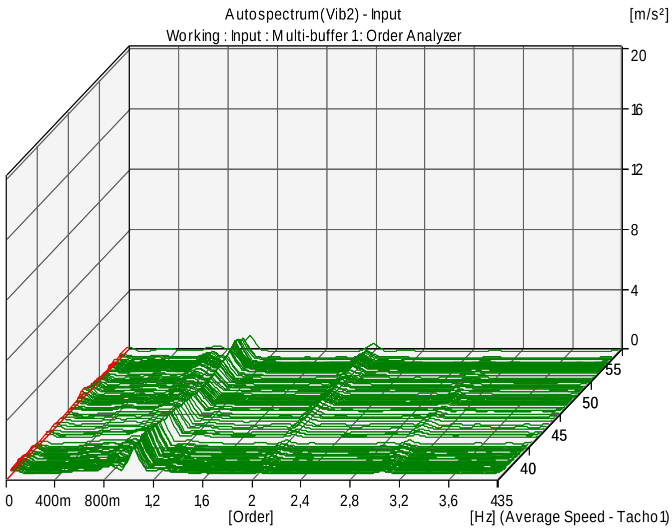

Figure D14b. Output vibrations, engine - handle, idle rpm, y-direction, standard springs.

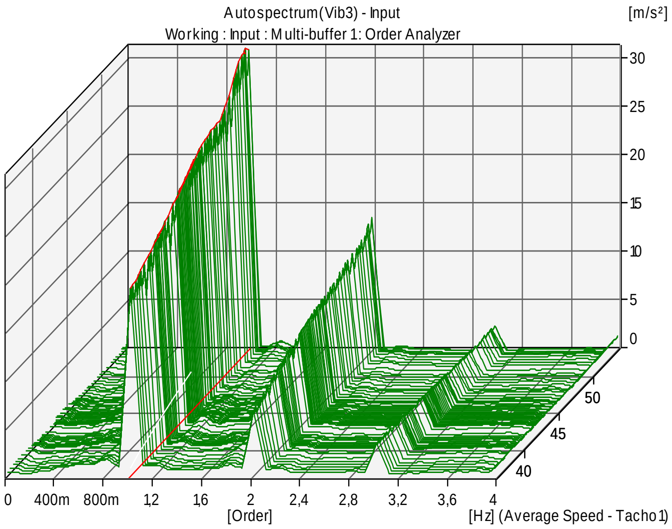

Figure D15a. Input vibrations, engine - handle, idle rpm, z-direction, standard springs.

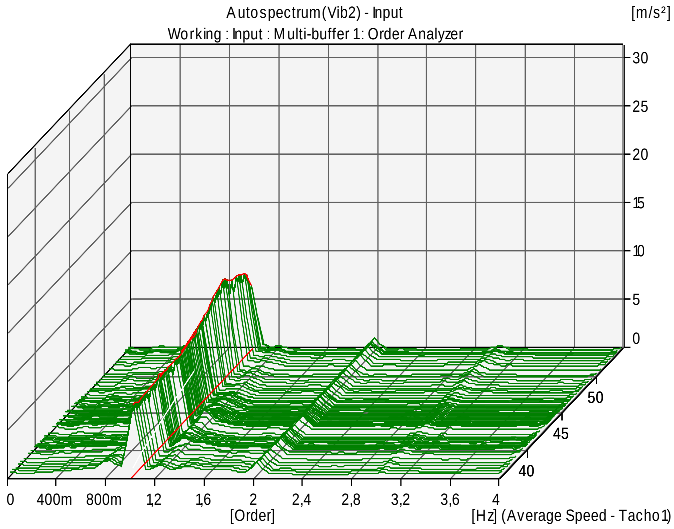

Figure D15b. Output vibrations, engine - handle, idle rpm, z-direction, standard springs.

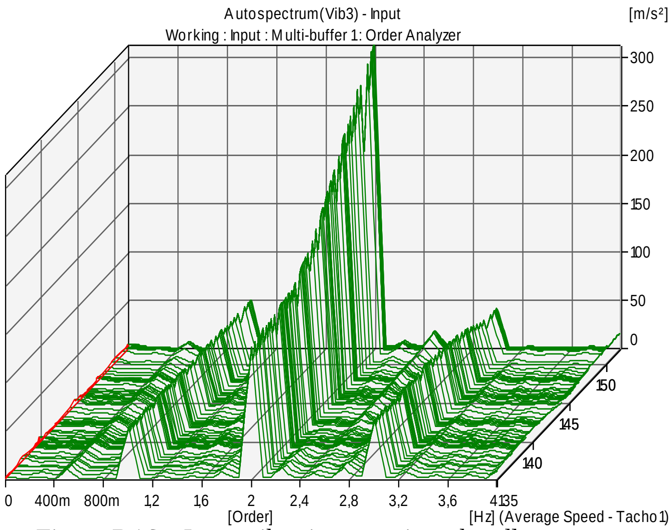

Figure D16a. Input vibrations, engine - handle, work rpm, x-direction, standard springs.

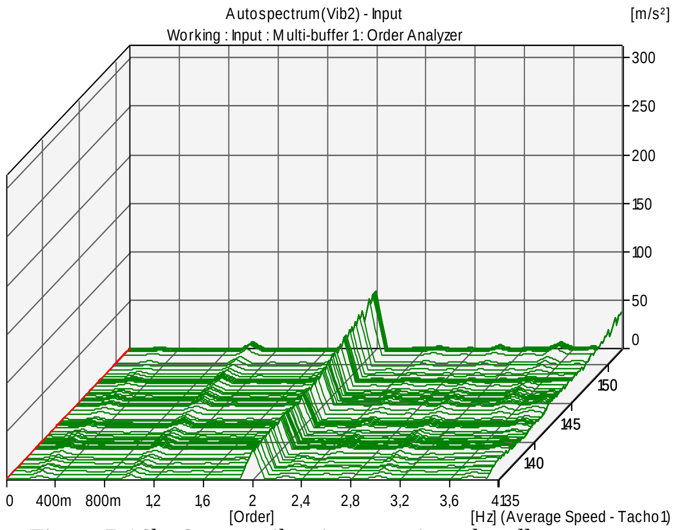

Figure D16b. Output vibrations, engine - handle, work rpm, x-direction, standard springs.

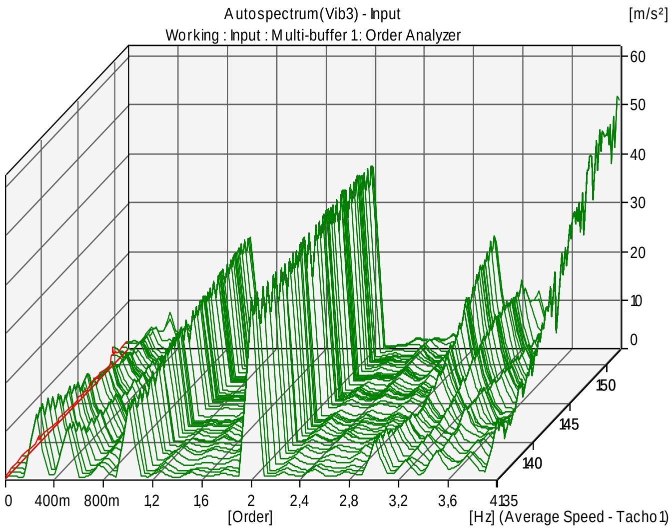

Figure D17a. Input vibrations, engine - handle, work rpm, y-direction, standard springs.

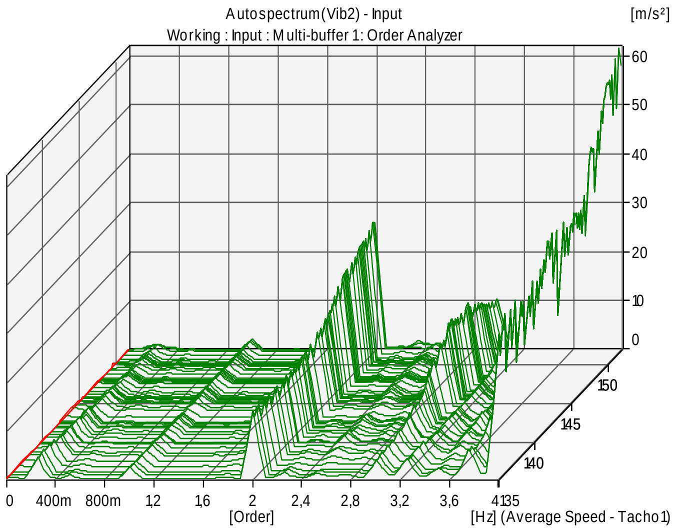

Figure D17b. Output vibrations, engine - handle, work rpm, y-direction, standard springs.

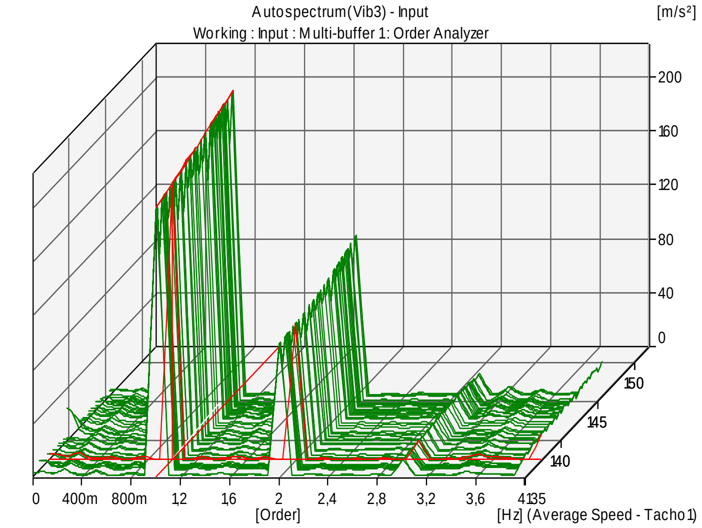

Figure D18a. Input vibrations, engine - handle, work rpm, z-direction, standard springs.

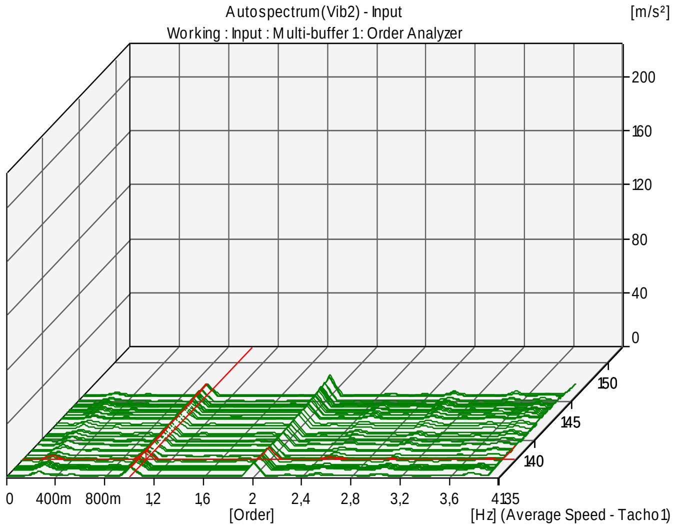

Figure D18b. Output vibrations, engine - handle, work rpm, z-direction, standard springs.

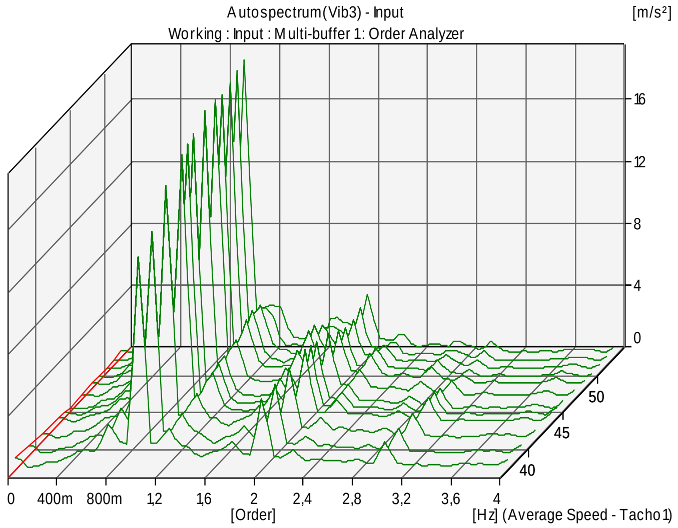

Figure D19a. Input vibrations, cutting arm - feet, idle rpm, x-direction, stiff springs.

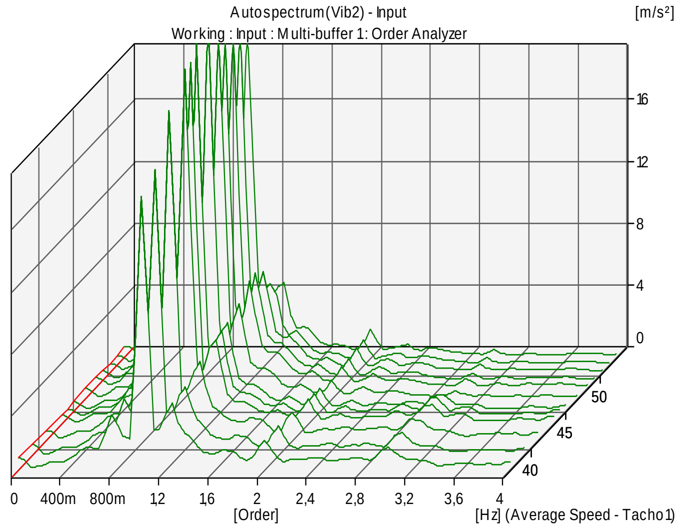

Figure D19b. Output vibrations, cutting arm - feet, idle rpm, x-direction, stiff springs.

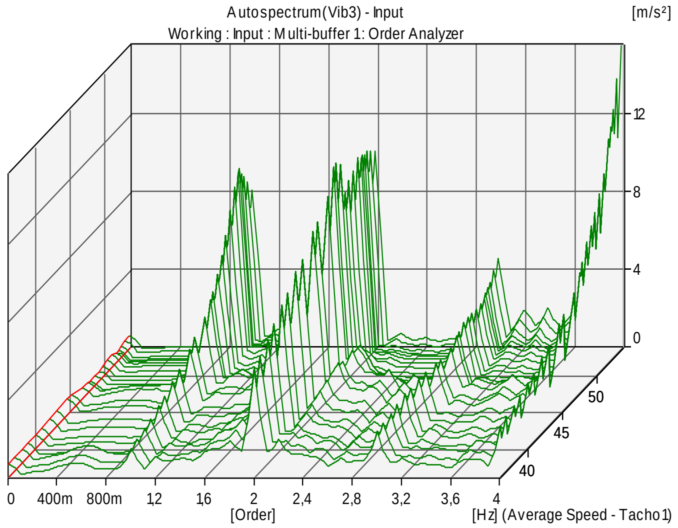

Figure D20a. Input vibrations, cutting arm - feet, idle rpm, y-direction, stiff springs.

