YRKESHÖGSKOLAN NOVIA

Flexible mounting – Theory and practice

A vibration study based on Wärtsilä's electrical cabinet

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Bachelor´s thesis

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BACHELOR'S THESIS

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Abstract

This thesis is done on a request from the Noise- and vibration team at Wärtsilä Finland Oyj, Vaasa. The task is to investigate the main electrical cabinet's double flexible mounting which is used on Wärtsilä's main engines.

The purpose for this work is to investigate the double flexible mounting. The double flexible mounting has created challenges in vibration levels on the main electrical cabinet used on Wärtsilä's 32 and 34 engines. This thesis does focus on the origin of the problem rather than focusing on individual engine setups.

The results are based on vibration theory and extensive tests. The theory presents information and know-how to understand the basics of the vibration theory, with focus on understanding the double flexible mounting.

The tests which are all performed in Vaasa, Finland, on different setups and with different methods, is the base for the results which are presented. The results show that even though we have two systems, they move as a single unit due to its configuration and the phenomenon of the double flexible mounting.

Through this work an understanding of the double flexible mounting is given, though it does not solve the problem which it creates. Because of the complicated problem the double flexible mounting creates, recommendation for further work are given which is outside the limitations of this thesis.

Language: English Key words: vibration, vibration theory, flexible mounting

EXAMENSARBETE

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Titel: Flexibel uppställning – Teori och praktik

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Abstrakt

Detta examensarbete är gjort enligt uppdrag av Ljud- och vibrationsgruppen vid Wärtsilä Finland Abp, Vasa. Uppdraget gick ut på att undersöka huvud-apparatskåpets dubbla flexibla uppställning, som används på Wärtsiläs huvudmotorer.

Syftet för arbetet var att undersöka den dubbla flexibla uppställningen. Den dubbla flexibla uppställningen har skapat problem med vibrationsnivåerna på huvud-apparatskåpets som används på Wärtsiläs 32- och 34-motorer. Detta arbete fokuserar på att undersöka problemets ursprung istället för att fokusera på enskilda motoruppställningar.

Resultatet baseras på vibrationsteori och omfattande mätningar. I teorin presenteras information som krävs för att förstå den grundläggande vibrationsteorin, med fokus på att förstå den dubbla flexibla uppställningen.

Mätningarna, som alla har gjorts i Vasa, Finland, på olika uppställningar och metoder, är grunden till resultatet som presenterats i detta arbete. Resultatet visar att även om man har två olika system, så rör sig dessa som att om de vore ett system. Detta på grund av systemets konfiguration och fenomenet som uppstår av den dubbla flexibla uppställningen.

Genom detta examensarbete gavs förståelse i hur den dubbla flexibla uppställningen fungerar, men inget sätt att lösa problemet som det skapar. På grund av det komplicerade problem som den dubbla flexibla uppställningen orsakar, ges det i detta slutarbete rekommendationer om ett fortsatt arbete som ligger utanför slutarbetets avgränsningar.

Språk: engelska Nyckelord: vibrationer, vibrationsteori, flexibel uppställning

OPINNÄYTETYÖ

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Otsikko: Joustava asennus – Teoria ja käytäntö

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Tiivistelmä

Tämä opinnäytetyö on tehty osana Wärtsilä Finland Oyj Vaasan Melu- ja värähtely-työryhmän toimeksiantoa. Toimeksiantona oli tutkia sähkökaapin kaksinkertaista joustavaa asennusta, jota käytetään Wärtsilän päämoottoreissa.

Tämän työn tavoitteena oli tutkia kaksinkertaista joustavaa asennusta. Kaksinkertaista joustavaa asennusta käytetään Wärtsilän 32- ja 34-moottoreiden sähkökaapeissa, joissa se on aiheuttanut ongelmia värähtelytasoilla. Tämä opinnäytetyö keskittyy tutkimaan ongelmien alkuperää yksittäisten moottoriasennusten sijaan.

Tulokset perustuvat värähtelyteoriaan ja laajoihin mittauksiin. Teoriaosassa esitellään tietoa, jota vaaditaan värähtelyteorian perusteiden ymmärtämiseen, fokuksena kaksinkertaisen joustavan asennuksen ymmärtäminen.

Tässä työssä saadut tulokset perustuvat eri asennuksilla ja menetelmillä tehtyihin mittauksiin, jotka on suoritettu Vaasassa. Tulokset osoittavat, että vaikka käytössä on kaksi eri systeemiä, niin ne liikkuvat kuin ne olisivat yksi systeemi. Tämä johtuu systeemin konfiguraatiosta sekä ilmiöstä, joka syntyy kaksinkertaisesta joustavasta asennuksesta.

Tämä opinnäytetyö auttaa ymmärtämään miten kaksinkertainen joustava asennus toimii, mutta se ei tuo ratkaisua siitä aiheutuviin ongelmiin. Johtuen kaksinkertaisen joustavan asennuksen aiheuttamista monimutkaisista ongelmista, tässä opinnäytetyössä annetaan kuitenkin suosituksia jatkotutkimuksille, jotka rajauksen takia jäävät opinnäytetyön ulkopuolelle.

Kieli: englanti Avainsanat: värähtely, värähtelyteoria, joustava asennus

Preface

I would like to thank Wärtsilä for the opportunity to do this thesis, it has been an exciting adventure filled with a lot of new experiences.

A Special thanks to my supervisors, Kari Saine and Kaj Rintanen, who have given me a lot of valuable information, along with fun and educational discussions throughout this thesis.

Thanks to my colleagues throughout this thesis, Claus Paro and Patil Manjunath, who have assisted me with many measurements and the learning of the different equipment used by the vibration group.

Lastly, I would like to thank all other who have supported me, and all who have helped me in any way throughout this thesis, it would not have been possible to achieve this without your help.

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Abbreviations

3D	Three dimensions
AD	Anno Domini, which is Latin for “the year of our Lord” and represents the years after Christ was born
DFT	Discrete Fourier transformation
DOF	Degree of freedom
FEM	Finite element method
FFT	Fast Fourier transformation
FRF	Frequency response function
GW	Gigawatt
Hz	Hertz
LNG	Liquefied natural gas
ODS	Operating deflection shape
PV	Photovoltaic
RMS	Root mean square
Rpm	Rotations per minute
SDOF	Single degree of freedom

Symbols

A_P – Acceleration position

c – Damping constant

c_c – Critical damping

ε $(epsilon)$ – Fractional tensile elongation

ε_l – Longitudinal tension strain

ε_W – Lateral contraction strain

E – Young´s modulus of elasticity

f – Frequency

f_n – Natural frequency in rotations per second

F – Force

F_d – Damping form

γ (gamma) – Shear

G – Shear modulus

k – Stiffness

K – Bulk compression

m – Mass

n – Engine speed in rpm

ν (nu) – Poisson’s ratio

p – Pressure

φ (phi) – Phase angle

s – Constant

t – Time, in seconds

T – One period, in seconds

ΔV $(delta\ V)$ – Compressed volume

V_0 – Original volume

V_P – Velocity position

ω $(omega)$ – Angular frequency

ω_n – Natural frequency in radians per second

W_i – Induced force

W_r – Reflected force

W_t – Transmitted force

x, u – Displacement

$v, \dot{x}, \dot{u}$ – Velocity

$a, \ddot{x}, \ddot{u}$ – Acceleration

X_P – Displacement position

ζ $(zeta)$ – Damping ratio

1. Introduction

The computer revolution took the world by storm and changed everything, old designs quickly became obsolete and new compact designs emerged which all strive to outperform the previous. Engines are becoming smaller and smaller, yet efficiency and power increases. As power increases, so do internal forces which leads to more vibrations and a need for better optimized solutions.

Vibrations and isolation have been studied on engines as long as they have existed, even though it may seem like ages, vibrations still create problems. This is because vibrations have always been an undesired effect on engines as they contribute to the total noise and vibration levels to the surrounding. The demand for a high performing, silent and low vibration level engine creates great challenges.

Especially vibrations from large diesel engines like the ones discussed in this thesis, could have devastating effects on components if they are not isolated good enough from the engine. One of the biggest concerns is electrical components. The digital revolution we live in constantly creates valuable tools we want to utilize on our engines. The problem with electrical components is that they are sensitive to vibrations and can break down instantly, and in absolute worst-case scenarios could create a total engine stop. Therefore, it is important to isolate the electrical components well to reduce vibration levels. Though it is not easy choosing an isolator and this will be discussed later in this thesis.

1.1. Wärtsilä

Wärtsilä is a global leader in advanced technologies and complete lifecycle solutions for the marine and energy market. They currently employ about 18,000 employees across 70 countries. Wärtsilä had an installed base of around 180 GW in 2016. They are listed on NASDAQ in Helsinki and their net sales totalled 4.8 billion € in 2016. [1]

Energy solutions

Wärtsilä energy solutions supplies flexible combustion engine based power plants, Utility-scale solar/PV power plants, LNG terminals as well as the distribution systems. Wärtsilä had at the beginning of 2016, 63 GW of installed power plant capacity in 176 countries around the world. [2] To put this into perspective, an average nuclear reactor yields a little less than one GW of electrical power. [3]

Marine solutions

Wärtsilä marine solutions provide high flexibility in their portfolio delivering not only engines but having a presence in all the major marine and offshore oil & gas segments. They also have an unmatched track record of dual-fuel technology. [2] Their newest engine, the Wärtsilä 31 received a Guinness world record as the most efficient 4-stroke engine. This while keeping fuel efficiency at an all-time low and limits far below current environmental requirements. [4]

Services

Wärtsilä services provide their customers with lifecycle long support and services of all their installations. The numbers 11,000 professionals supporting 12,000 customers annually is unmatched in the industry. Their portfolio containing everything from services and spare parts to complete operational, maintenance and optimisation services is always being developed. [1]

1.2. Thesis specification

This thesis is done on a request to investigate the flexible mounting used on the main electrical cabinet on Wärtsilä engines. The main cabinet is currently causing problems on engines mounted as a main engine, this because of its double flexible mounting setup. Standard measurement tests noticing high vibration levels and recent complaints from customers have created an interest in solving this problem quickly.

This thesis was made to get a better understanding on how the double flexible mounting behaves. This is to better know how electrical components should be isolated from similar vibration setups in the future.

This thesis will be limited to the main cabinet used on Wärtsilä 31, 32 and 34 engines being produced at the factory in Vaasa. It is limited only to its physical properties and does not account for anything but its weight inside the cabinet. It is also limited to only investigate simple and double flexible mounting systems.

1.3. Disposition

These are the chapters included in this thesis and shortly what they contain.

- Chapter 1 introduces the reader to the subject and the specification of this thesis.
- Chapter 2 tells the basic history and theory about vibrations, as well as necessary information about vibrations needed for the reader to understand this thesis.
- Chapter 3 tells about the equipment and measurements.
- Chapter 4 explains the methods used to complete this thesis.
- Chapter 5 presents the results.
- Chapter 6 presents the conclusions.
- Chapter 7 gives recommendations on future work regarding this subject.
- Chapter 8 discusses the results and the conclusion.