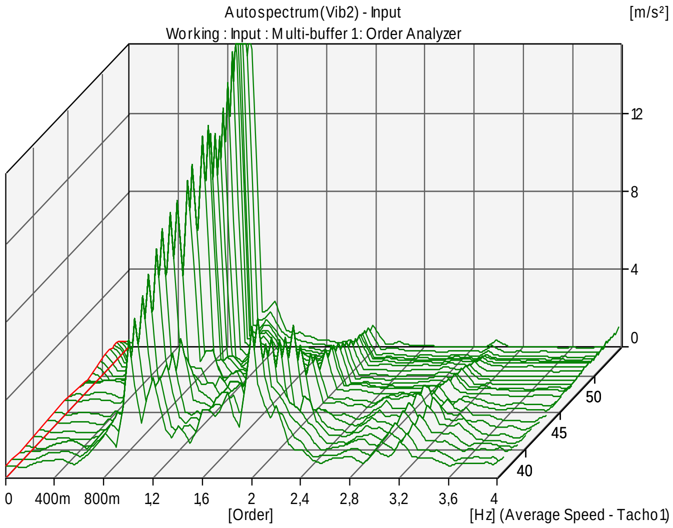

Figure D20b. Output vibrations, cutting arm - feet, idle rpm, y-direction, stiff springs.

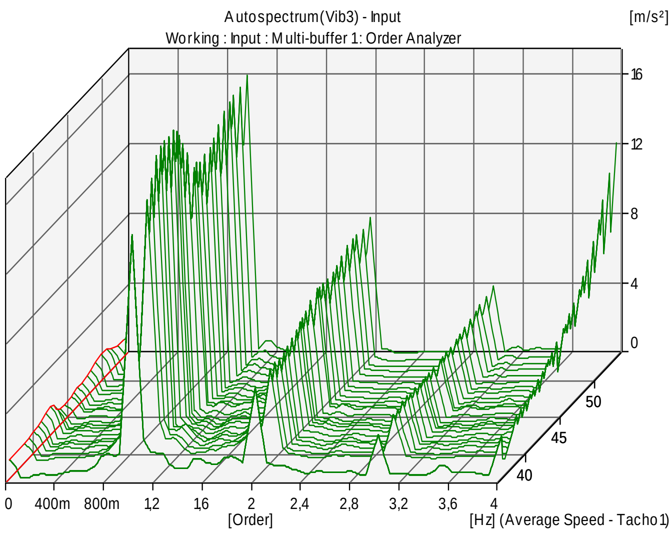

Figure D21a. Input vibrations, cutting arm - feet, idle rpm, z-direction, stiff springs.

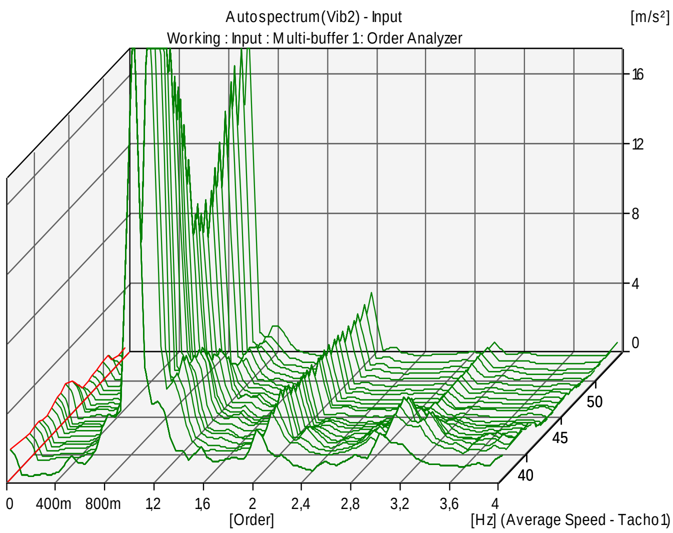

Figure D21b. Output vibrations, cutting arm - feet, idle rpm, z-direction, stiff springs.

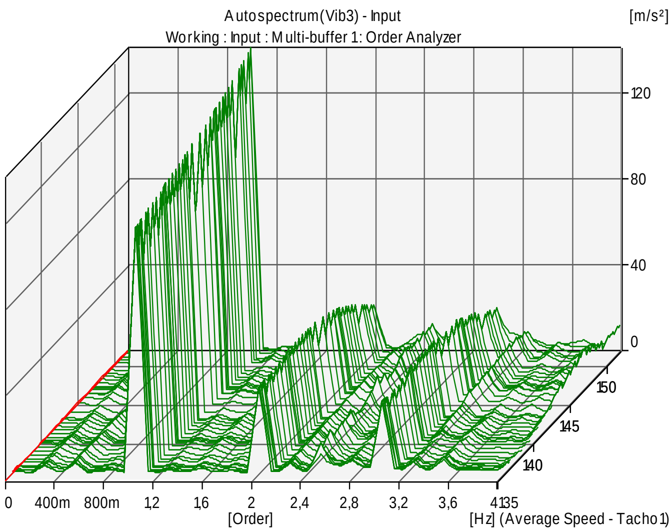

Figure D22a. Input vibrations, cutting arm - feet, work rpm, x-direction, stiff springs.

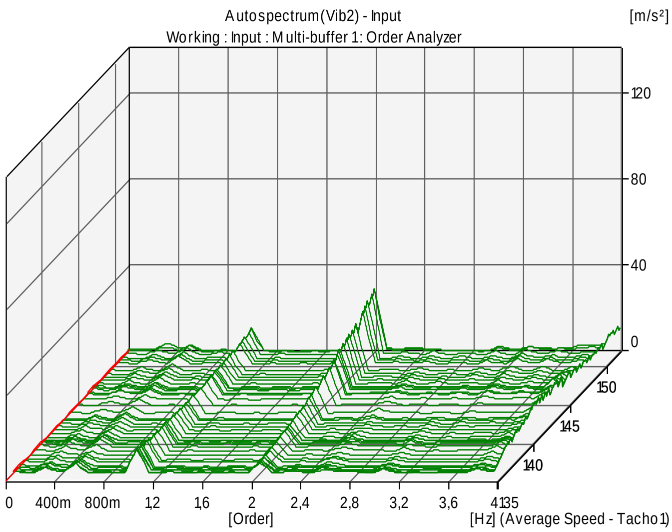

Figure D22b. Output vibrations, cutting arm - feet, work rpm, x-direction, stiff springs.

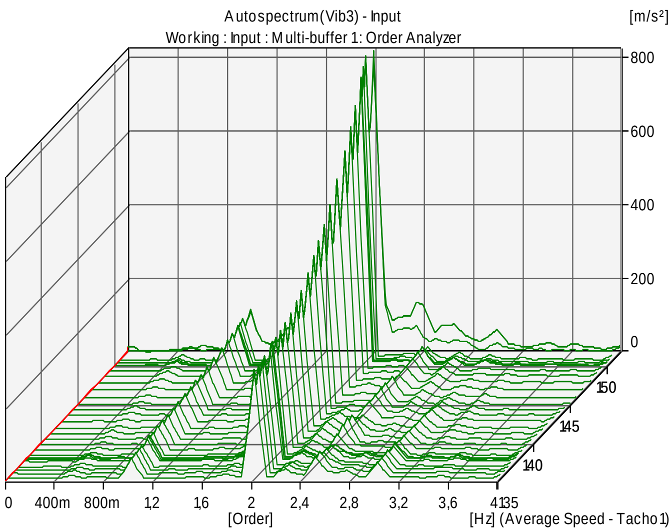

Figure D23a. Input vibrations, cutting arm - feet, work rpm, y-direction, stiff springs.

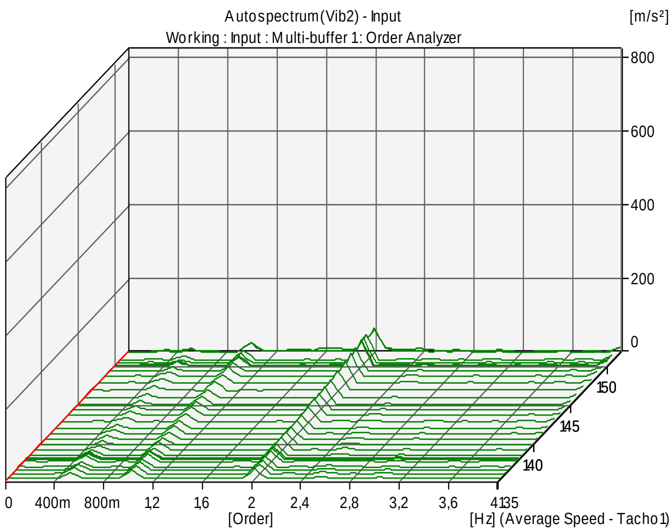

Figure D23b. Output vibrations, cutting arm - feet, work rpm, y-direction, stiff springs.

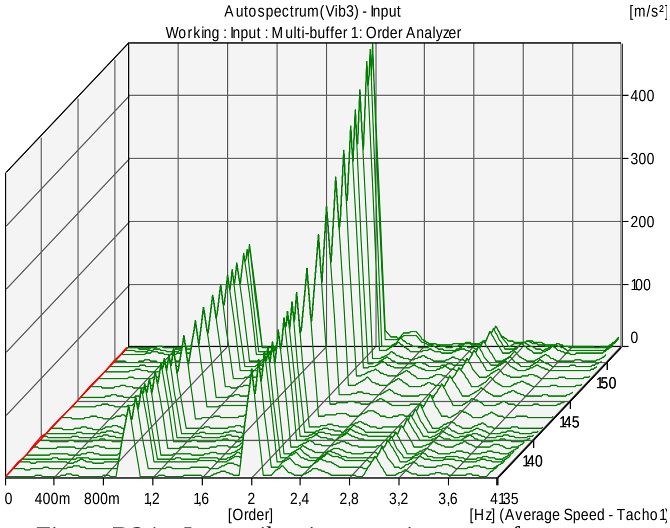

Figure D24a. Input vibrations, cutting arm - feet, work rpm, z-direction, stiff springs.

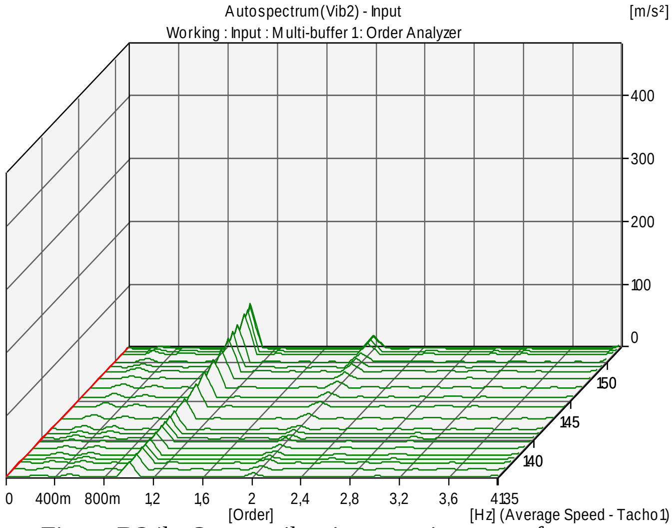

Figure D24b. Output vibrations, cutting arm - feet, work rpm, z-direction, stiff springs.

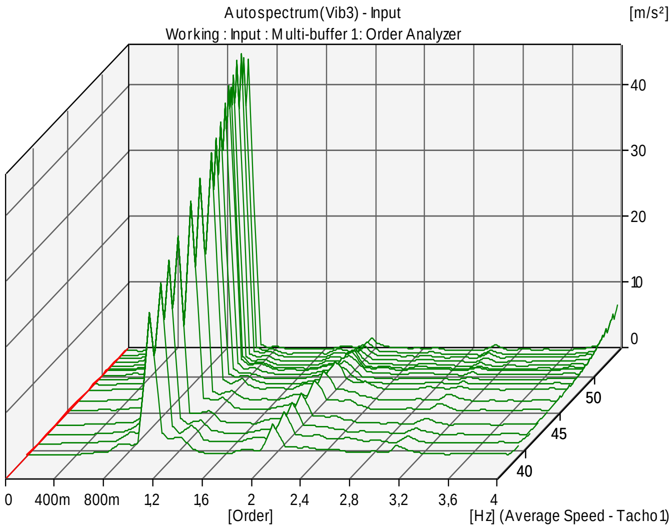

Figure D25a. Input vibrations, air intake - tank, idle rpm, x-direction, stiff springs.

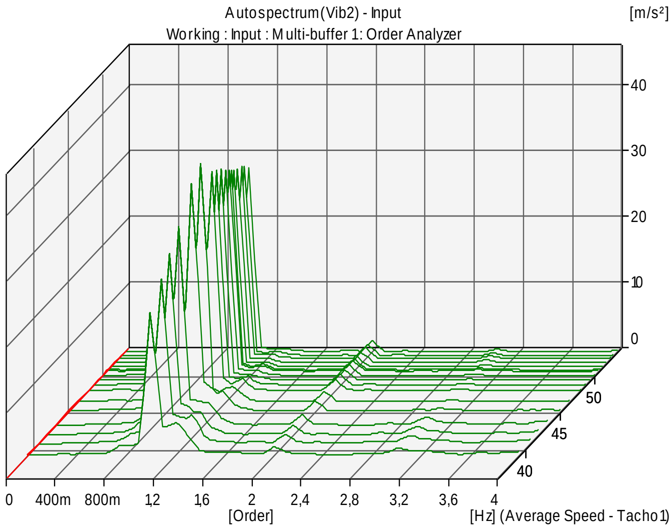

Figure D25b. Output vibrations, air intake - tank, idle rpm, x-direction, stiff springs.

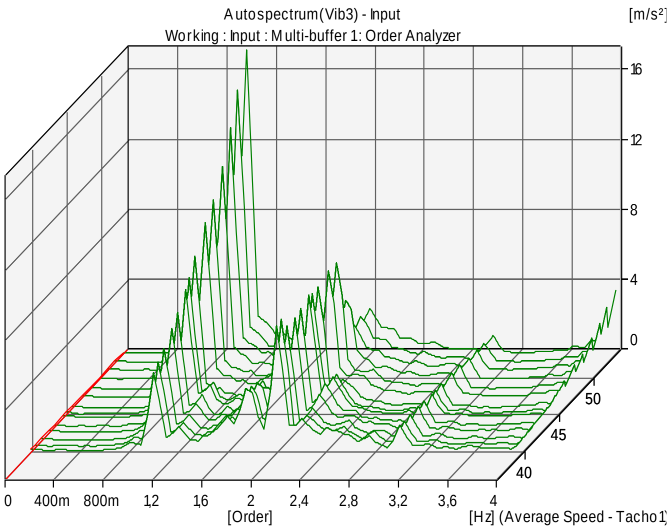

Figure D26a. Input vibrations, air intake - tank, idle rpm, y-direction, stiff springs.

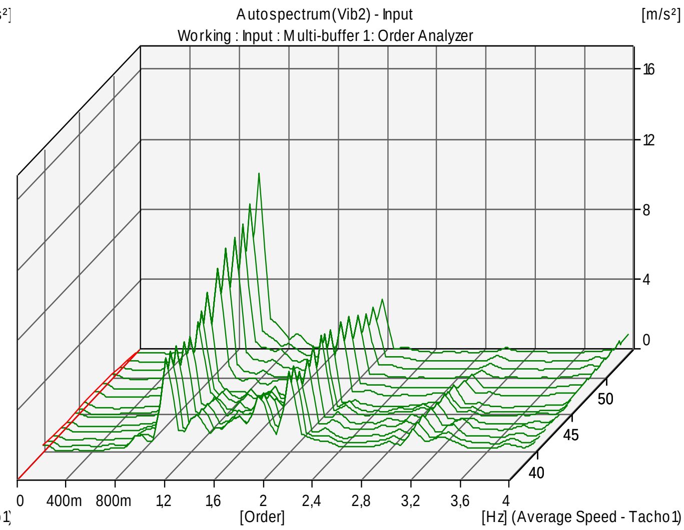

Figure D26b. Output vibrations, air intake - tank, idle rpm, y-direction, stiff springs.

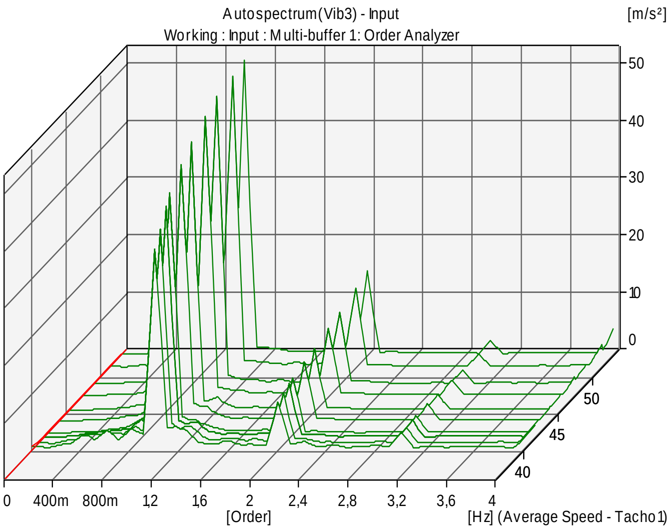

Figure D27a. Input vibrations, air intake - tank, idle rpm, z-direction, stiff springs.

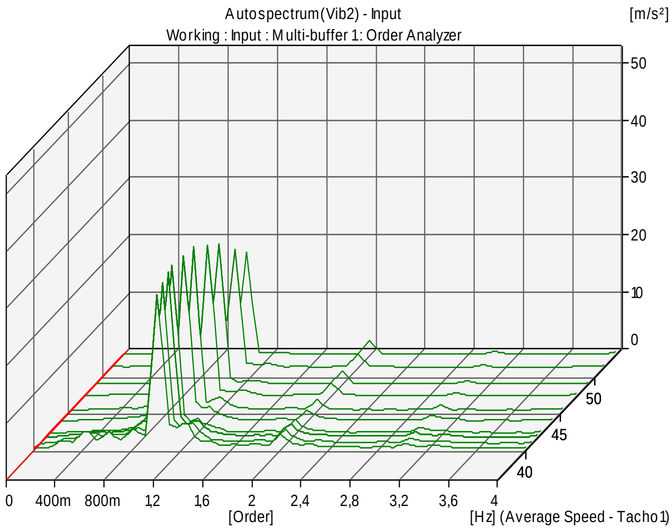

Figure D27b. Output vibrations, air intake - tank, idle rpm, z-direction, stiff springs.

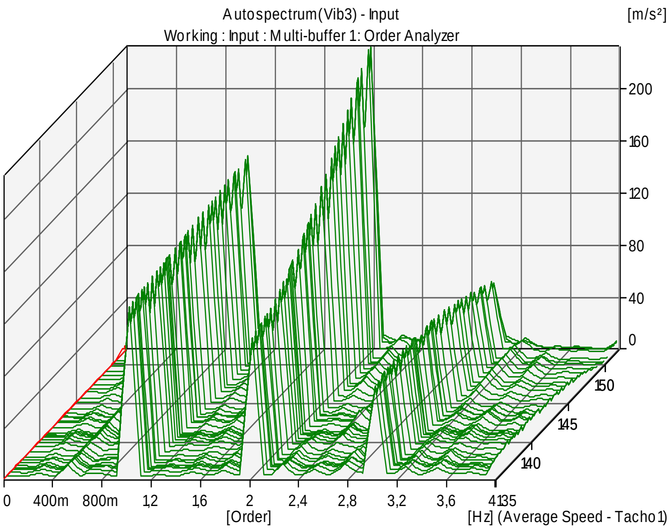

Figure D28a. Input vibrations, air intake - tank, work rpm, x-direction, stiff springs.

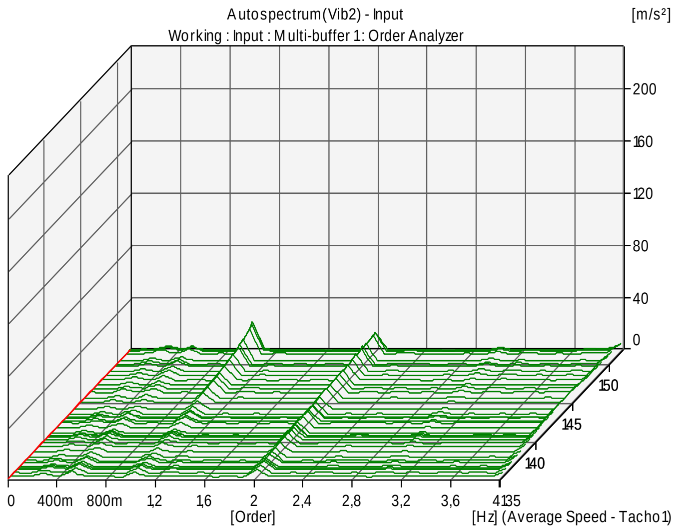

Figure D28b. Output vibrations, air intake - tank, work rpm, x-direction, stiff springs.

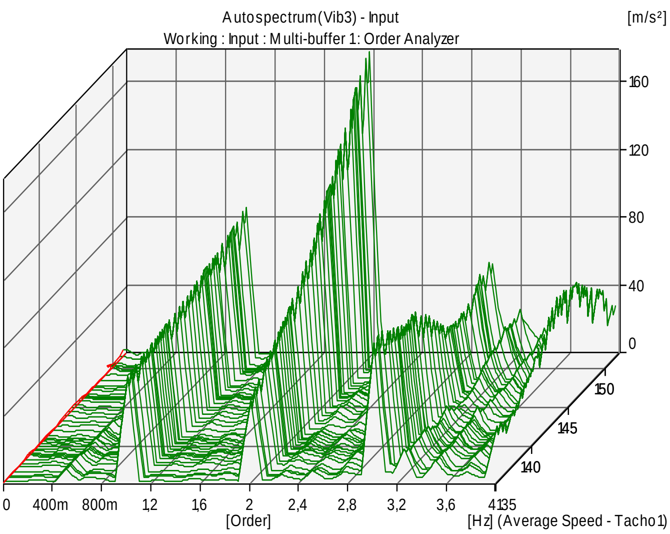

Figure D29a. Input vibrations, air intake - tank, work rpm, y-direction, stiff springs.

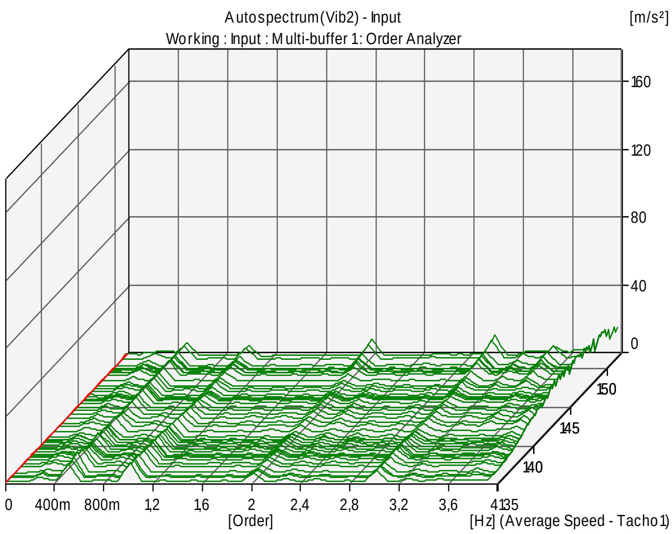

Figure D29b. Output vibrations, air intake - tank, work rpm, y-direction, stiff springs.

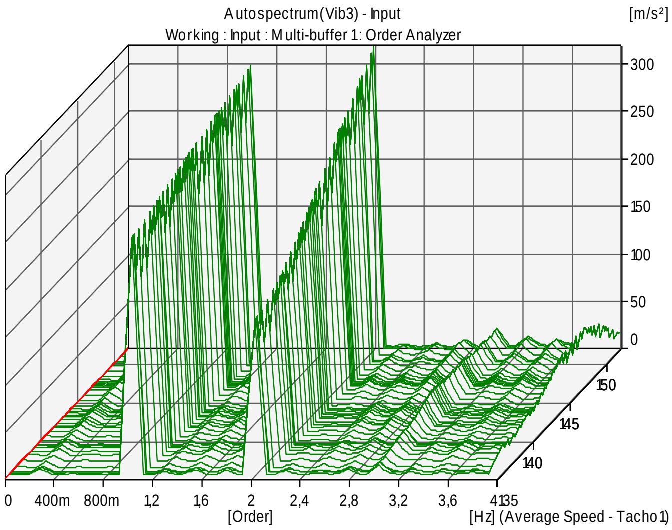

Figure D30a. Input vibrations, air intake - tank, work rpm, z-direction, stiff springs.

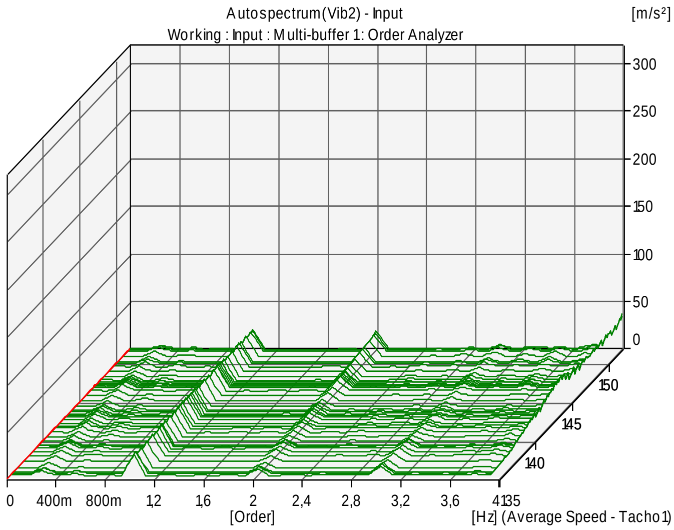

Figure D30b. Output vibrations, air intake - tank, work rpm, z-direction, stiff springs.

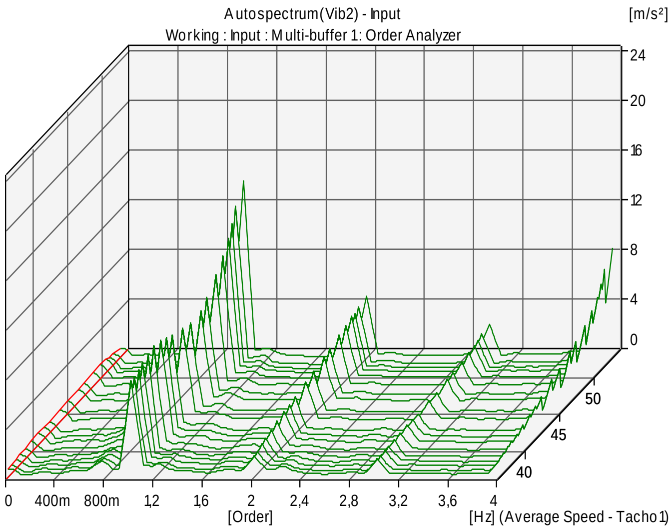

Figure D31a. Input vibrations, engine - handle, idle rpm, x-direction, stiff springs.

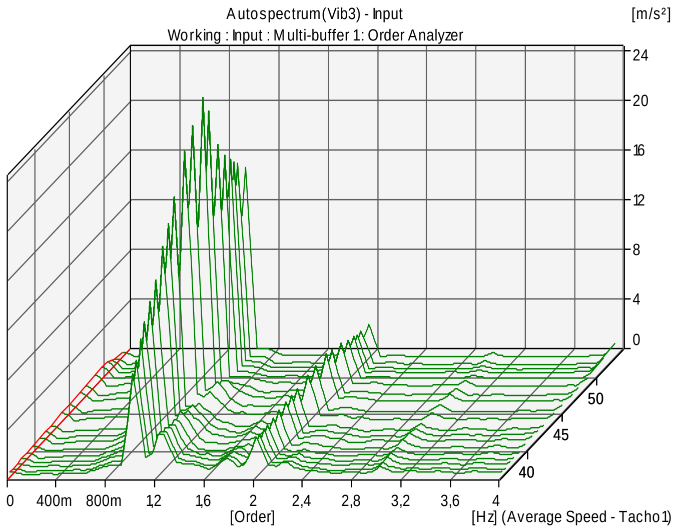

Figure D31b. Output vibrations, engine - handle, idle rpm, x-direction, stiff springs.

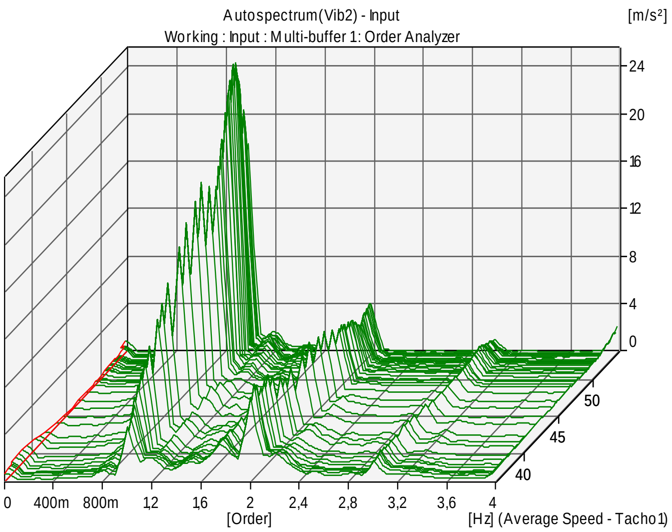

Figure D32a. Input vibrations, engine - handle, idle rpm, y-direction, stiff springs.

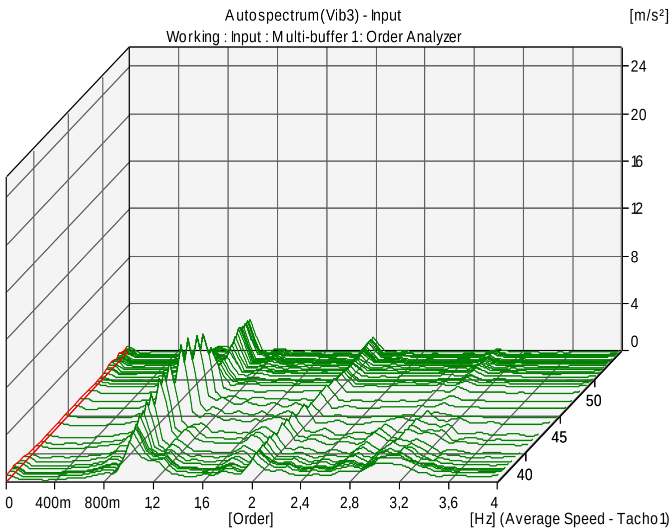

Figure D32b. Output vibrations, engine - handle, idle rpm, y-direction, stiff springs.