2. Theory

Vibrations probably started to interest human kind when the first instruments were created. Math and physics regarding vibrations did not exist before the 17th century but there are artefacts. For example, a fully functional seismograph invented in China in AD 132, not only capable of telling there is an earthquake coming but also the direction of it. This tells that there has always existed an understanding for vibrations. [5]

Galileo Galilei, who is considered one of the founders of modern experimental science, also studied vibrations in form of a pendulum. Inspired by Galileo a lot of scientist have studied vibrations to this day. [5]

The human brain has worked as a vibration analyser for thousands of years. Even to this day many machine operators and maintenance people can diagnose a machine by simply touching it or using some sort of handle to transmit sound from eventual bearings to the ear. The only thing human hearing or touch cannot do, is to put a number on the vibration levels, and this led to the first vibration meters in the 1950s. [6]

In the 1950s the first electrical vibration analysers and computer models was created, but it was not until the 1980s and the beginning of the computer revolution, before more complex vibrations could be analysed with the help of portable vibration analysers. [5] [6]

2.1. Basic vibration theory

Periodic motion or simply vibrations, is something we are familiar with in our day to day life. Whatever it is a heart beating, engines running, a pendulum swinging back and forth or industrial machinery, they all create vibrations. Throughout this thesis though I will focus on mechanical vibrations, although mechanical vibrations have strong parallels with many other fields such as electricity or thermodynamics.

Very often are real-life vibration problems very complicated. A single motion, for example a pendulum, is easy to understand. Though more complex structures like a car engine traveling on a rough road requires a lot more data and analysis. As an engineer it is important to be able to break down a more complex structure into smaller simpler structures. The simplified system can then be used to create and calculate the main features of the real structure and this is especially useful to present and find real problems. [7]

Excitation and forces

All vibrations or some kind of periodic motions are created from a corresponding force, for example the counterweight in a grandfather clock is keeping the pendulum swinging. But there are far more complex forces, for example the internal forces of a combustion engine, where factors like: gas forces, misalignment, rotating masses, oscillating masses and the crank star all have a great influence on the total vibration levels on the engine. [8]

Frequency

The number of times a complete cycle happens during one second, also known as hertz (Hz). The motion may consist of a single re-occurring motion, like a tuning fork or as multiple occurring simultaneously motions, for example the vibrations occurring in a combustion engine.

The frequency can easily be calculated with the formula:

$$f = \frac{1}{T} \tag{1}$$

Where T represents the time for a complete cycle and is measured in seconds, this is also known as one period. [7]

Natural frequency

The phenomena when you hit a tuning fork and it always vibrates at the same frequency is a very good example of natural frequency. When the tuning fork is excited it will try to oscillate at its natural frequency, if its mass or stiffness is changed then the natural frequency will also be altered. It is also important to remember that changing the structures shape, even though the mass stays the same, will alter the stiffness of the structure. Meaning that all changes create a new natural frequency which can be seen in the formula:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \qquad (2)$$

Where f_n is the natural frequency in hertz, k is the stiffness (Newton/meter) and m is the mass (kg). [9]

Natural frequencies can be found in every structure. Therefore, a structure which have several components will have several natural frequencies as they all want to vibrate at their own natural frequency.

Resonance

When the excitation force approaches the same frequency as the natural frequency of the structure we are approaching the resonant frequency. When approaching the resonant frequency, amplitude will start to build up, creating very high vibration levels. Most vibration problems are related to resonance. [9]

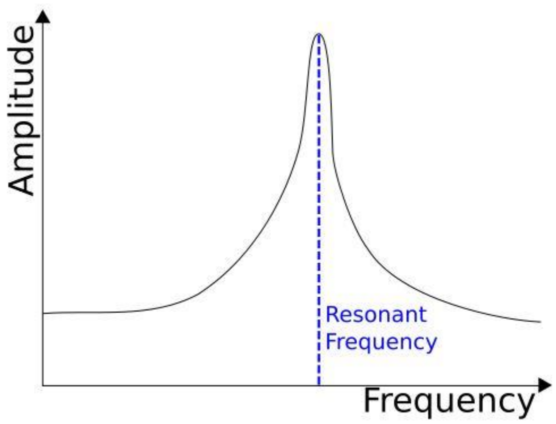

Figure 1. The resonance phenomena. *[10]*

As can be seen on Figure 1, the amplitude of the excited object increases as the frequency is near the resonant frequency. [9]

Order

When speaking about a combustion engine there will be several excitation frequencies. They are normally expressed as orders, where the 1st order is the running speed of the engine and all orders are multiples of the engine running speed. The frequency of the kth order is:

$$x * \frac{n}{60} \text{ [Hz]} \tag{3}$$

Where X is the order and n is the engine speed in rpm. The engine speed is divided by 60 (seconds/minute) to get the number of revolutions per second, Hz. [9]

Firing frequency

The firing frequency tells how often cylinders fire in an engine. In a 2-stroke single piston engine, it will fire every time the piston has done a single revolution. In a 4-stroke single piston engine, it will fire every second revolution. To be able to calculate the order of the firing frequency we need the number of cylinders per row. A line engine having just one row while for example a V engine has two. The order can then be calculated with the formula:

$$Firing\ frequency\ order = \frac{Number\ of\ cylinders\ in\ a\ row}{2\ (in\ 4{-}stroke\ engines)} \tag{4}$$

The highest levels of torque variation occur at the engines firing frequency or at harmonic orders of the firing frequency. For example, a 6-cylinder line engine or a 12-cylinder V engine is going to have its firing frequency at order 3, assuming it is a 4-stroke engine. It will also have harmonic orders of the firing frequency at order 6,9,12 and so on. [8]

2.2. Vibration theory

2.2.1. Free Vibration

All systems containing mass and elasticity is capable of vibrations. The simplest system is illustrated as a spring and a mass, where the spring stiffness is expressed as k (N/mm) and the system is restricted to a single degree of freedom. This means that all motion is describe with a single coordinate x, a more in depth of degrees of freedoms will be discussed later in this thesis. When this system is excited it will oscillate at its natural frequency f_n. [11]

When speaking about vibrations, Newton's second law is the first step in analysing the motion of the system. In the system static position Δ, where the spring is just holding the mass still, and the spring force $k\Delta$ is equal to the gravitational force *w* acting on the mass *m* then:

$$k\Delta = w \tag{5}$$

If we then pull down the spring with the distance *x*, with *x* chosen as positive in the downward position, then force, velocity and acceleration are also positive in downward direction. Newton's second law of motion is then applied on the mass *m*:

$$m\ddot{x} = \sum F = w - k(\Delta + x) \tag{6}$$

Where $\ddot{x} = \frac{d^2 * x}{dt^2}$ which is equal to the acceleration of the system.

It is known from equation five that $k\Delta = w$, therefore we get the formula:

$$m\ddot{x} = -kx \tag{7}$$

Which is normally used in standard form:

$$m\ddot{x} + kx = 0 \qquad [11] \tag{8}$$

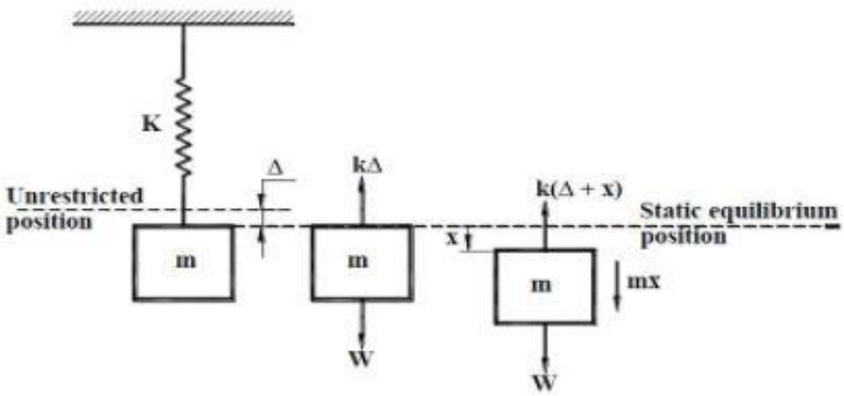

Figure 2. A free vibration system with free-body diagram. *[12]*

Though when speaking about a free vibration system in theory, the system will continue to vibrate for eternity, because it does not have any damping, but in lifelike situations it will always have damping, and will eventually stop vibrating. For example, a guitar string when excited will have a free vibration oscillating at its natural frequency but because of losses when the string swings back and forth the string loses momentum and will eventually fade out. This will be further discussed in chapter 2.2.3.

2.2.2. Forced vibration

Forced vibrations can be found in almost all multi-component structures. This is because if one component within the structure vibrates. It will transfer the vibrations out to the other components. For example, a combustion engine, the rotating masses, gas forces and misalignment of the crankshaft all create vibrations. They also lead these vibrations to the surrounding components forcing them to vibrate at the same frequency as themselves, therefore named "forced vibrations".

2.2.3. Damped vibration

As already stated in the free vibration chapter, there is no system that will oscillate as a free vibration for eternity, this is because of damping which the object is exposed to. The equation for a simple damped can be further developed from equation 8. The equation of motion in general form will be described as:

$$m\ddot{x} + F_d + kx = F(t) \tag{9}$$

Where $F(t)$ is the excitation and F_d is the damping form. Describing the damping form is very difficult and only ideal damping models can be assumed. Therefore, these often result in satisfactory predictions of the response.

The damping force can be expressed as:

$$F_d = c\dot{x} \qquad [13] \tag{10}$$

Where c is a constant of proportionality, typically designated by a dashpot as can be seen in Figure 3.

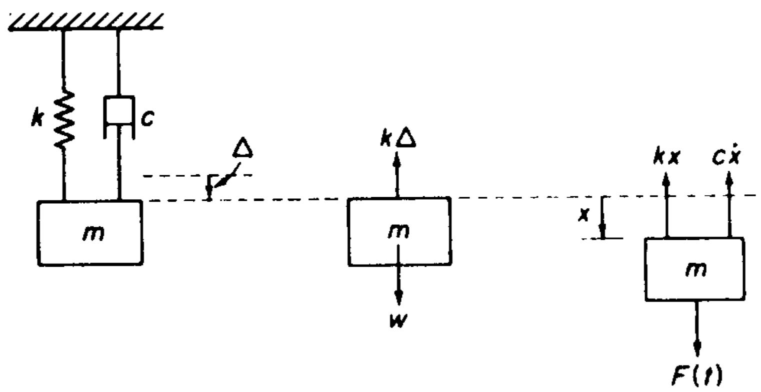

Figure 3. A damped vibration system with free-body diagram. *[14]*

From the free body diagram, the equation of the motion is seen to be:

$$m\ddot{x} + c\dot{x} + kx = F(t) \tag{11}$$

2.3. Damping

A damper is used to isolate a system from an excitation structure. Even though a damper cannot remove all vibrations, the levels are lower when a vibration damper is correctly used. If the wrong damper is installed the vibration levels could become even worse. This is when the rubber elements have not been carefully selected and are for example at resonance with the excitation force. [9]

Formula 11 can be written in its homogeneous state that will give some understanding to the role of damping.

$$m\ddot{x} + c\dot{x} + kx = 0 \qquad (12)$$

We approach the problem by substituting, $x = e^{st}$ and solving as a second-degree equation.

$$s^2 + \frac{c}{m}s + \frac{k}{m} = 0 \qquad (13)$$

This is known as the characteristic equation, which has two roots, (14).

$$s_{1,\ 2} = -\frac{c}{2m} \pm \sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}} \qquad (14)$$

The general solution is given by the equation:

$$x = Ae^{s_1 t} + Be^{s_1 t} \qquad (15)$$

Where A and B are constants from the initial conditions $x(0)$ and $\dot{x}(0)$. Formula 14 is substituted into 15 which gives:

$$x = e^{-\left(\frac{c}{2m}\right)t} * \left(Ae^{\sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}} * t} + Be^{-\sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}} * t}\right) \qquad (16)$$