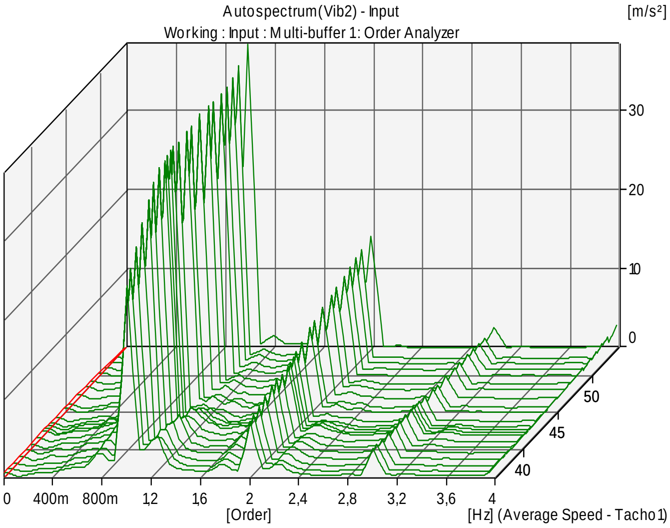

Figure D33a. Input vibrations, engine - handle, idle rpm, z-direction, stiff springs.

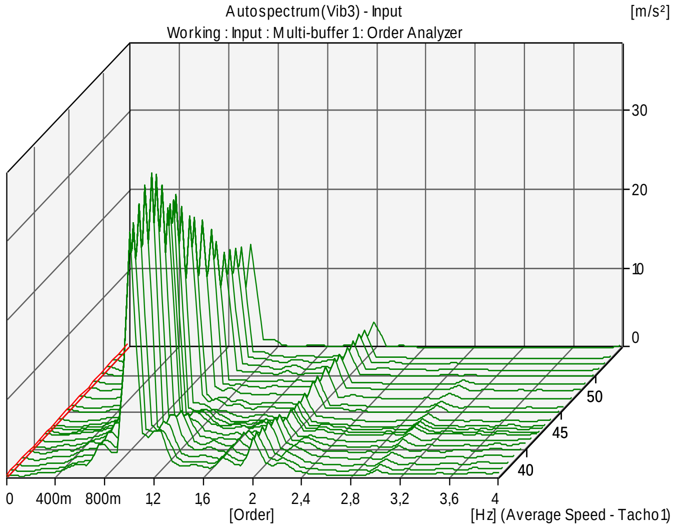

Figure D33b. Output vibrations, engine - handle, idle rpm, z-direction, stiff springs.

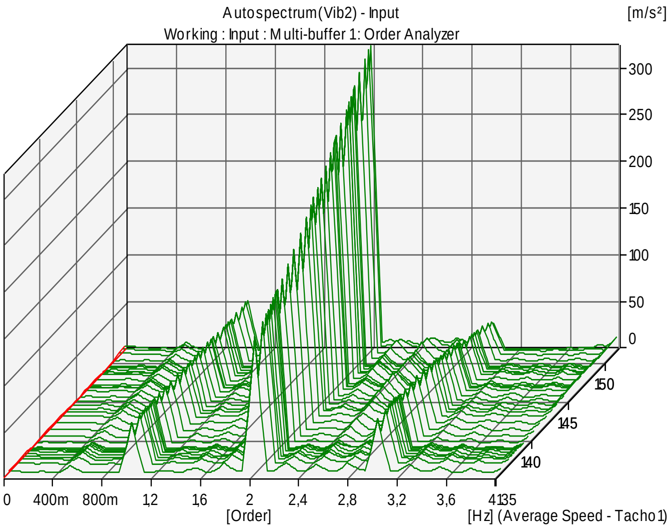

Figure D34a. Input vibrations, engine - handle, work rpm, x-direction, stiff springs.

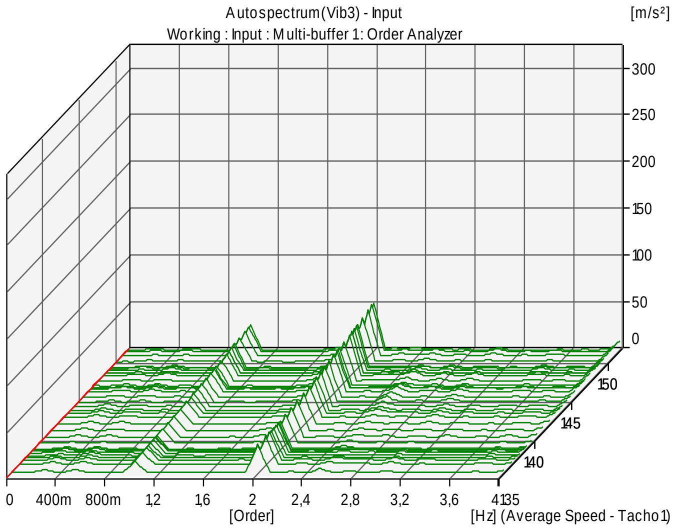

Figure D34b. Output vibrations, engine - handle, work rpm, x-direction, stiff springs.

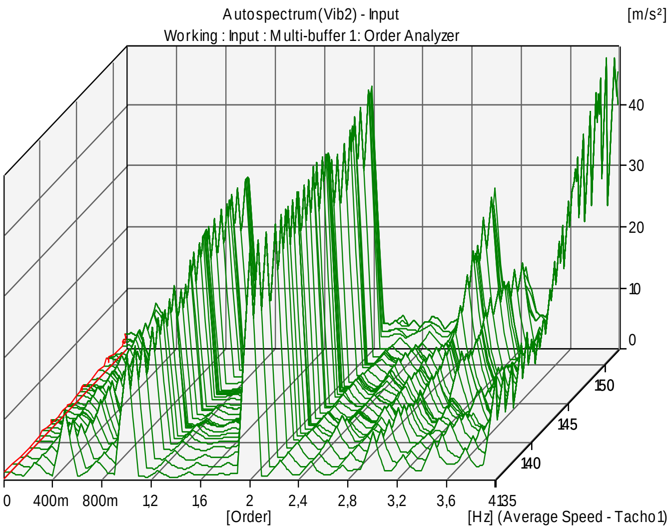

Figure D35a. Input vibrations, engine - handle, work rpm, y-direction, stiff springs.

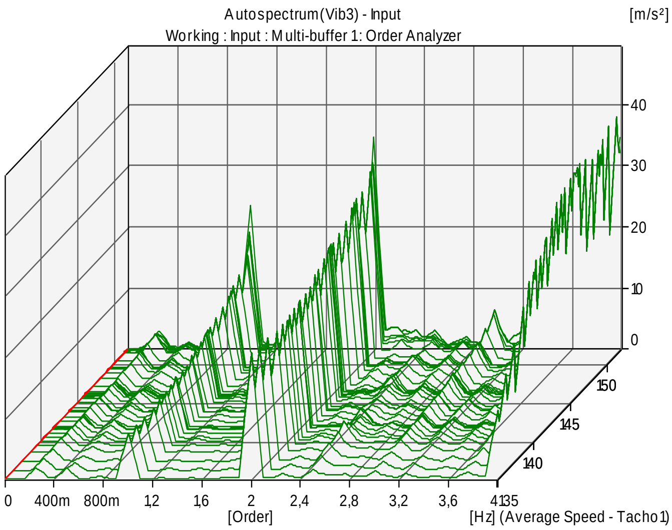

Figure D35b. Output vibrations, engine - handle, work rpm, y-direction, stiff springs.

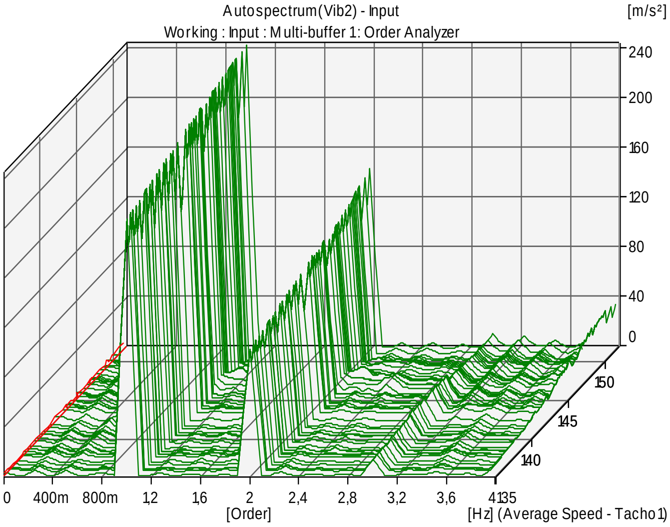

Figure D36a. Input vibrations, engine - handle, work rpm, z-direction, stiff springs.

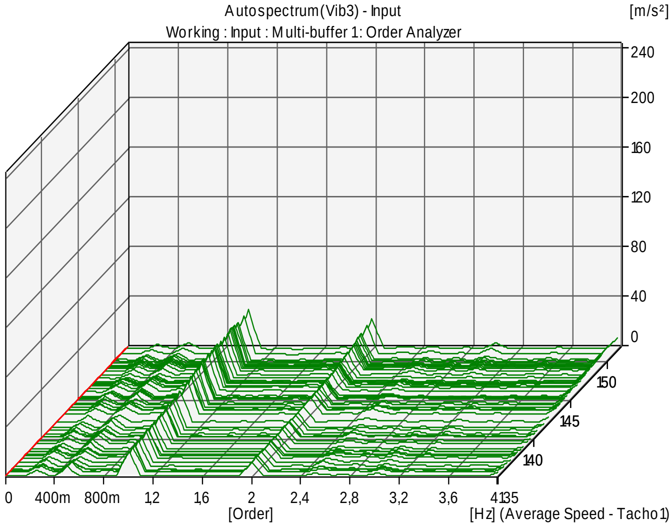

Figure D36b. Output vibrations, engine - handle, work rpm, z-direction, stiff springs.

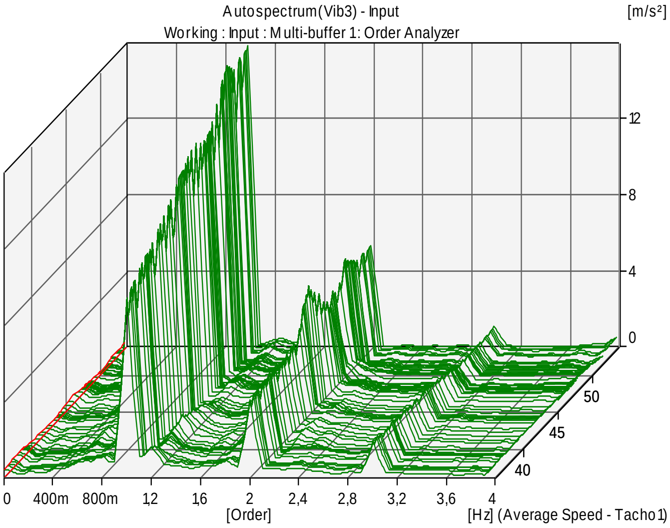

Figure D37a. Input vibrations, cutting arm - feet, idle rpm, x-direction, weak springs.

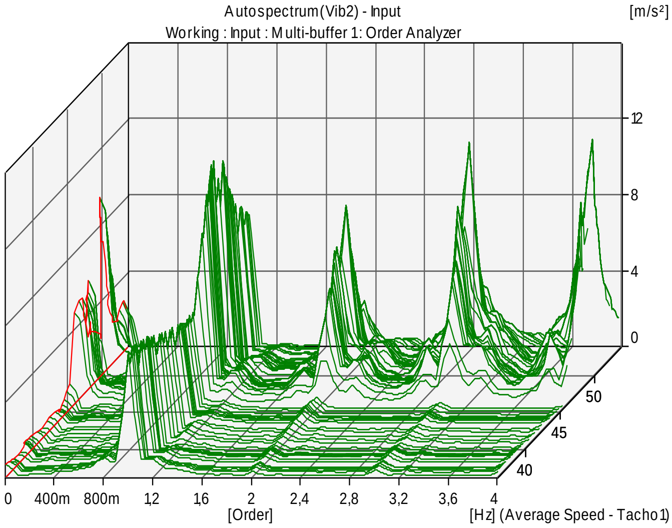

Figure D37b. Output vibrations, cutting arm - feet, idle rpm, x-direction, weak springs.

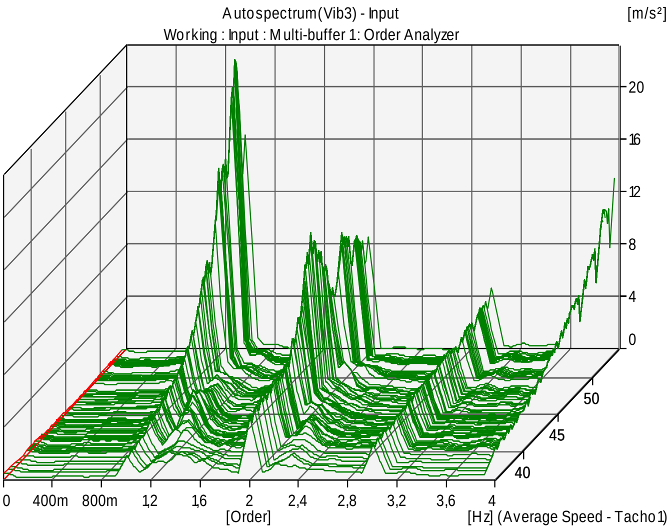

Figure D38a. Input vibrations, cutting arm - feet, idle rpm, y-direction, weak springs.

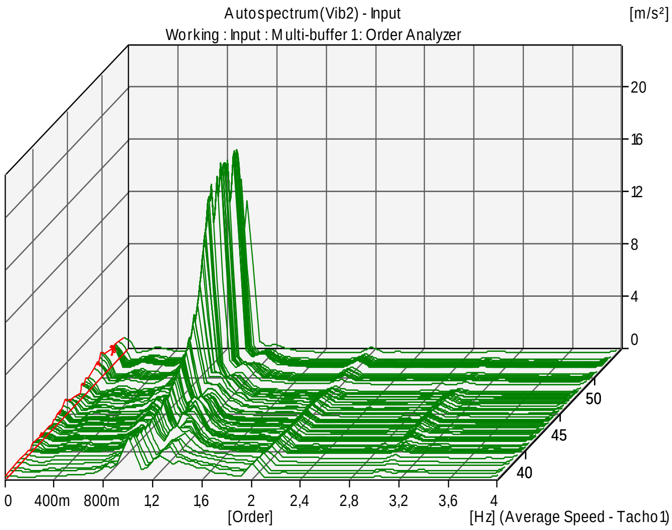

Figure D38b. Output vibrations, cutting arm - feet, idle rpm, y-direction, weak springs.