The term $e^{-\left(\frac{c}{2m}\right)t}$ is an exponential function of time. The term inside the bracket on the other hand varies, its numerical value can be zero, positive or negative. When $\left(\frac{c}{2m}\right)^2$ is larger than $\frac{k}{m}$, then the exponents in equation 16 are real numbers and no oscillation is possible. This is referred to as **overdamping.** This means that the damping is so strong that it takes longer than necessary to return to the equilibrium position. [13]

When the term $\left(\frac{c}{2m}\right)^2$ is smaller than $\frac{k}{m}$, then the exponents in equation 16 becomes an imaginary number $\pm i\sqrt{k/m - (c/2m)^2} * t$. This means the terms are oscillatory and this case is referred to as **underdamping.** This is when the damping is too weak and shoots over the equilibrium position and oscillates back and forth until it stops. [13]

When the term $\left(\frac{c}{2m}\right)^2$ is equal to $\frac{k}{m}$ the radical in formula 16 is equal to zero. This is known as **critical damping**. This is the shortest time before the mass returns to the equilibrium position and stays there.

$$c_c = 2\sqrt{km} = 2m\omega_n \tag{17}$$

Where ω_n is the natural frequency in radians per second.
Damping is often expressed by the ratio of the actual damping over critical damping.

$$\zeta = \frac{c}{c_c} \tag{18}$$

Where: $\zeta > 1$ is overdamped, $\zeta = 1$ is critically damped and $\zeta < 1$ is underdamped. [13]

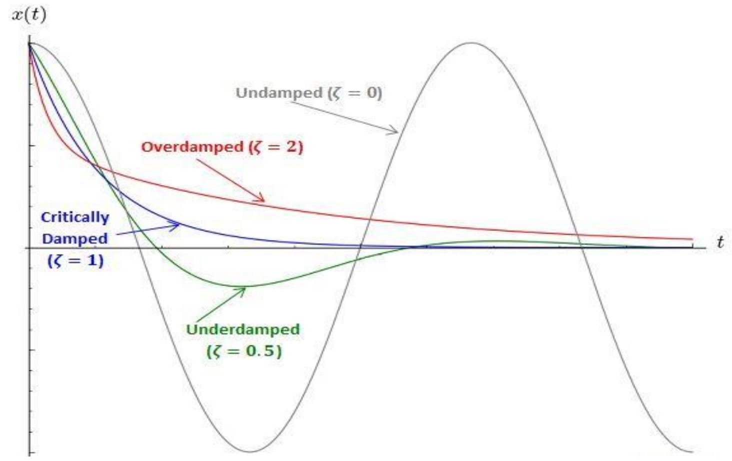

Figure 4. Damping ratio visualized. *[15]*

2.3.1. Isolators

Vibration isolators exists in many materials and shapes. The most common ones are rubber elements and steel springs, even though there are more expensive options. For example, hydraulic or pneumatic vibration controls.

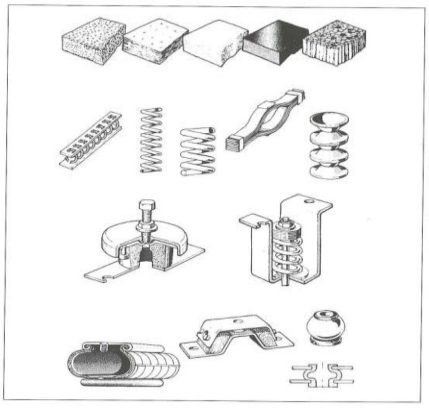

Figure 5. Example on commercially available dampeners. *[36]*

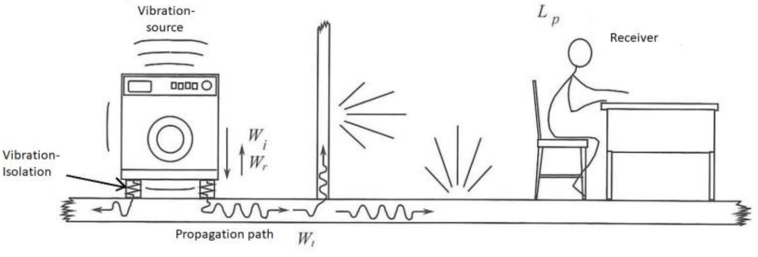

Figure 6. Example on a machine where vibration isolation is used. Where W_i is the induced power from the source towards the vibration isolation, W_r is the power reflected back against the source and W_t is the power transmitted to the floor. *[35]*

2.3.2. Rubber isolators

Rubber elements are the most used vibration isolators, and they exist in all types of sizes, materials and forms, with examples that can be seen on Figure 5.

Natural rubber is created from the latex of the Hevea brasiliensis tree. Before coagulation the latex is stabilized with preservatives and afterwards vulcanized, typically with sulphur. [16]

Elastomers, which refers to both natural and synthetic rubber, describes better the term we know as rubber, because in the last decades there has been a very high increase of synthetic rubber components. [17]

Elastomers can be designed to have different stiffness in non-linear directions. Rubber is also good at converting some energy into heat when deflected, the phenomena known as hysteresis. Natural rubber without filler exhibits very little hysteresis, providing damping to a system subjected to vibrational forces and therefore so fitting to use as a vibration isolator. [18]

The only downside of rubber elements is that they are sensitive to aging, where oxidation and temperature has the most influence, but also chemicals and fatigue speeds up the process that creates cracks in the rubber elements. This results in a stiffness change, meaning especially small rubber components that are exposed to a lot of stress must be changed with regular intervals to make sure their stiffness and characteristics stay within given tolerances. [16]

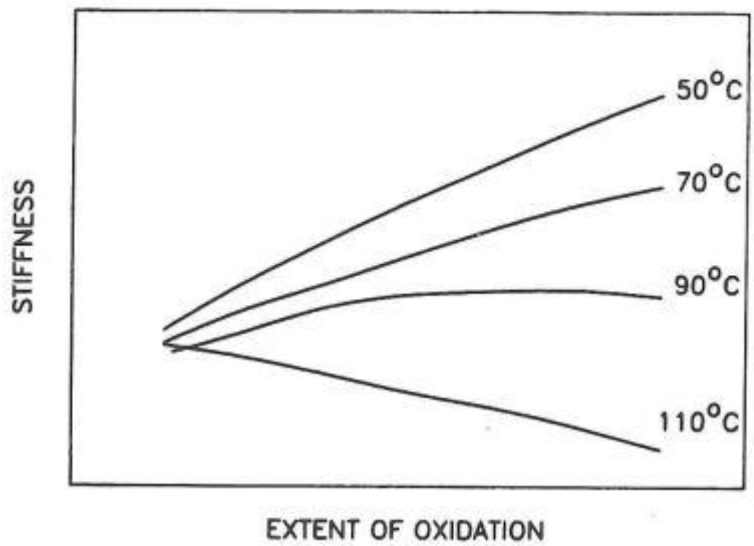

Figure 7. Stiffness of a natural rubber measured after being exposed to oxidation at given temperatures. *[19]*

Elastic material that are isotropic in their undeformed state like rubber, can be described by two elastic constants. In the first constant K is the bulk compression, ΔV is the compressed volume and V_0 is the original volume as can be seen in Figure 8 A). [19]

$$p = K\left(\frac{-\Delta V}{V_0}\right) \tag{19}$$

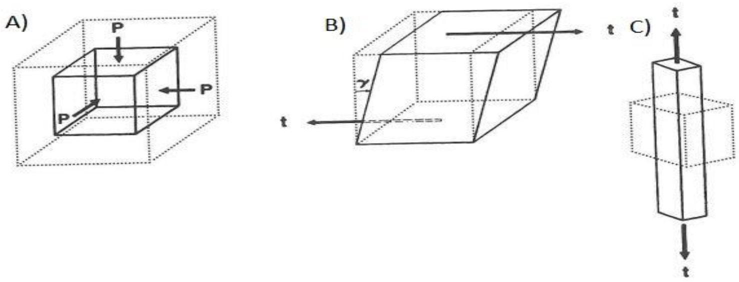

Figure 8. A) Bulk compression, B) Simple shear, C) Simple extension. *[20]*

The second constant is the resistance to simple shearing stress t_s, G is the shear modulus and γ is the amount of shear defined as the ratio: the displacement over the height of the block. Which can be seen in Figure 8 B):

$$G = \frac{t_s}{\gamma} \tag{20}$$

Other common constants can be derived from the formulas 19 and 20. The tensile modulus E (Young´s modulus of elasticity), defined by simple tensile stress t over the corresponding fractional tensile elongation ε is given by:

$$E = \frac{t_s}{\varepsilon} = \frac{9KG}{3K+G} \tag{21}$$

Poisson’s ratio ν, defined by the ration lateral contraction strain ε_W over longitudinal tension strain ε_I which can be seen in Figure 8 C) is given by:

$$\nu = \frac{\varepsilon_W}{\varepsilon_I} = \frac{(1/2)(3K-2G)}{(3K+G)} \quad [20] \tag{22}$$

2.3.3. Steel springs

Steel springs just as rubber, exist in many different shapes and sizes, examples are shown in Figure 5. The most common design that comes to mind when speaking about springs is the helical spring but as can be seen, there is a lot more. The military started to interest in steel vibration dampers because of the rigid design that could be achieved. Therefore, has the wire-rope dampener been highly studied because of its versatility within the military. [21] [22]

Steel springs are mostly used to support heavy equipment but they can be tailored to suit all frequency ranges as the stiffness is easily calculated and altered to suit all loads hence the required resonant frequency. But still, steel springs are most suitable for the low frequency range of about 3-60 Hz (180-3600 rpm shaft speed). Steel springs have a natural frequency of about 1-6 Hz depending on the static deflection which limits the use of steel springs in some cases. Steel springs also transmit high frequency structure-borne noise, but this can be prevented with high frequency rubber pads or washers at both sides of the steel spring. Advantages with steel springs is that they are easily produced at the given stiffness and is not subject of the degradation from oxidation, oils etc. like elastomers. [21] [22]

2.3.4. Flexible mounting

Flexible mounting is when a vibration damper is used to reduce the transmission of vibrations from the source to the receiver. The damper is placed between the diesel engine and the foundation which it is mounted onto. Though flexible mounting can also be used to isolate receivers away from the diesel engine, like the electrical cabinet discussed in this thesis. [23]

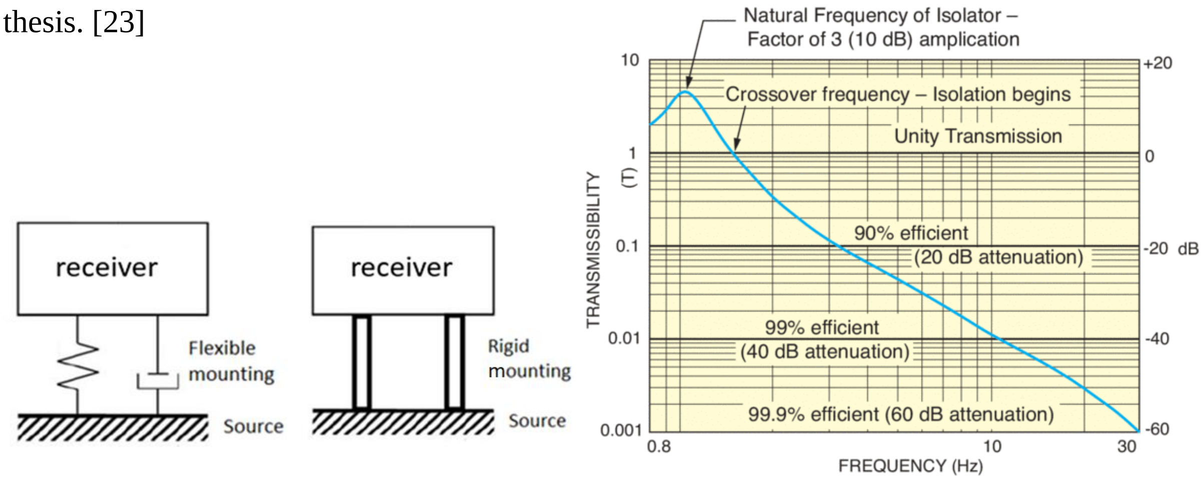

Figure 9. An example on the difference between flexible and rigid mounting and an example on how their transmissibility curve could look like. *[23]*

The goal with the flexible mounting is to isolate the structure away from the "unity transmission" (which is the case of the rigid mounting) though it needs to be enough far away from the natural frequency of the damper. The crossover frequency seen in the figure above starts at $\sqrt{2} * f_n$ or approximately 1.4 times the natural frequency of the damper. This is consistent to all dampers. Anything closer than that will create amplification to the system. The peak of the natural frequency is also determined by the damping value. The higher the damping value the lower the peak but it also does not insulate as good as one with a lower damping value. [23] [24]

Double flexible mounting

Double flexible mounting is when the engine is isolated from the foundation using a damper and simultaneous using a damper to isolate the electrical cabinet engine. The objective of this setup is to reduce transmission from the engine to the surrounding, while still needing to protect the electronics that are inside the electrical cabinet. Therefore, the double flexible system is used, though it has proven to come with many challenges.

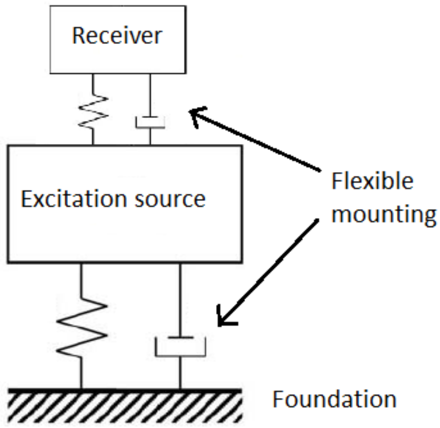

Figure 10. Example on a double flexible mounting system.

2.4. Degrees of freedom

One degree of freedom

All theory regarding a single degree of freedom (SDOF) is processed as mentioned in Chapter 2.2 and forward. In vibration analysis it is often idealized that structures and machines only have one free displacement coordinate, or *one degree of freedom*. Figure 2 and 3 are examples on where the model has been idealized to only have a single degree of freedom. A SDOF can also be rotational instead of transversal and is often indicated by the small angle $\emptyset(t)$ alternatively by a curved arrow. This is later shown in Figure 12. Topics in interest for elastic SDOF systems consists of: free and forced harmonic motions without or with damping, response varying loads or supporting motions, response spectra for dynamic loads and step by step calculations. [25]

Two degrees of freedom

It is possible to calculate more than a single degree of freedom; the following would be to discuss systems having several degrees of freedoms and the simplest to start with would be systems having two degrees of freedom system. The configuration is in other words completely defined by two coordinates or displacements. Figure 11 shows two masses $m_1$and m_2 that are connected with two springs, k_1 and k_2 respectively. Just as the SDOF the equation of motion for the system is found using Newton´s second law. The only different is that because of the two motions occurring at the same time, two differential equations are required to describe its motions. [26] In this case the homogeneous equations are:

$$m_1\ddot{u}_1 + (k_1+k_2)u_1 - k_2u_2 = 0 \quad \text{(23 a)}$$

$$m_2\ddot{u}_2 - k_2u_1 + k_2u_2 = 0 \quad \text{(23 b)}$$

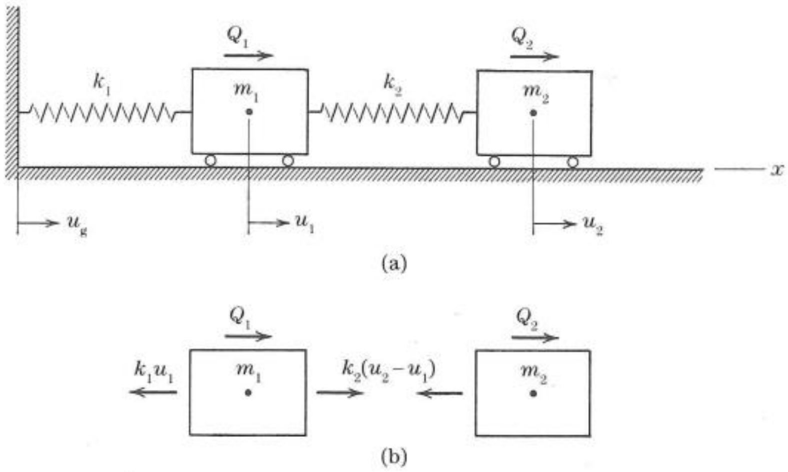

Figure 11. Example on a two degrees of freedom system. *[26]*

The two differential equations can be rewritten into a matrix form:

$$\begin{bmatrix} M_{11} & 0 \\ 0 & M_{22} \end{bmatrix} \begin{bmatrix} \ddot{u}_1 \\ \ddot{u}_2 \end{bmatrix} + \begin{bmatrix} S_{11} & S_{12} \\ S_{21} & S_{22} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad \text{(24)}$$

Where u is the displacement, $\ddot{u}$ is the acceleration and the coefficient matrices are:

$$\begin{bmatrix} M_{11} & 0 \\ 0 & M_{22} \end{bmatrix} = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \; \& \; \begin{bmatrix} S_{11} & S_{12} \\ S_{21} & S_{22} \end{bmatrix} = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \quad [27] \quad \text{(24.1 \& 24.2)}$$

Multiple degrees of freedom

After two degrees of freedom comes three, then four and so on. Within this category are all systems that have more than one degree of freedom but fever than an infinite number of degrees of freedom. The configuration of the system is then determined by a finite number of displacement coordinates. If there are *n* degrees of freedoms for different masses then it is required *n* differential equations to describe its motion. Because of the rising numbers to calculate, the terms grow easily very big. Matrixes are used to manipulate large numbers of terms, still it is very hard to analyse in the original coordinates, especially in damping cases. Therefore, the *normal-mode method of dynamic analysis* was developed. The method generalises the coordinates and the equation for undamped equation becomes uncoupled, meaning that each equation may be solved as a SDOF system. [28]

The general form for undamped systems with n degrees of freedom is:

$$\begin{bmatrix} M_{11} & M_{12} & M_{13} & \cdots & M_{1n} \\ M_{21} & M_{22} & M_{23} & \cdots & M_{2n} \\ M_{31} & M_{32} & M_{33} & \cdots & M_{3n} \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ M_{n1} & M_{n2} & M_{n3} & \cdots & M_{nn} \end{bmatrix} \begin{bmatrix} \ddot{u}_1 \\ \ddot{u}_2 \\ \ddot{u}_3 \\ \cdots \\ \ddot{u}_n \end{bmatrix} + \begin{bmatrix} S_{11} & S_{12} & S_{13} & \cdots & S_{1n} \\ S_{21} & S_{22} & S_{23} & \cdots & S_{2n} \\ S_{31} & S_{32} & S_{33} & \cdots & S_{3n} \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ S_{n1} & S_{n2} & S_{n3} & \cdots & S_{nn} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ \cdots \\ u_n \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \cdots \\ 0 \end{bmatrix} \quad [28] \quad (25)$$

2.4.1. Rigid body modes

As discussed previously, there are many natural frequencies within a structure. Rigid body modes mean that the structure moves as a rigid body and the structure is not subjected to any deformation. Six body modes exist in all structures which are mounted onto isolators. Three correspond to translation motion respectively three to rotational motion. [8]

The rigid body modes and their abbreviated forces are:

- *Longitudinal, X-translation,* F_L
- *Transversal, Y-translation,* F_T
- *Vertical, Z-translation,* F_V
- *Rolling, X-rotation,* M_R
- *Pitching, Y-rotation,* M_P
- *Yawing, Z-rotation,* M_Y

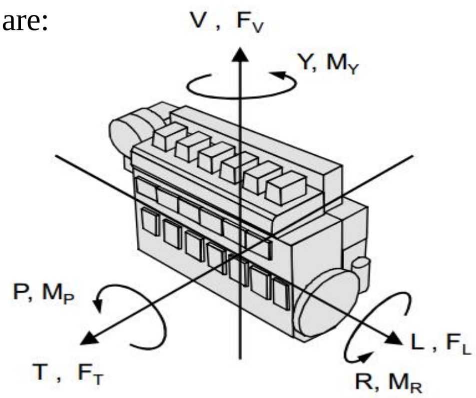

Figure 12. Rigid body mode directions and forces on a Wärtsilä engine. *[8]*

2.4.2. Elastic body modes

As the name reflects to, at elastic natural frequencies the points on the structure and its parts will move relative to each other, resulting in bending, torsion and on higher frequencies even more complex modes. Elastic body modes are occurring at a higher frequency than the rigid body modes, making them easier to separate from each other. The elastic body modes are also more dangerous to the system than rigid body modes because of the internal deformation. Elastic body modes neither have a specific number of modes, there are several hundreds of them in a single structure, but it is mostly the first modes that are of interest because of their higher amplitude. [8]

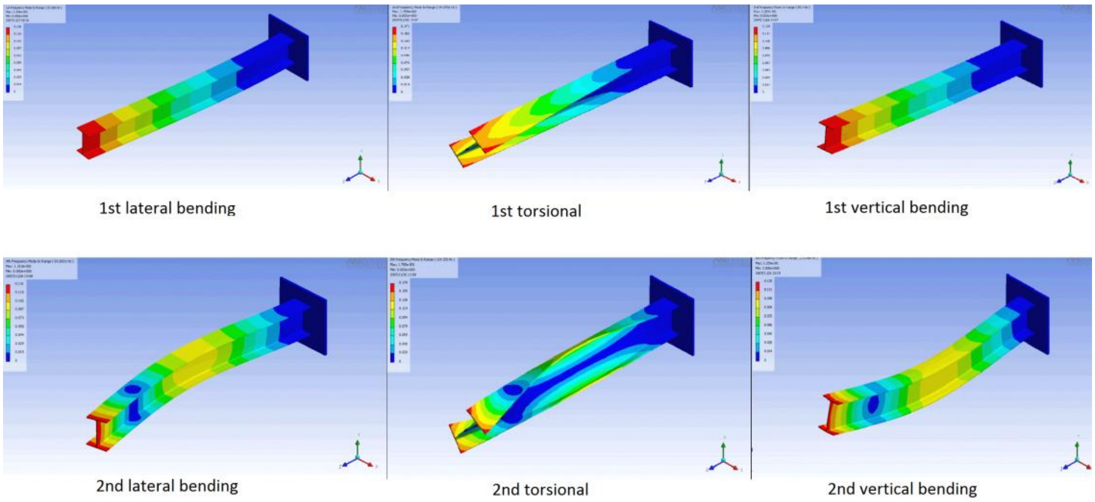

Figure 13. The first elastic mode shapes of a cantilevered I-beam. *[29]*

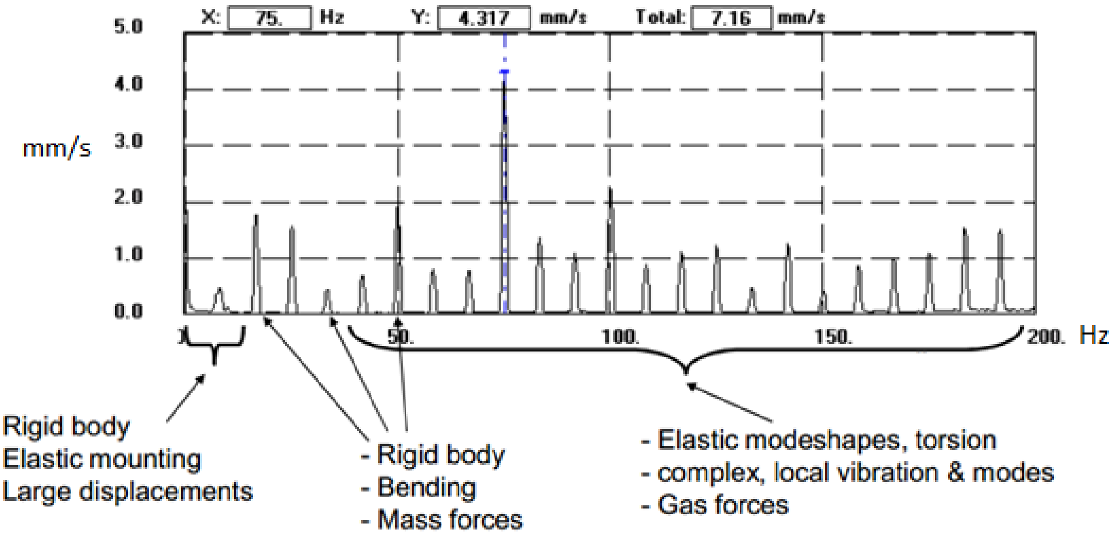

Figure 14. A typical vibration spectrum measured from a diesel engine. *[8]*

2.5. Vibration units

These are the most common units when measuring vibrations:

Displacement, which gives the results in mm, shows how much the structure is moving. This is measured on for example, engine start up and shutdowns to show movement. Displacement is relevant at low frequencies with an approximate range of 0-30 Hz due to mass forces and resilient mounting systems. [9]

Velocity, which is normally recommended to use, gives the result in mm/s. The typical frequency range is 10-300 Hz. This tells how much the structure moves over the given time and correspond to a given energy level of the system. This is also why it is recommended to use, because it gives the stress level of the measured component. [9]

Acceleration measured in m/s^2 tells how much force the object is moving with. This is recommended when measuring high frequency vibration components starting from approximately 100 Hz. Typically measured object are turbochargers, electronics or small mechanical components that are rigidly connected to the engine structure. [9]

The relationship between displacement (x), velocity (v) and acceleration (a) can be described by derivation and integration:

$$x(t) = X * \sin(\omega * t + \varphi) \tag{26}$$

$$v(t) = \frac{dx(t)}{dt} = \omega * X_P * \cos(\omega * t) = V_P * \cos(\omega * t) \tag{27}$$

$$a(t) = \frac{dv(t)}{dt} = \frac{d^2x(t)}{dt^2} = -\omega^2 * X_P * \sin(\omega * t) = -A_P * \sin(\omega * t) \tag{28}$$

Where ω is the angular frequency, t is the time and φ is the phase of the sinusoid. [9]

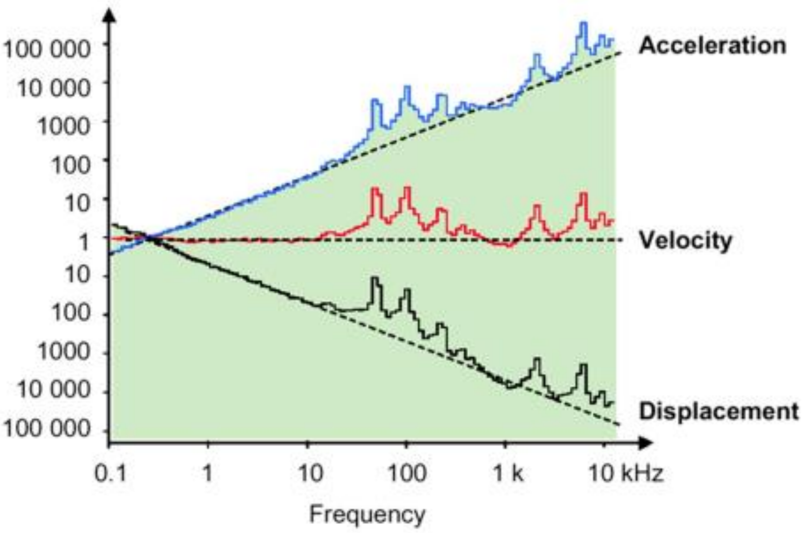

Figure 15. The relationship between acceleration, velocity and displacement. *[9]*

2.5.1. Time signal

When measuring vibrations, it is shown as a time history over the vibration response for displacement, velocity or acceleration. Normally, the time history gives a lot of useful information but it is very difficult to study. Therefore, peak-to-peak value is the most common value read from the time signal.

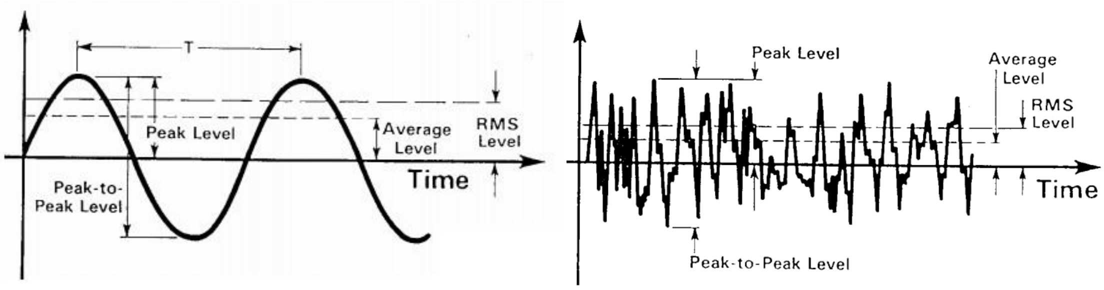

Figure 16. Example on a simplified (left) and typical (right) time signal. *[9]*

The vibration amplitude can be quantified in several ways:

The peak values indicate only the maximum levels without considering the time history of the spectrum.

The average level does not provide any useful information because it accounts for the time history but has no direct relationship to the physical quantities.

$$Average\ level = \frac{1}{T}\int_0^T |x|\, dt \quad (29)$$

The RMS (root mean square) level is the most relevant way of processing the time history. This is because it is directly related to the energy content over the time history, therefore also describing the potential destructive force of the vibrations.

$$RMS\ level = \sqrt{\frac{1}{T}\int_0^T X^2\,(t) dt} \qquad [9] \quad (30)$$

2.5.2. Spectrum

The information in the time signal can be processed with the Fourier transformation. FFT, which stands for Fast Fourier Transformation, is normally processed by computers. FFT is a more complex process than the DFT, which stands for Discrete Fourier Transformation. Both the FFT and DFT processes the time signal from the time domain to the frequency domain. Physically they will tell the frequency components of the given time signal and they are used to display the many frequencies a normal structure would have. [30]

Joseph Fourier proved that all 2π periodic functions can be "decomposed" into or made up from sinusoids of different frequencies using the formulas:

$$F(t) = a_0 + \sum_{n=1}^{\infty}[a_n * Cos(n * t) + b_n * sin(n * t)] \quad (31)$$

Where:

$$a_0 = \frac{2}{\pi}\int_{-\pi}^{\pi} f(t)dt \quad (31.1)$$

$$a_n = \frac{1}{\pi}\int_{-\pi}^{\pi} f(t)\cos(n * t)dt \quad (31.2)$$

$$b_n = \frac{1}{\pi}\int_{-\pi}^{\pi} f(t)\sin(n * t)dt \quad [30] \quad (31.3)$$

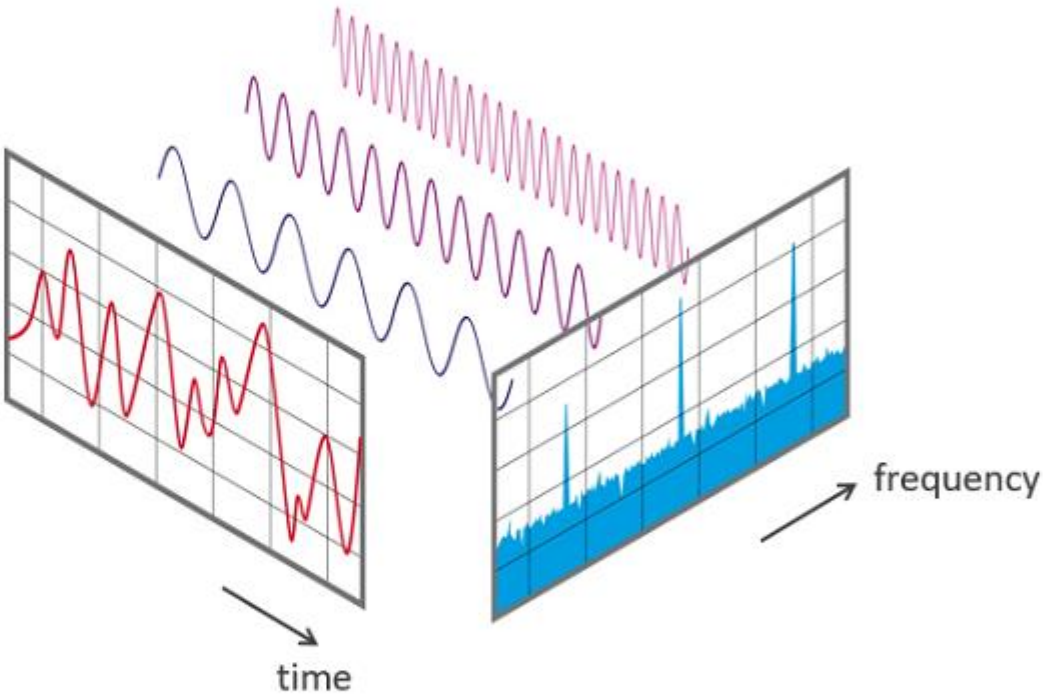

Figure 17. FFT visualised: The left graph shows the time domain, in the middle is the decomposed time signal and the right graph shows the frequency domain. *[31]*

Mathematically it is often simpler to manipulate the function in the frequency domain and when processed you can often determine just by looking at the FFT graph if there is anything wrong with the vibration levels in the structure. [30]

3. Measurements

3.1. Measurement equipment

This Chapter tells about the measurement equipment which the noise and vibration group uses in the Wärtsilä factory in Vaasa. There are a lot more equipment and accessories used for measuring vibrations in Wärtsilä but I have only chosen to include the instruments and equipment which are relevant for this thesis.

3.1.1. Accelerometers

Figure 18. Typical accelerometers and equipment. The pen is used as a scale.

Picture 18 shows typical accelerometers and equipment used at Wärtsilä by the noise and vibration group at the factory in Vaasa. The top row from left to right: rubber filter used to reduce electrical disturbances, fastening plate with bolts on both sides and to the right a ceramic filter used with high temperatures. Middle row shows different magnets. Bottom row from left to right: A tri-axial accelerometer with a frequency range up to 50 g, tri-axial accelerometer with a frequency range up to 500 g and to the right a single direction accelerometer which is normally used as a reference and capable of measurements up to the frequency of 50 g. Except for the equipment to the accelerometers on the picture there are also glue plates with or without bolt, that are often used when measuring on for example aluminium where magnets do not stick.