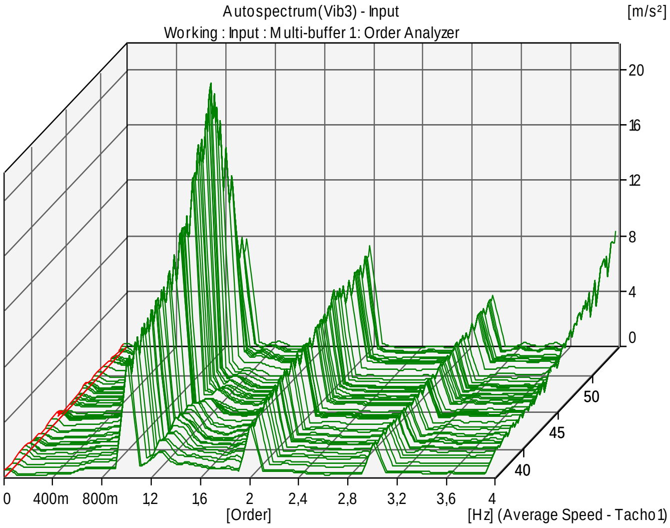

Figure D39a. Input vibrations, cutting arm - feet, idle rpm, z-direction, weak springs.

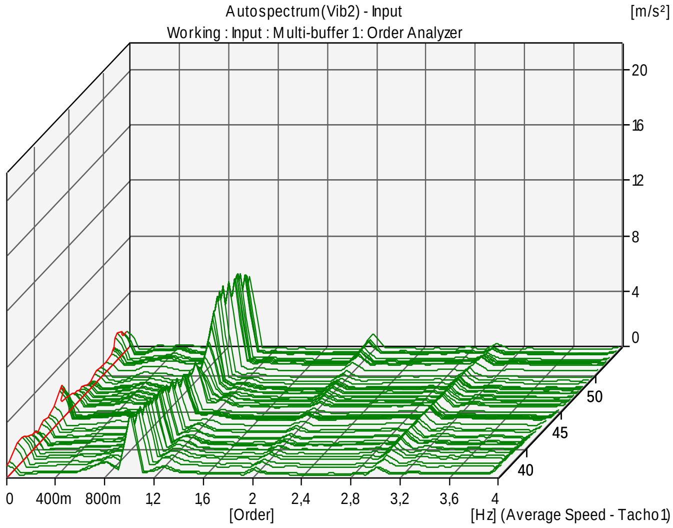

Figure D39b. Output vibrations, cutting arm - feet, idle rpm, z-direction, weak springs.

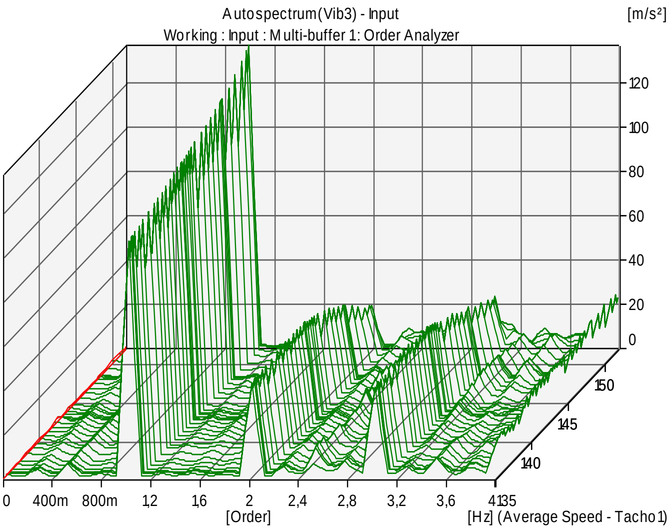

Figure D40a. Input vibrations, cutting arm - feet, work rpm, x-direction, weak springs.

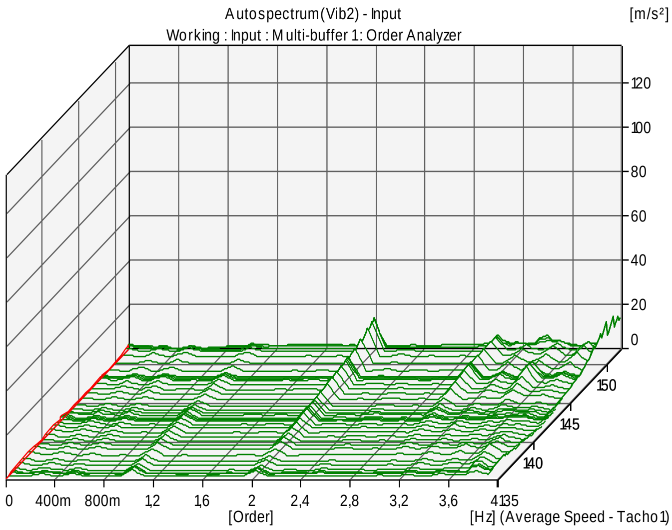

Figure D40b. Output vibrations, cutting arm - feet, work rpm, x-direction, weak springs.

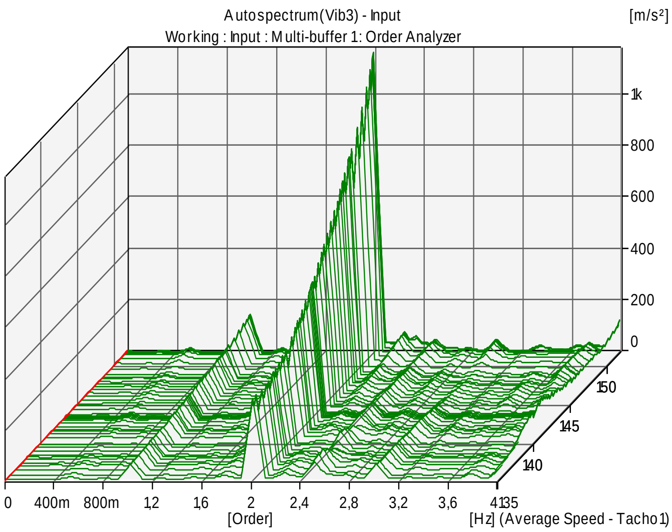

Figure D41a. Input vibrations, cutting arm - feet, work rpm, y-direction, weak springs.

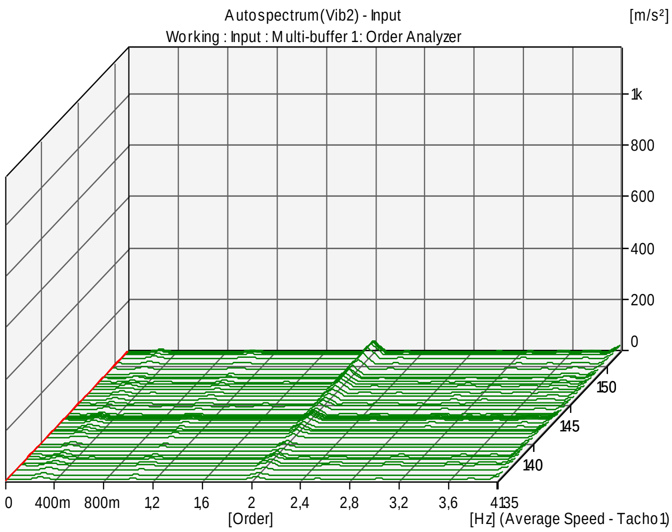

Figure D41b. Output vibrations, cutting arm - feet, work rpm, y-direction, weak springs.

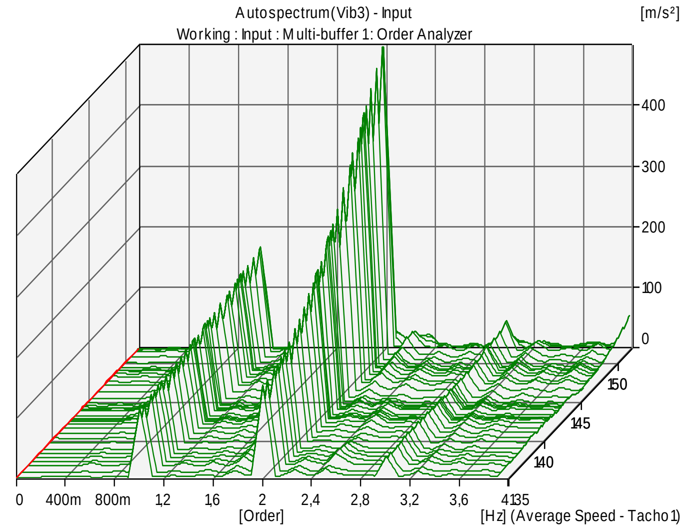

Figure D42a. Input vibrations, cutting arm - feet, work rpm, z-direction, weak springs.

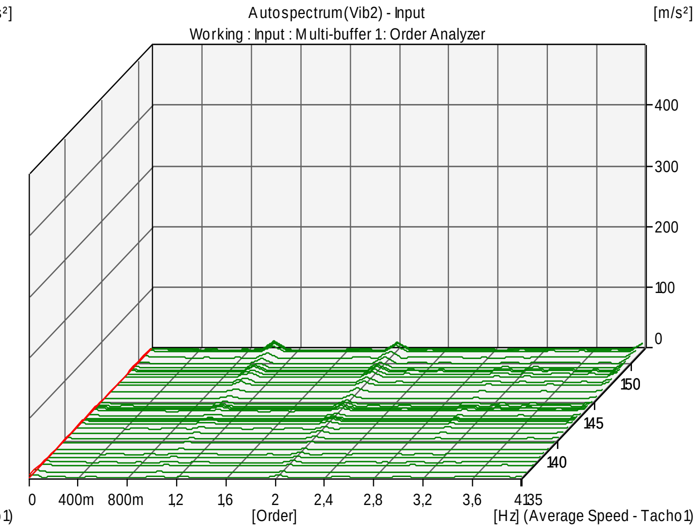

Figure D42b. Output vibrations, cutting arm - feet, work rpm, z-direction, weak springs.

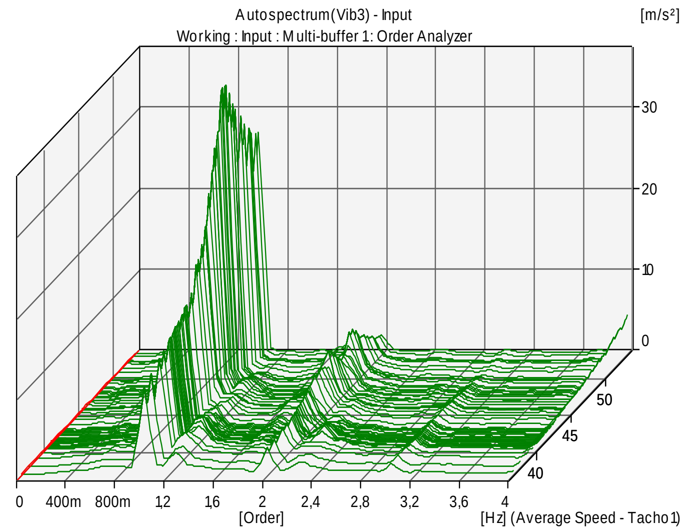

Figure D43a. Input vibrations, air intake - tank, idle rpm, x-direction, weak springs.

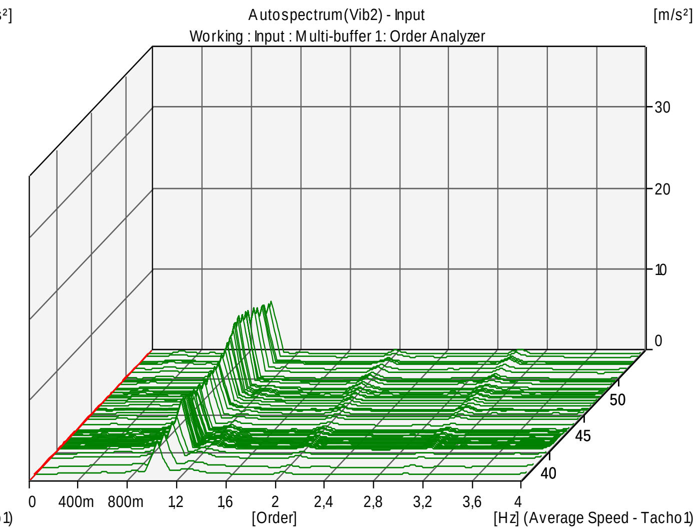

Figure D43b. Output vibrations, air intake - tank, idle rpm, x-direction, weak springs.

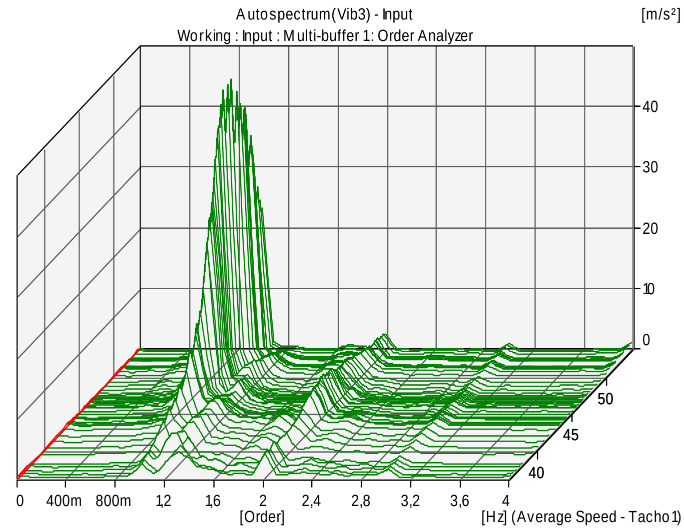

Figure D44a. Input vibrations, air intake - tank, idle rpm, y-direction, weak springs.

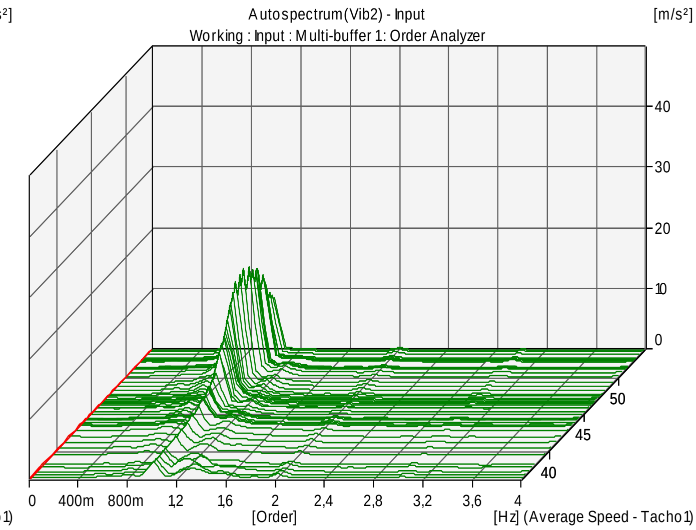

Figure D44b. Output vibrations, air intake - tank, idle rpm, y-direction, weak springs.