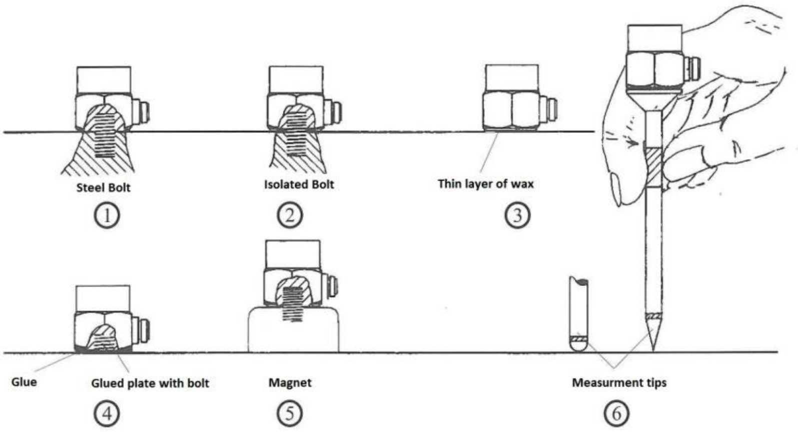

Figure 19. Different attachment methods for accelerometers. *[32]*

The method used to attach the accelerometer to the object have great influence on high frequency measurements. The accelerometer can never be assumed to be rigidly attached to the object, which means that the accelerometer creates a new spring and mass system, which have an internal resonance that sets the limit for how high frequencies it is possible to measure with the equipment. To be able to measure high frequency measurements the first 4 options is suitable. Even though the wax (option 3) can measure surprisingly high frequencies, the wax itself limits the measurements to a maximum of 40-50 °C. The magnet (option 5) can measure up to the frequency of 2 kHz and the handheld measurement tip (option 6) is suitable up to 500-1000 Hz. [32]

3.1.2. Benstone

The Benstone Impaq is a 4-channel handheld device weighing only 1.15 kg that is used by the vibration group in Vaasa. The Benstone is capable to measure a frequency range up to 40 kHz with 12800 lines of resolution and can have the functions: FFT spectral analysis, rotor dynamic balancing, vibration meter, 1/3 octave analysis, order tracking, data collection, data recorder and bearing analysis, making it a very efficient tool for fast measurements. The only drawback with the handheld analyser is that the measurements is not possible to analyse during the measurements.

Figure 20. The 4-channel handheld Benstone analyser. *[33]*

3.1.3. OROS

The OROS 3-Series instruments are multichannel FFT analysers, dynamic signal analysers, FFT spectrum analysers and portable vibration analysers. They are designed for the field, packed with advanced laboratory instruments. The analysers are normally 2-32 channels but can be cascaded and support up to 1,000 channels. They support real time analysis with the ability to post analyse the recorded data and offer a great ability to be customized to the needed situation. The equipment seen on Figure 21 is used by the vibration group in Vaasa.

Figure 21. A 32 channel Oros 38 (Left) and a 10 Channel Oros 35 (Right). *[34]*

3.2. Measurements

There are several measurements needed to fully understand a vibration problem. The following subchapter presents which measurements that was carried out to investigate the flexible mounting.

FFT spectrum is the normal spectrum that is the first one to be checked for faults within a structure. To perform the measurement, it only requires the engine in question to be running. Though it is almost always perfomed on the engine running at nominal speed and 100 % load. About three to five avarages creates the result file.

Engine sweeps are performed to search for the structures natural frequencies. Two sweeps are often performed, one at 25 % load from 420 to 600 rpm and one sweep at 75 % load from 600 to 780 rpm. The sweeps normally take 5-10 minutes depending on how slow the ramp is. The slower the ramp, the clearer the result. The measurement is active during the whole ramp and the settings are set to hold the highest peak within every FFT spectra and overlap them by 75 %, keeping the highest peaks that resembles the peaks of the natural frequencies.

Engine stopping is pretty much self-explanatory but this measurement is carried out to make sure we pass by every natural frequency. This is because we measure from max to zero rpm in a single sweep. There is no load on the engine during this measurement.

ODS measurements which stands for operating deflection shape is a modal measurement method and used as a tool to visualise how a structure is moving. A reference is used when measuring the structure and with the information it is then possible to produce a moving system which gives us its relative moments against the reference. This is often used as a helpful tool to determine body modes. Another modal measurement method called **Frequency Response Function** or FRF is used to determine natural frequencies of a structure. In this case the force going into the structure is measured, for example, by putting a sensor on a hammer and measuring the response in the structure when it gets hit by the hammer. Alternatively, by having a hydraulic shaker with a force sensor sending out random excitations within a given spectrum and then measuring the input to get the response of the structure.

Hydraulic shaker. Instead of having an engine running to excite the structure, we have a hydraulic shaker which we fasten to the engine. This allows for much more control while shaking the structure. This helps to isolate the structure from any disturbances from the engine itself, meaning the structure itself can be more thoroughly examined. Also measuring things like sweeps on a hydraulic shaker allows for the same amplitude to be used over a set frequency area. This allows for a more accurate result than a normal engine sweep.

Figure 22. Hydraulic shaker attached to engine.

Electromagnetic shaker table test was measured on an electromagnetic shaker in Technobothnia, Vaasa. A test rig was created which resembles the main cabinet in dimensions. The rig allows for different mounting setups to be used including the ability to place different side stoppers and different weights inside the box.

Figure 23. Shaker table in vertical position.

Figure 24. Shaker table in longitudinal position.

Measuring on a shaker table also has the benefit of isolating the components you like to investigate, this is because it removes the natural frequencies of the engine itself. Leaving us with the clear natural frequencies of the structure which we excite. Figures 23 and 24 shows the electromagnetic shaker in Technobothnia and its two configurations that was used.

3.3. Main object of investigation

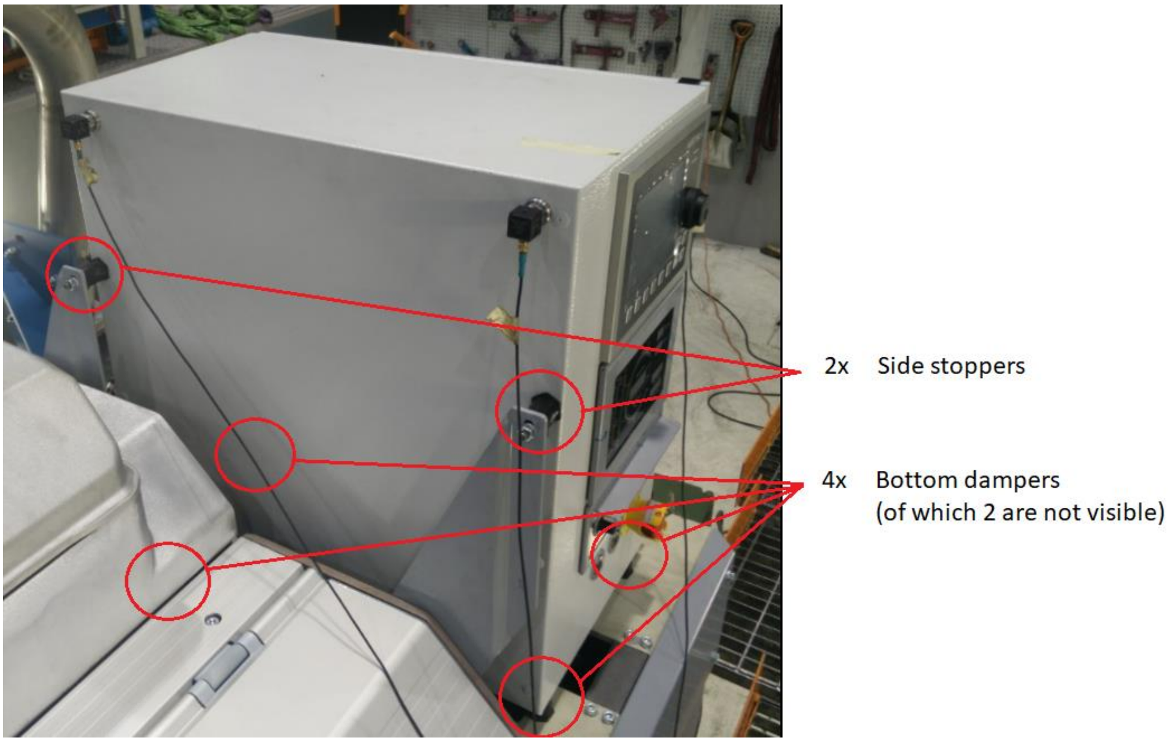

Figure 25. The main electrical cabinet on a Wärtsilä 6L34 engine.

Figure 25 shows the main electrical cabinet, which uses flexible mounting. This main cabinet has several different dimensions and is mounted both standing and laying on different setups. Therefore, the project could be made huge if all were to be investigated, instead it is focused on studying the double flexible mounting and its behaviour. The main cabinet also uses a variety of side stoppers, the two brackets seen in Figure 25 with two longitudinal stoppers is just one of them.

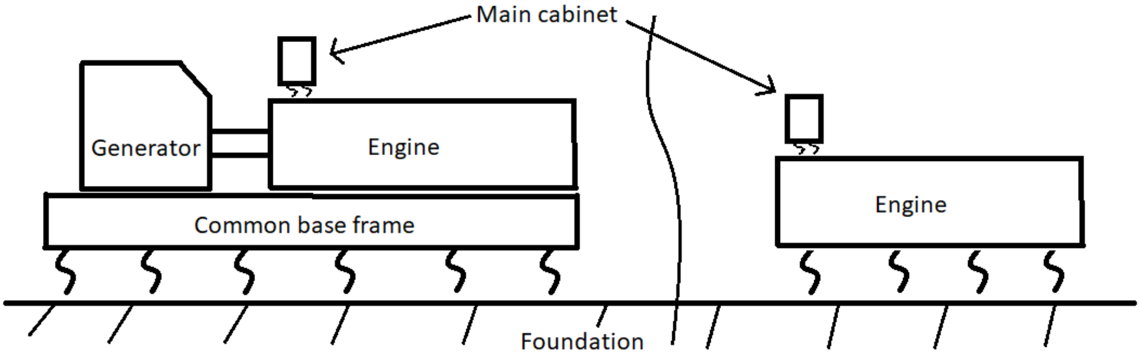

Figure 26. Example on an engine mounted as a Genset (left) and as a main engine (right).

Figure 26 shows the two main setups. The one to right which shows the engine mounted as a main engine has proven to cause some vibration problems on the main cabinet.

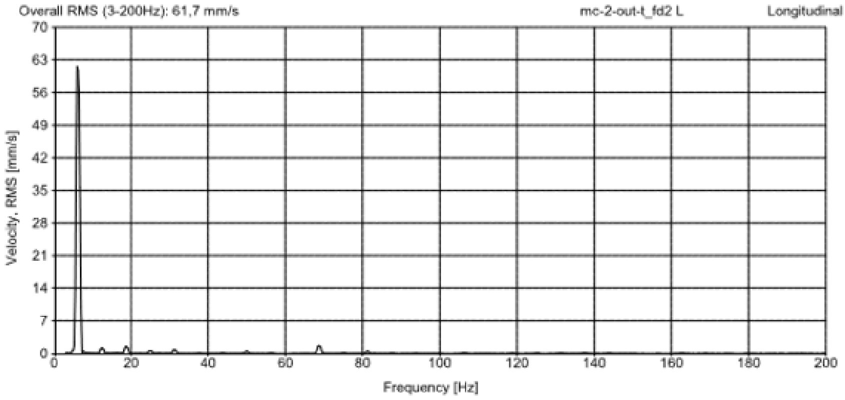

Figure 27. Example on a FFT spectrum from a main engine running at 100 % load.

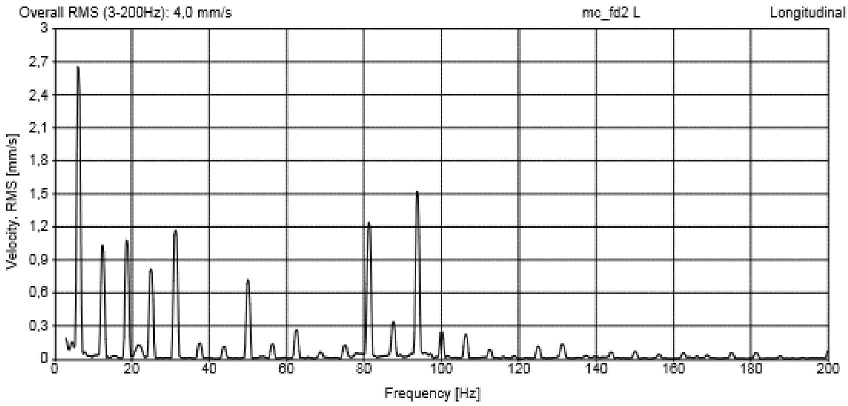

Figure 28. Example on a FFT spectrum from a genset engine running at 100 % load.

The graph in Figure 27 shows a clear resonance at the engines first half order, or around 6 Hz. The RMS level is at 62mm/s, though in Figure 28, which shows the same spectra on a genset engine, has a maximum level of about 2.7mm/s.

This phenomenon has been noticed several times on different engines and even on engines which have been tested both as a main engine and as a genset. This means that the problem must be influenced by the setup, not the engine and thus creating a need to investigate the double flexible mounting used for the main engine.

Also during one of the measurements a hypothesis about the side brackets being too soft came to mind. This is because the brackets that can be seen in Figure 25 were soft enough that they could be bent just by swinging the cabinet back and forth. It was decided that four brackets were to be welded into two much stiffer brackets and these brackets were also to be measured. In the test results these brackets will be named as “Reinforced brackets”.

4. Method

The method of investigation for the flexible mounting is purely based on measurements. The measurements are based on the expertise and know-how of the vibration team.

FFT spectra using the handheld vibration analyser Benstone was collected for several main cabinets that was going to be shipped as a main engine. Though there are not many main engines shipped per year, so often when an engine that was going to be shipped as a main engine was in production or in the test cell, several tests were performed.

After noticing high peaks on some engines, sweeps and stoppings were carried out on some engines. Though these tests did not give the needed information and further investigation was needed.

A hydraulic shaker test was done on an engine that was going to be shipped as a main engine and FRF spectra were collected for the engine block, brackets and the main cabinet itself and all measurement points were measured in three directions. The hydraulic shaker attached to the engine can be seen in Figure 22, and one of the measurements can be seen on Figure 25. This was done to be able to create a moving image on how the structure moves at different frequencies. The FRF spectra was used to search for the different natural frequencies within the structures.

It was during the hydraulic shaker test the brackets discussed in Chapter 3.3 was thought about. Later when the engine entered the test cell, sweeps and ODS measurements with original and reinforced brackets were measured.

An electromagnetic shaker table test was done in Technobothnia, Vaasa. A test rig as mentioned was created and all setups measured can be found in Appendix A. During this measurement accelerometers were placed followingly. They were placed in the upper corner one side and diagonally on the bottom corner on the opposite on the box and diagonally across on the frame. The reference was located on the middle of the shaker table in the excitation direction.

On the shaker table, a single flexible mounted system with only rubber elements was tested first. This to have some reference to the other test and to check for the natural frequencies of the rubber, which also is the reference as the natural frequency for the box. Secondary a double flexible mounting system was tested. Steel springs were attached between the hydraulic shaker and the frame. A mass of 60 kg was added to the frame in-between the shaker and the box and rubber elements were mounted between the frame and the box. Afterwards a second double flexible mounting system was tested. The rubber elements were now placed between the shaker and the frame, and the steel dampers were placed between the frame and the box. Finally, the double flex special test was thought about during the measurements and resulted in some last-minute welding. The test is to check how much of a difference it would make if the side stoppers would be directly from the shaker instead of the frame. This to investigate if it would help to build a support directly from the foundation to the main cabinet instead of trying to support it from the engine.

Figure 29. Example on one of the double flexible setups.

To create a baseline and to easier determine changes, the same sweep settings was used on all sweeps done in Technobothnia. After some initial tests it became clear that the sweeps all need to be between 20-2 Hz and the level of output to the shaker table should be 20 mm/s. Meaning that we started the shaker table at 20 Hz and went down to 2 Hz. The reason why we went from 20 to 2 Hz was because of the natural frequency of the hydraulic shaker table, which was at about 2.5 Hz and created some difficulties in getting the shaker not to shut down because of its safety regulations. A slight peak can also be seen in some graphs in Appendix F at 2.5 Hz due to the resonance of the shaker. A reference single axial sensor was used to check that the levels of 20 mm/s was consistent during all sweeps.

5. Results

Firstly, the single flexible mounting results will be presented to get a reference and to understand some key notes on the flexible system.

The Hydraulic shaker table measurements were as stated, done with the same sweep settings and all the setups measured can be found in Appendix A.

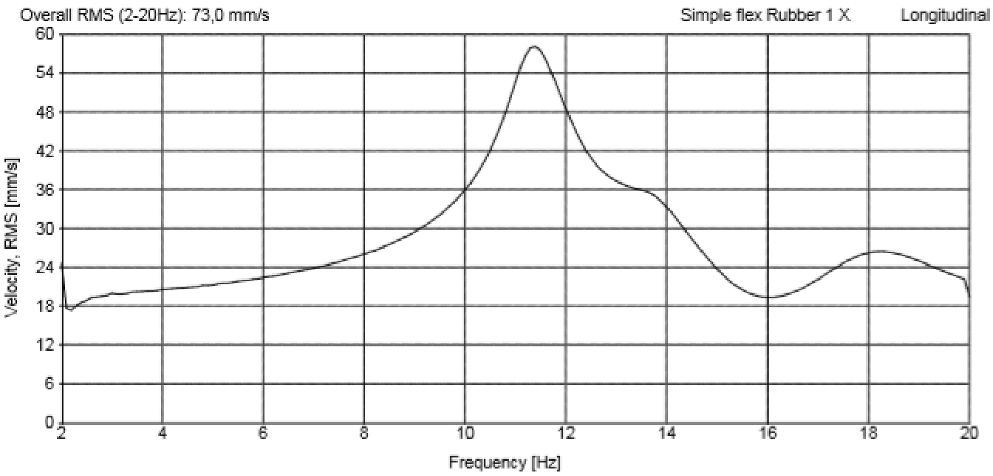

Figure 30. FFT spectrum from the shaker table test where both side stoppers were used.

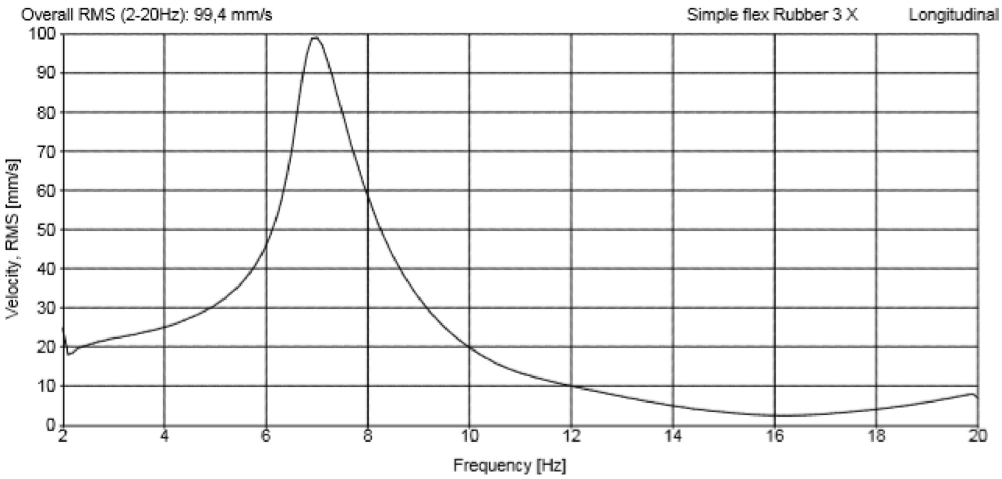

Figure 31. FFT spectrum from the shaker table test where no side stoppers were used.

Figure 30 has a peak level of 58.1mm/s at 11.4 Hz which uses both side stoppers while Figure 31 has a peak level of 99.1mm/s at 7.0 Hz. This is a total drop of 41mm/s and about 40 % of the total level when both side stoppers were used. These tests were identical except for the difference in side stoppers and this clearly shows the importance of the side stopper, it also shows the clear influence the stiffness of the added dampers has on the increase of the natural frequency.

The great influence of the side stoppers is further confirmed from the results in Appendix D, which shows the test of the reinforced bracket. The vibration levels did not change much, but the two highest peaks both decreased by about 20 %. Therefore, Appendix D shows that it is important that the side brackets for the side stoppers must be stiff.

The double flexible system was tested with the same setup used in Figure 30 but with the added frame and weight, creating the double flexible system.

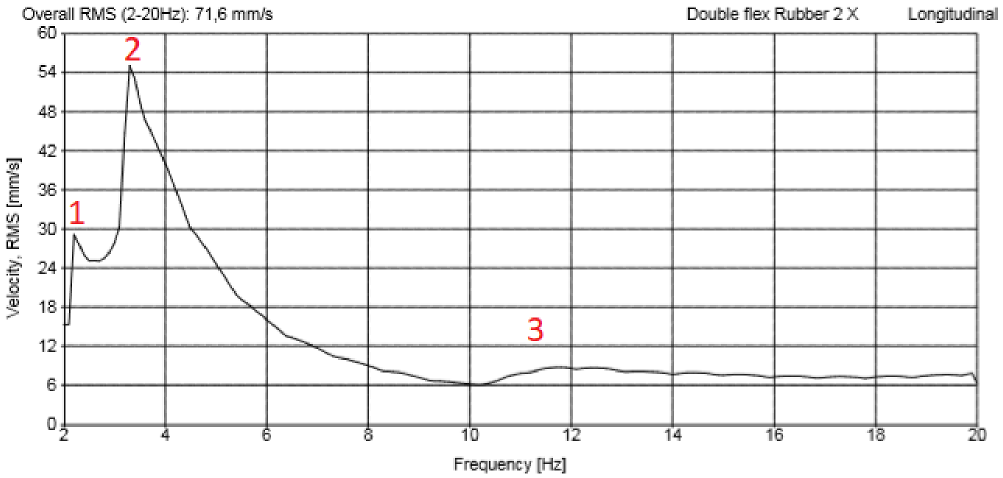

Figure 32. ***FFT spectrum results from the box in the shaker table test.***

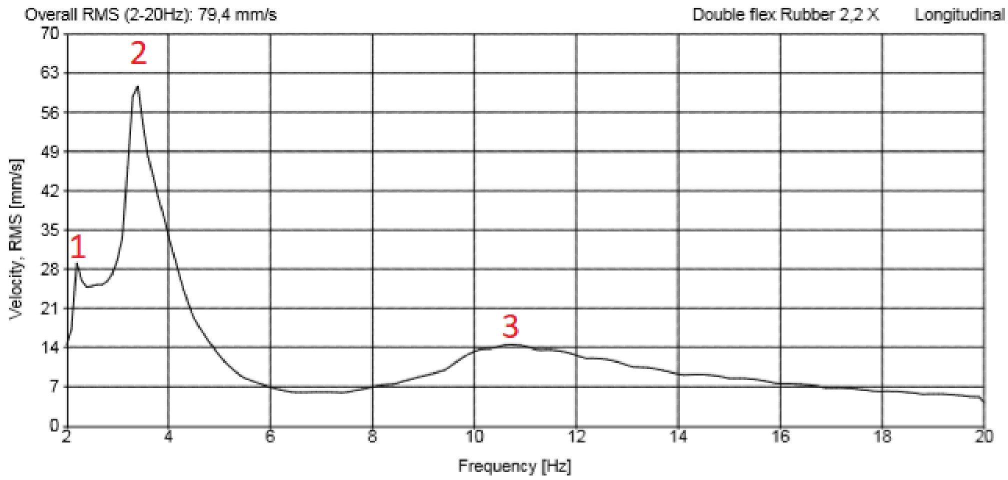

Figure 33. ***FFT spectrum results from the frame in the shaker table test.***

As mentioned in Chapter 4, the peak which is labelled as number 1 in Figure 32 and 33 is from the natural frequency of the shaker table and is not to be considered in any of the graphs.