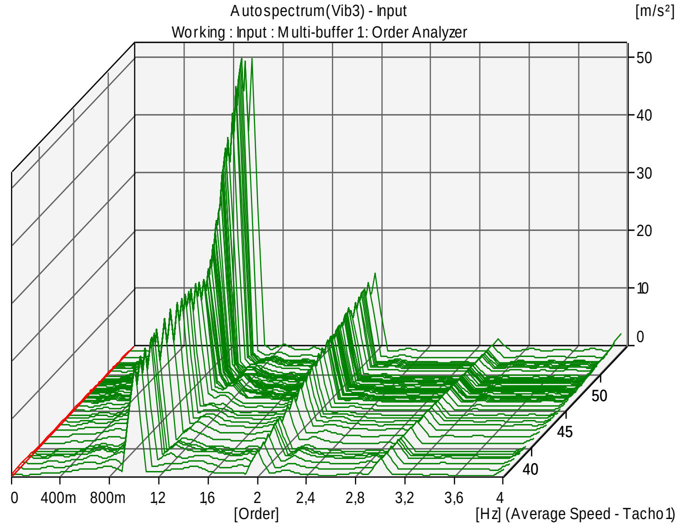

Figure D45a. Input vibrations, air intake - tank, idle rpm, z-direction, weak springs.

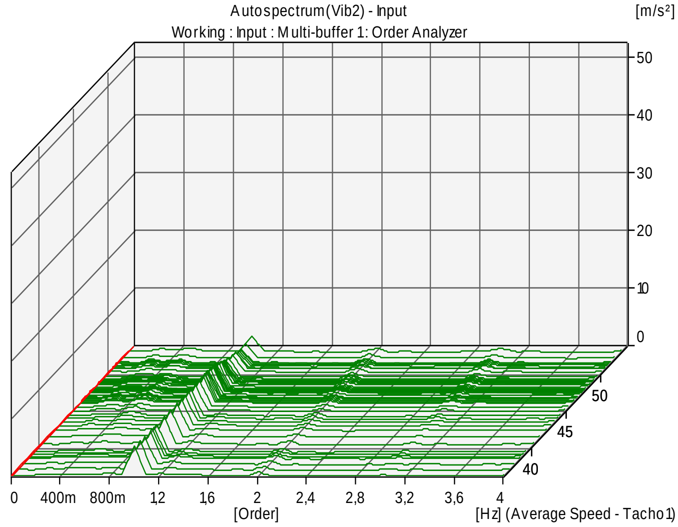

Figure D45b. Output vibrations, air intake - tank, idle rpm, z-direction, weak springs.

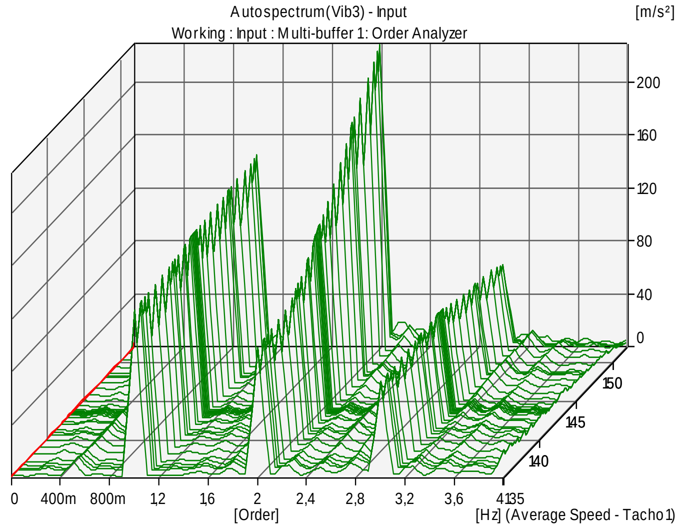

Figure D46a. Input vibrations, air intake - tank, work rpm, x-direction, weak springs.

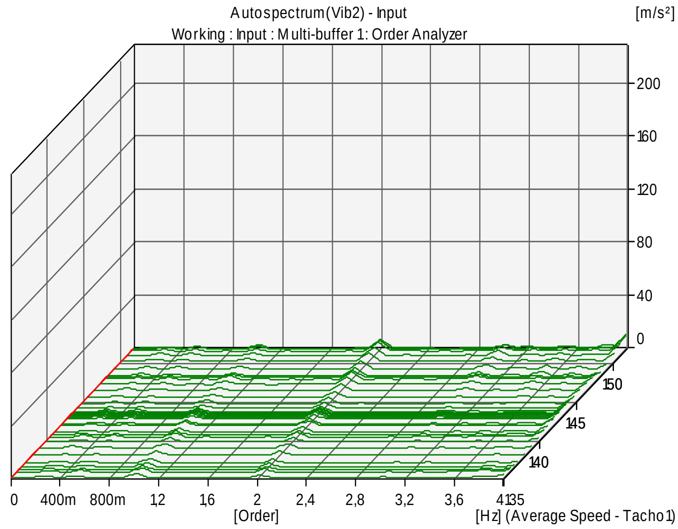

Figure D46b. Output vibrations, air intake - tank, work rpm, x-direction, weak springs.

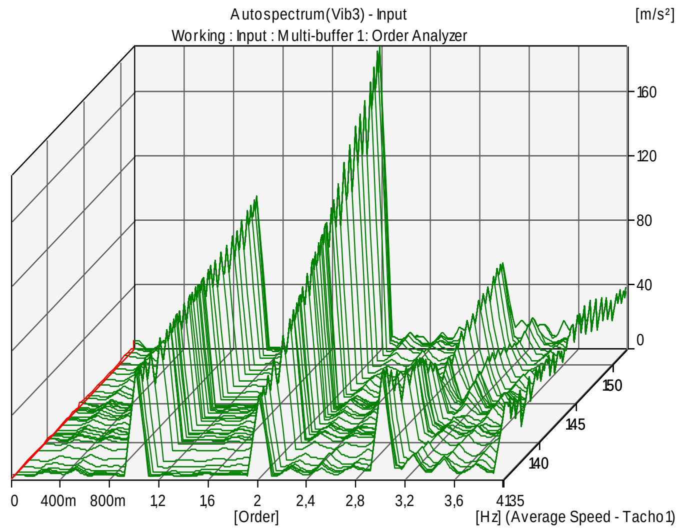

Figure D47a. Input vibrations, air intake - tank, work rpm, y-direction, weak springs.

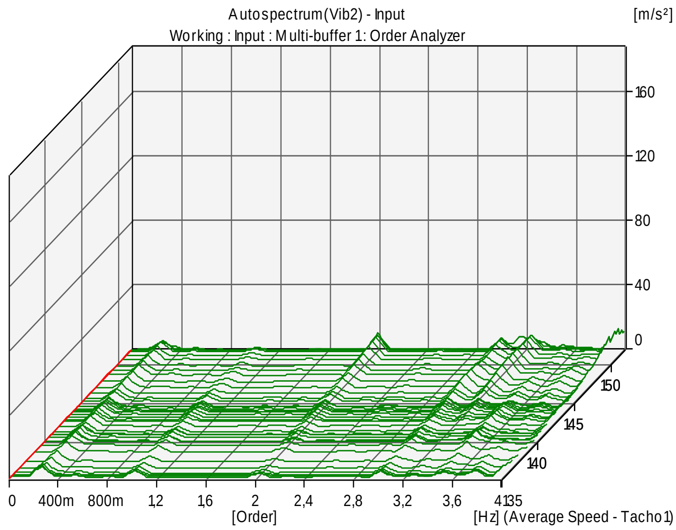

Figure D47b. Output vibrations, air intake - tank, work rpm, y-direction, weak springs.

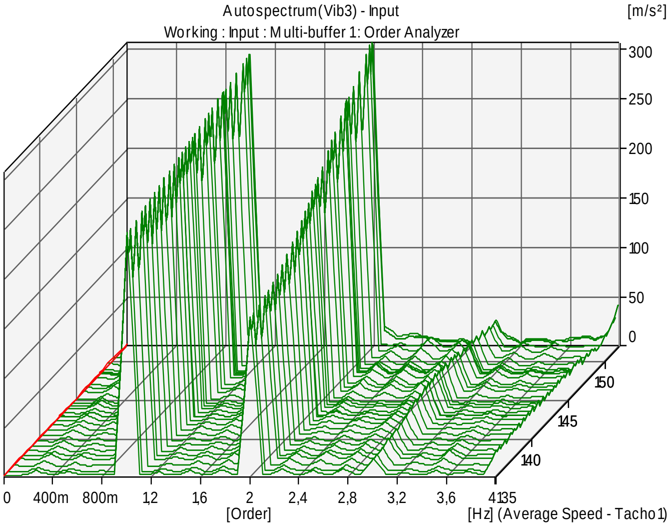

Figure D48a. Input vibrations, air intake - tank, work rpm, z-direction, weak springs.

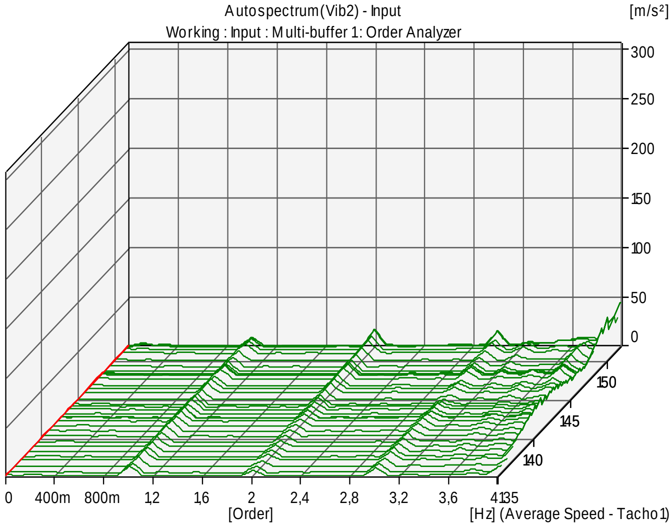

Figure D48b. Output vibrations, air intake - tank, work rpm, z-direction, weak springs.

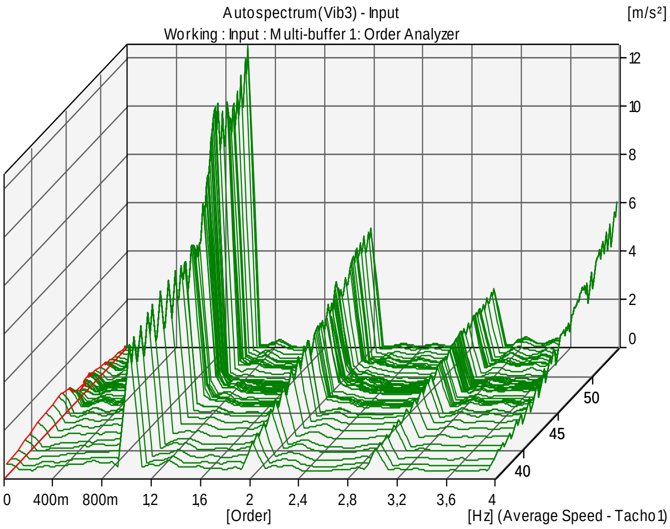

Figure D49a. Input vibrations, engine - handle, idle rpm, x-direction, weak springs.

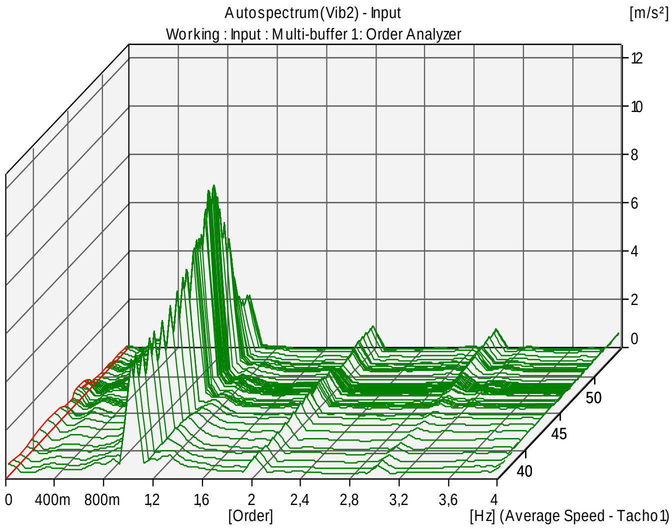

Figure D49b. Output vibrations, engine - handle, idle rpm, x-direction, weak springs.

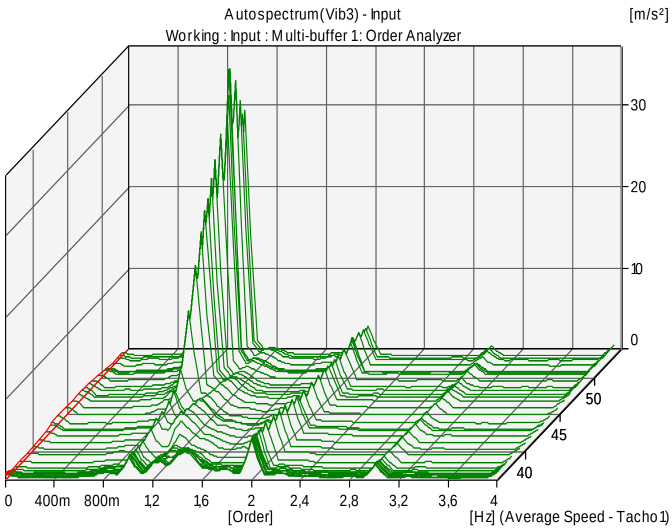

Figure D50a. Input vibrations, engine - handle, idle rpm, y-direction, weak springs.

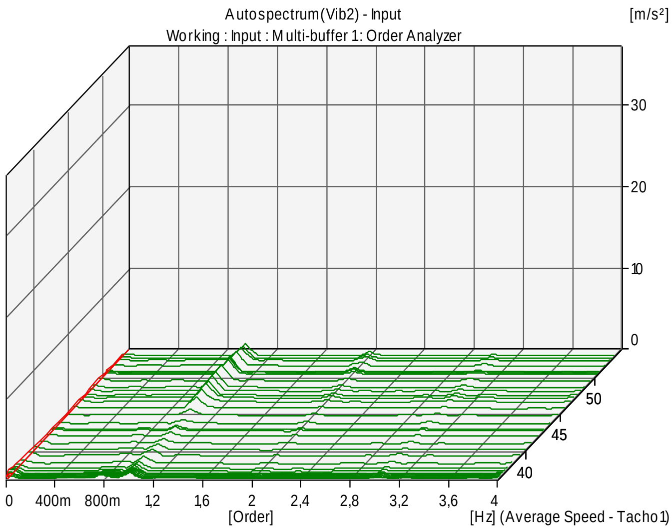

Figure D50b. Output vibrations, engine - handle, idle rpm, y-direction, weak springs.

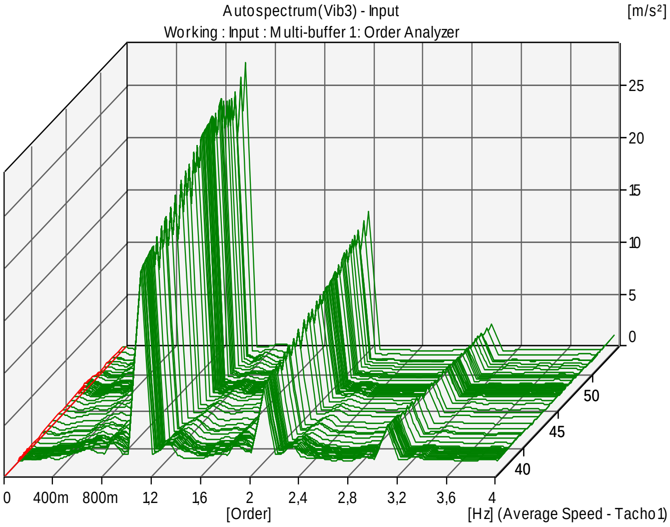

Figure D51a. Input vibrations, engine - handle, idle rpm, z-direction, weak springs.

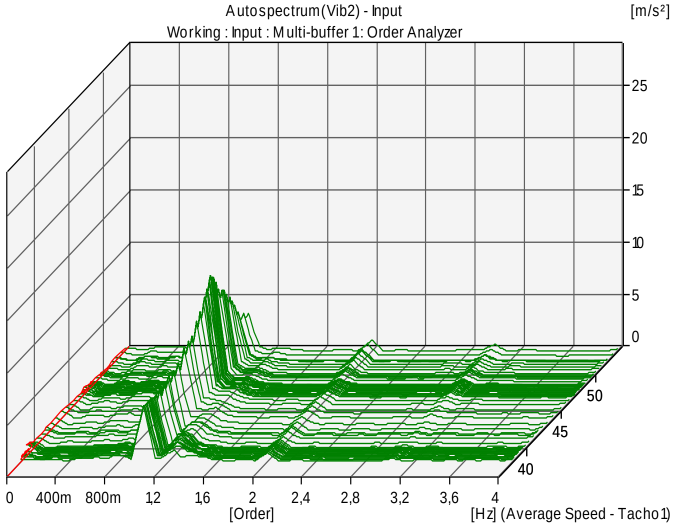

Figure D51b. Output vibrations, engine - handle, idle rpm, z-direction, weak springs.

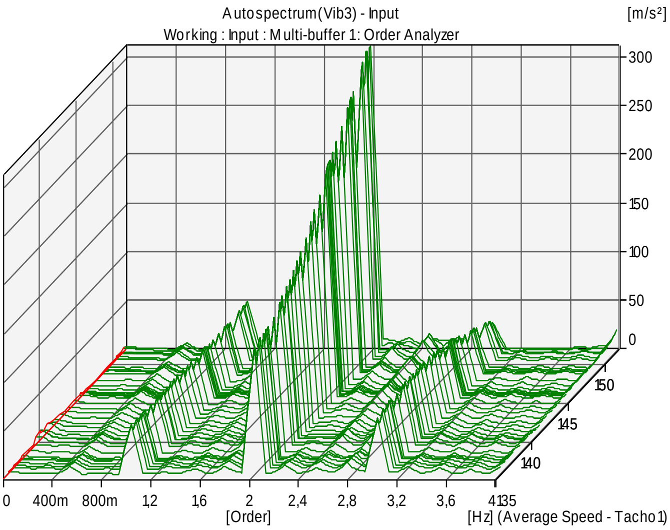

Figure D52a. Input vibrations, engine - handle, work rpm, x-direction, weak springs.

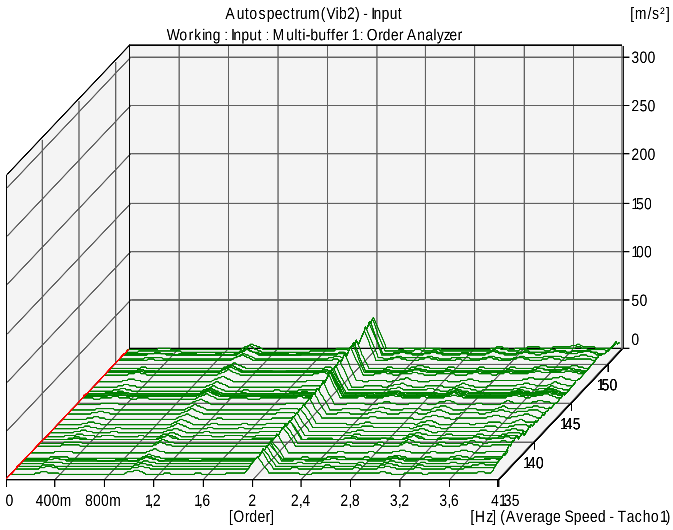

Figure D52b. Output vibrations, engine - handle, work rpm, x-direction, weak springs.

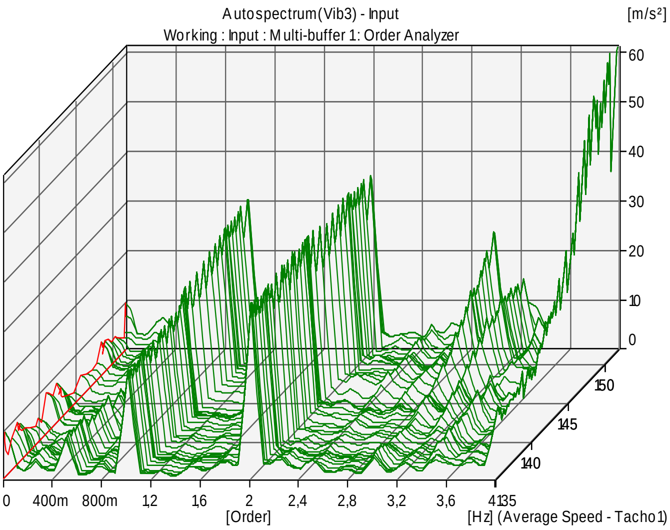

Figure D53a. Input vibrations, engine - handle, work rpm, y-direction, weak springs.

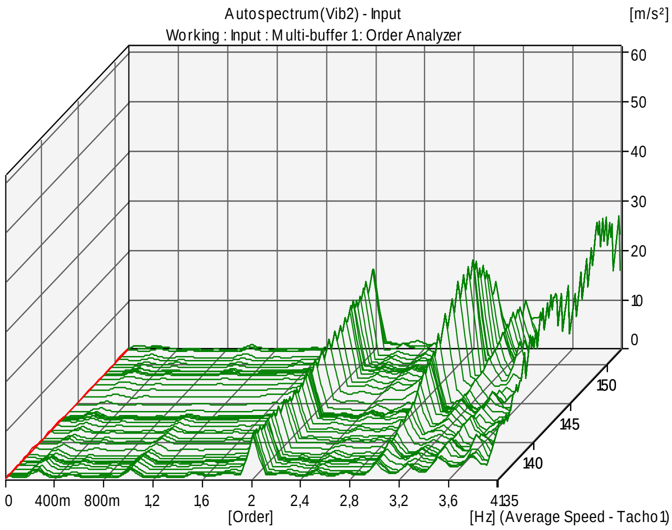

Figure D53b. Output vibrations, engine - handle, work rpm, y-direction, weak springs.

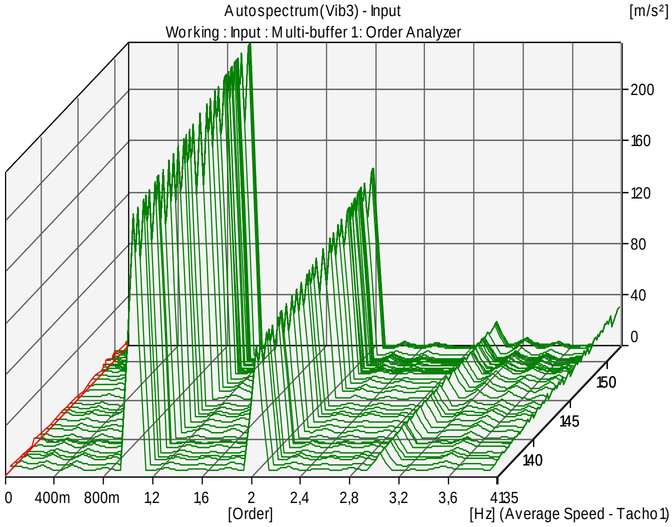

Figure D54a. Input vibrations, engine - handle, work rpm, z-direction, weak springs.

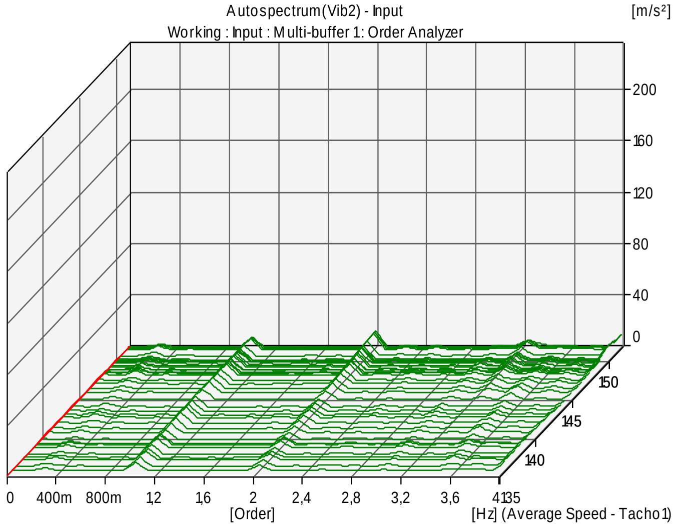

Figure D54b. Output vibrations, engine - handle, work rpm, z-direction, weak springs.

E

Comparison of Input Vibrations

In order to test whether or not individual spring optimisation was possible the vibrations on the vibrating part (engine part) were compared between the different spring sets; weak, standard and stiff.

The result from this comparison is presented in this appendix. Figure E1-E18 shows the time domain vibration curve for orders 0.5, 1, 2, 3 and 4. The green curves show the weak set, the purple curve show the standard and the red curve show the stiff.

This comparison suggested that it was possible to individually optimise the springs without having to consider the other springs of the set. The general shape of the curve is the same between the different sets in all cases and although there are differences they were believed to be small enough to fall within the standard error margin. Only in a few cases such as figure E2 and E11 were considerable differences detected, but only with a single set deviating from the other two, and between these cases there is no clear consistency in which set is the deviating one. This led to the belief that these results were due to random circumstances.

Cutting arm – foot

Idle

Work

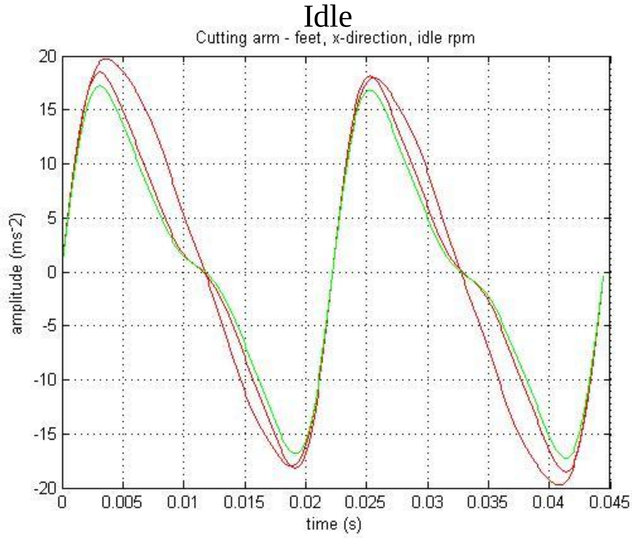

Figure E1. Cutting arm – foot, x-direction, idle rpm.

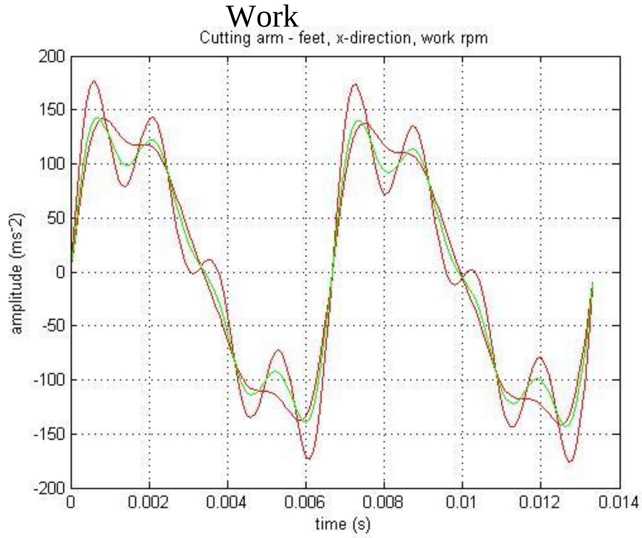

Figure E2. Cutting arm – foot, x-direction, work rpm.

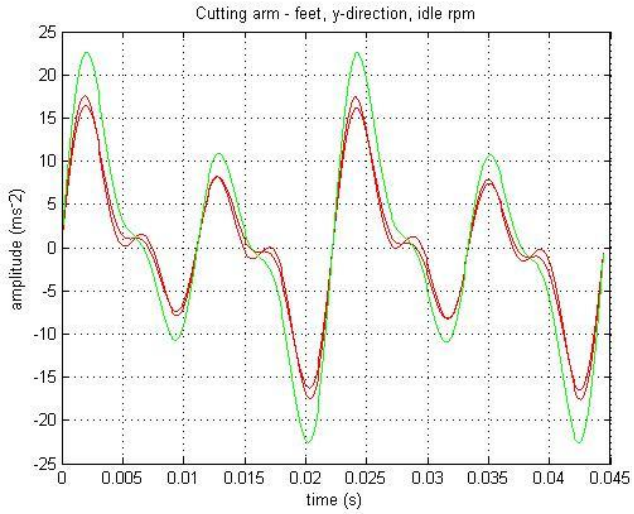

Figure E3. Cutting arm – foot, y-direction idle rpm.

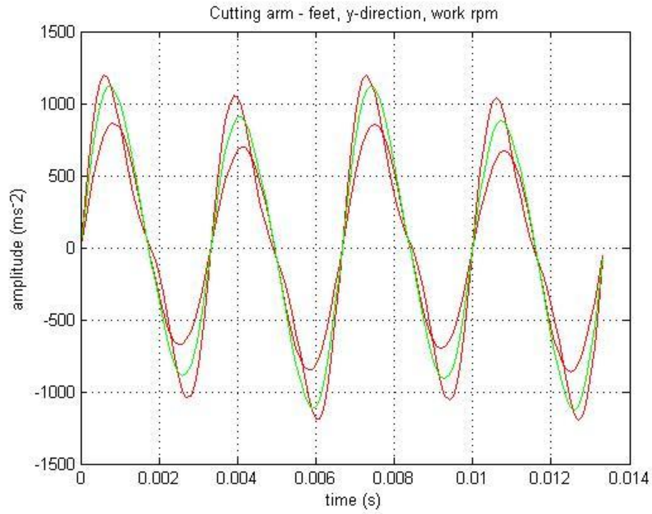

Figure E4. Cutting arm – foot, y-direction work rpm.

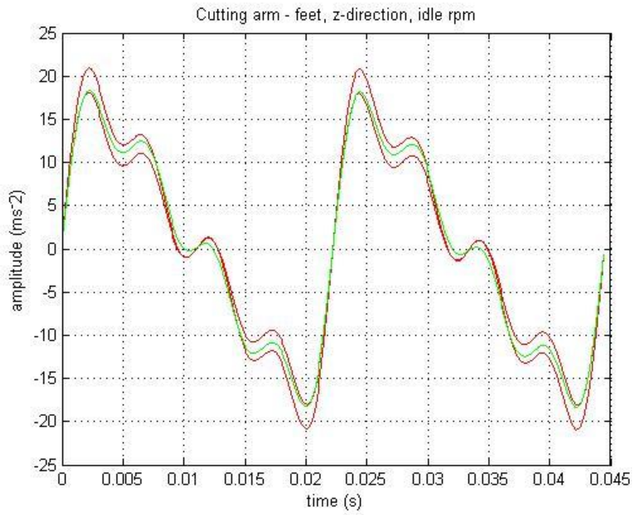

Figure E5. Cutting arm – foot, z-direction idle rpm.

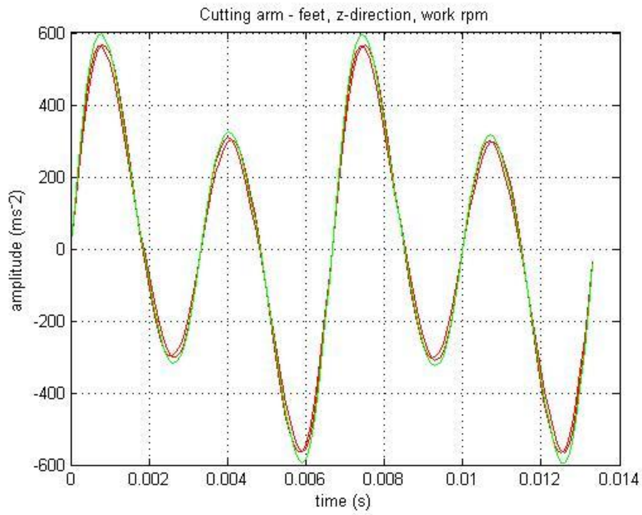

Figure E6. Cutting arm – foot, z-direction work rpm.

Air intake – tank

Idle | Work

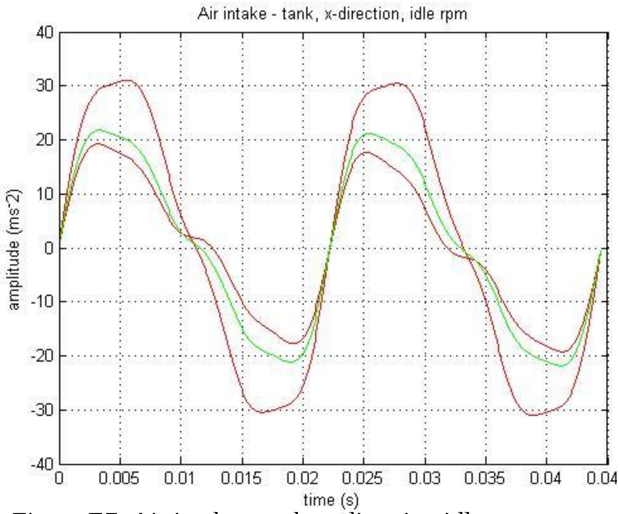

Figure E7. Air intake - tank, x-direction idle rpm.

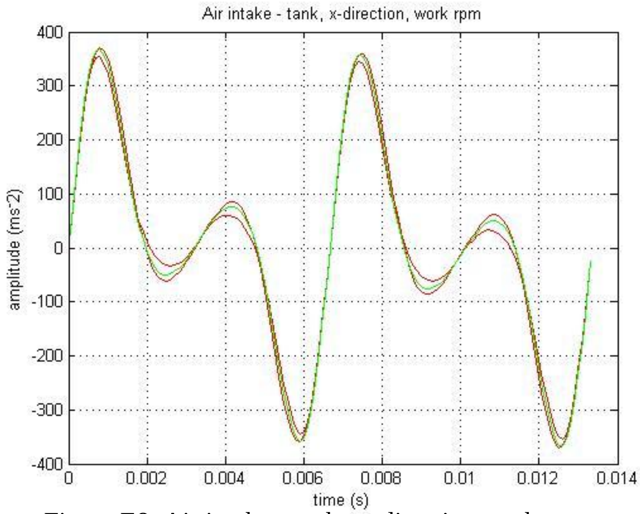

Figure E8. Air intake - tank, x-direction work rpm.

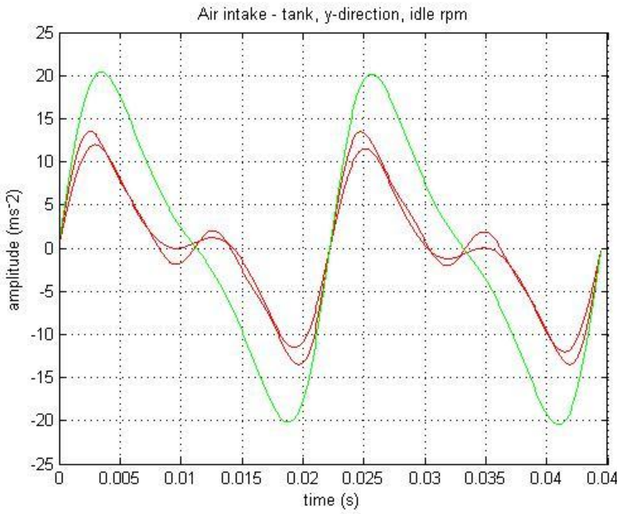

Figure E9. Air intake - tank, y-direction idle rpm.

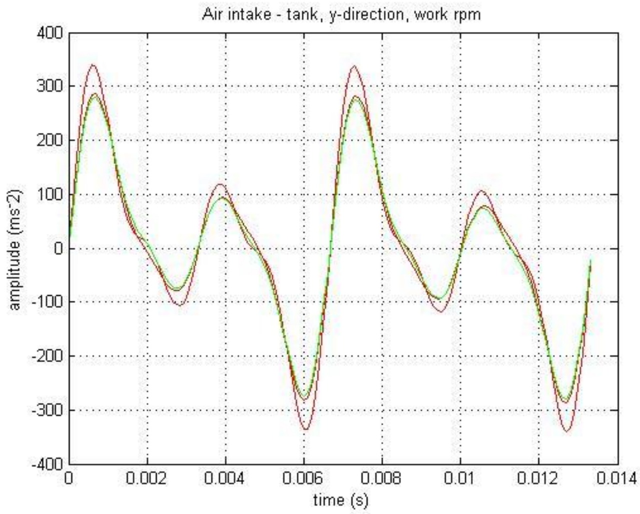

Figure E10. Air intake - tank, y-direction work rpm.

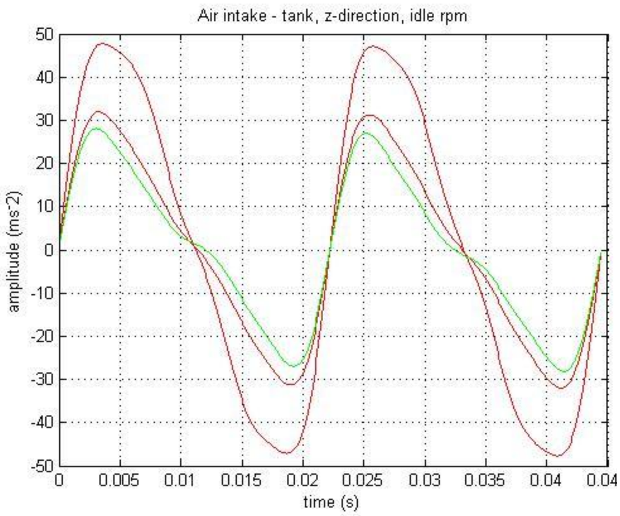

Figure E11. Air intake - tank, z-direction idle rpm.

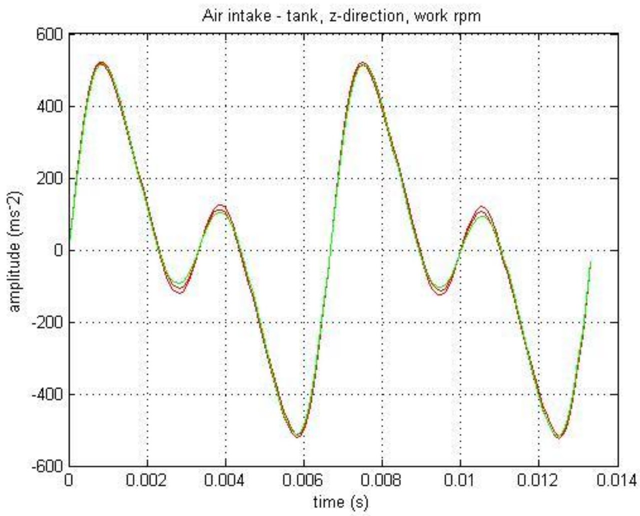

Figure E12. Air intake - tank, z-direction work rpm.

Engine – handle

Idle | Work

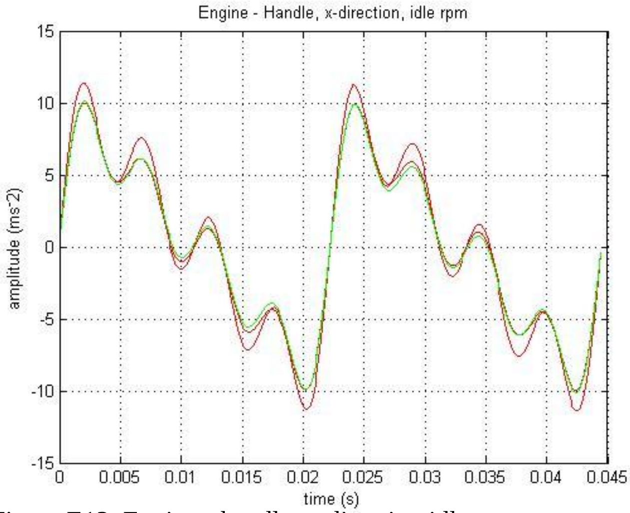

Figure E13. Engine - handle, x-direction idle rpm.

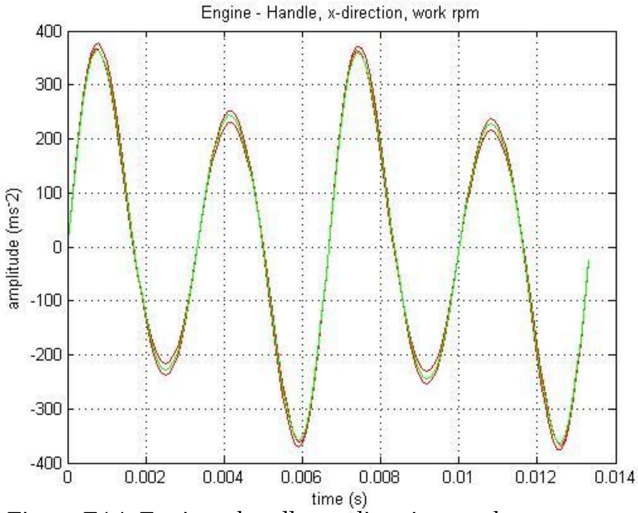

Figure E14. Engine - handle, x-direction work rpm.

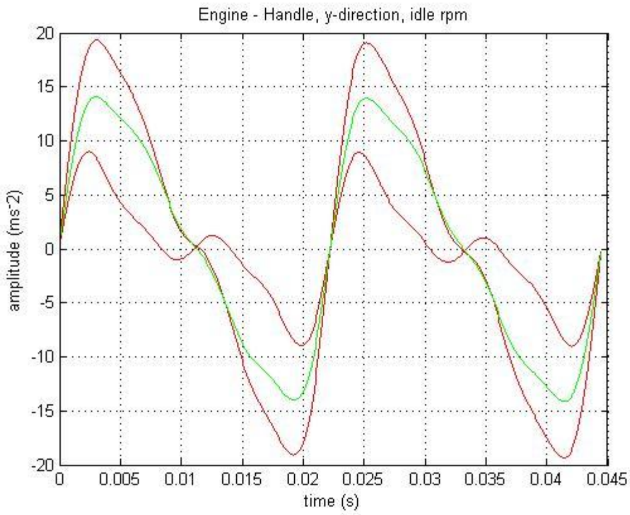

Figure E15. Engine - handle, y-direction idle rpm.

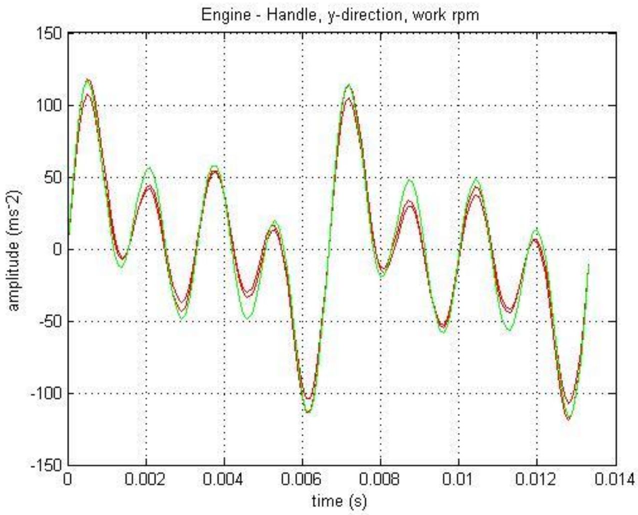

Figure E16. Engine - handle, y-direction work rpm.

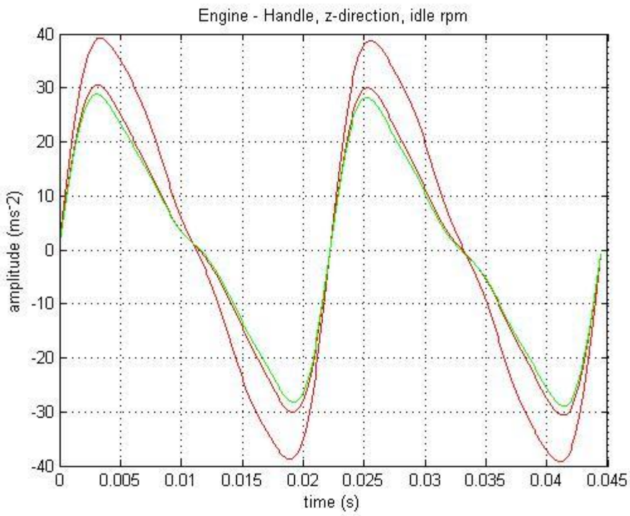

Figure E17. Engine - handle, z-direction idle rpm.

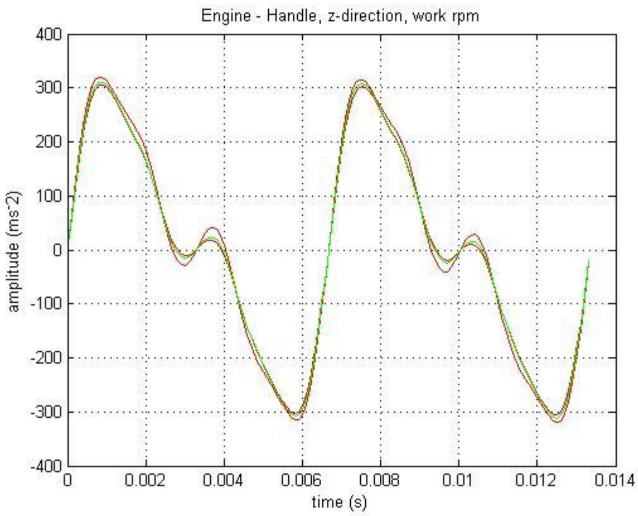

Figure E18. Engine - handle, z-direction work rpm.