Now if we recall from the single flexible system, the natural frequency of the box was 11.4 Hz. This is however not showing on the box in the measurement in the double flexible system, but it shows on the frame of the double flexible system as label 3. This means all energy transferred to the box is from the frame and the frame´s natural frequency which is labelled as number 2. Comparing the Figures 32 and 33 also shows that the levels in the frame are a little lower but otherwise identical.

Appendix E shows the result for all three directions for the hydraulic shaker table measurements. The results in the table has been simplified by presenting them as an excel table showing its highest peaks instead of the 40 pdf pages. Though every peak in Appendix E corresponds to a pdf page just as Appendix B, but some of the graphs are presented in Appendix F to show the double flexible phenomenon.

Table 1. Simplified results from the hydraulic shaker table. The left column shows the results from the longitudinal position (x-direction) and the right shows the results from the vertical position (z-direction). The number, X tells that the measurement was done with sensors on the box, while the number, X.2 tells that the measurement was done with sensors on the frame.

X-Direction	Frequency [Hz]	Amplitude [mm/s]
Double flex Steel-Rubber 2	3.3	55.0
Double flex Steel-Rubber 2.2	3.4	60.6
Double flex Steel-Rubber 4	3.3	75.8
Double flex Steel-Rubber 4.2	3.3	62.1
Double flex Steel-Rubber 5	3.2	67.5
Double flex Steel-Rubber 5.2	3.2	53.6

Z-Direction	Frequency [Hz]	Amplitude [mm/s]
Vertical double flex Steel-Rubber 1	7.1	62.1
Vertical double flex Steel-Rubber 1.2	7.0	61.3
Vertical double flex Steel-Rubber 2	7.2	64.9
Vertical double flex Steel-Rubber 2.2	7.0	59.8
Vertical double flex Steel-Rubber 3	7.7	62.3
Vertical double flex Steel-Rubber 3.2	7.0	54.4

X-Direction	Frequency [Hz]	Amplitude [mm/s]
Double flex Rubber-Steel 1	3.8	88.3
Double flex Rubber-Steel 1.2	3.8	66.7
Double flex Rubber-Steel 2	3.8	92.8
Double flex Rubber-Steel 2.2	3.7	68.3
Double flex Rubber-Steel 3	3.8	89.8
Double flex Rubber-Steel 3.2	3.7	62.6

Z-Direction	Frequency [Hz]	Amplitude [mm/s]
Vertical double flex Rubber-Steel 1	9.7	133.6
Vertical double flex Rubber-Steel 1.2	9.6	80.8
Vertical double flex Rubber-Steel 2	9.8	134.1
Vertical double flex Rubber-Steel 2.2	9.7	80.0
Vertical double flex Rubber-Steel 3	9.9	121.4
Vertical double flex Rubber-Steel 3.2	9.7	74.8

X-Direction	Frequency [Hz]	Amplitude [mm/s]
Double flex special 1	5.9	60.4
Double flex special 1.2	6.0	84.4

Z-Direction	Frequency [Hz]	Amplitude [mm/s]
Vertical double flex special 1	7.4	70.1
Vertical double flex special 1.2	7.3	80.8

All setups in table 1 contain the same weight. Steel-Rubber means that the bottom dampers are made of steel and the top dampers are made of rubber, while Rubber-Steel means that the bottom dampers are made of rubber and the top dampers are made of steel. Flex special is supported directly from the shaker table to the box.

The numbers correspond to a specific setup which can be found in Appendix A, but they go in sets of three from the top to the bottom in table 1. The top uses both stoppers, the one in the middle uses one stopper in longitudinal direction and the bottom one is without stopper.

The two columns, frequency and amplitude tells the frequency and the highest peak in the measured FFT spectrum, which is the natural frequency.

Overall does the Steel-Rubber maintain its amplitude quite well and has the lowest peaks both in longitudinal and vertical direction, while the Rubber-Steel setup gets amplification in both longitudinal and vertical direction.

As can be seen on the different setups, we will have the same frequency giving the highest amplitude on both the frame and the box (± 0.1 Hz). The amplitude and dominant frequency differs depending on which setup was used, this is because of the different natural frequencies in the dampers. In Appendix F can it be clearly seen that both double flex setups behave similar, the spectrum for the 16 graphs shows that the box and the frame moves almost the same in the setups: steel-rubber 1-5 and rubber steel 1-3. This trend is consistent in all directions on all double flex measurements.

The only result that differs from the others is the double flex special measurement, comparing it to the double flex steel-rubber 4 and 4.2 which both only use longitudinal stoppers. The special flex raises the natural frequency and lowers the amplitude on the box.

The phenomenon of the double flexible system was also noticed in a hydraulic shaker test that was done on a main engine.

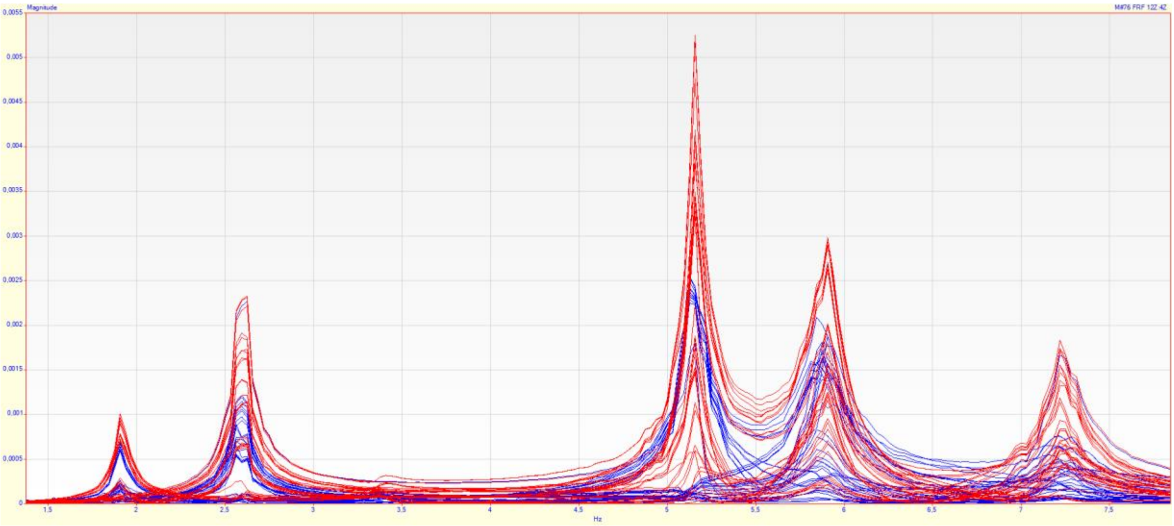

Figure 34. Overlapping FRF spectrum, engine block is presented as blue and cabinet as red.

Figure 34 shows the result from a hydraulic shaker test like the one in Figure 22. The figure shows the overlapping of the two first Figures in Appendix C. The blue represents the engine block of the engine and the red represents the main cabinet and its brackets. I chose to represent the two structures in different colours to show how close their rigid body motions are. This graph clearly shows that the problem is originating from the engine itself and the flexible mounting. All peaks in the system originates from when the engine is in resonance, and we also get amplification on the main cabinet.

6. Conclusion

As can be seen from the results, the two systems almost move as a single unit when we have a double flexible mounting. In this case when the engine or the frame in the hydraulic shaker test starts moving, the rest follows with it. This is going to happen on every engine mounted as a main engine regardless weight, stiffness or what kind of dampers are used. This is because the phenomenon of the double flex that will most probably occur in every double flexible system.

The side stoppers indicate to have an important role if the current setup is to still be used and for the dampers to work properly they need stiff brackets, otherwise the bracket just bends and may even create higher vibration levels.

The lowest amplitude for the setups in longitudinal direction is when we use all side stoppers. This is because they do only help in the direction that they are mounted, meaning that a cabinet should have dampeners mounted in three different directions for optimal results. Though adding side stoppers also increases transmission in other directions. As more side stoppers were added the levels also increased in for example vertical direction, though on the engine it was the longitudinal direction that caused the most problems so the vertical vibration levels should not be a problem.

As the choice of dampers, I would recommend the rubber elements which have lower overall levels and better isolation.

The double flex special test also indicates that the structures behaviour can be changed if we support the structure from the foundation, but this is not as easy implicated on an engine as it is on a test rig.

Overall, the double flexible mounting is a complex problem which needs more investigation. If lower levels are desired then other methods than changing bracket, dampers or weight must be investigated and implicated.

7. Recommended future work

As recommended future work for the flexible mounting I would suggest further investigating the double flexible mounting using other methods, this because it is outside the limitations for this work but I have come across it several times during this thesis. This is normally used by mounting 8 wire rope dampers, four on top and four on the bottom suspending the object floating mid-air or with a mass damper mounted on top of the main cabinet. The military uses the 8 damper setups and the mass damper has already been tested in the form of a “ReKi” (Resonance killer) within Wärtsilä, but an extensive test for the main cabinet is recommended. This uses the phenomena of anti-resonance, meaning that the box would stay still even though the engine is moving. The problem with this is that the setups are so different from each other and while the anti-resonance can be calculated, it will be different for every engine, and engines using different rpms will create even more challenges.

8. Discussion

The steel brackets used in the shaker table test is in my opinion way too stiff to be compared to the rubber elements in the same test, at least when comparing peak levels. This is because I based the wire rope elements on the W31 setup for the main cabinet, which only uses 2 bottom dampers and 4 side stoppers. These wire ropes were designed for a load of approximately 30 kg each while the rubbers were designed for approximately 20 kg each. This is too stiff, because it is recommended to load the dampers between 70-100 % for best isolation and damping. Though because it almost took 4 months to receive these dampers from Italy it was not worth waiting for new dampers instead of using dampers that were a little too stiff. Therefore, it is a possibility that the peak levels in the result for the steel springs would have been lower using better optimised dampers. Though they do not influence the results in investigating the movement of the system, just its peak level.

The reliability of the result could have been made better using FRF graphs instead of FFT´s because of the possibility of variations in the sweep from the electromagnetic shaker. But the reference results are saved and possible to investigate for every sweep if it should be necessary. There was no need to add 40 more graphs in the data files just to prove the consistence of the electromagnetic shaker table.

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Appendix A

Simple Flexible mounted – Rubber elements on box
Settings: 20mm/s   20-2Hz Sweep
Longitudinal

Simple flex, Rubber, both stopper

Simple flex Rubber 1	Box + 22,8kg	Sensors on box

Simple flex, Rubber, one stopper (longitudinal)

Simple flex Rubber 2	Box + 22,8kg	Sensors on box

Simple flex, Rubber, No stopper

Simple flex Rubber 3	Box + 22,8kg	Sensors on box

Simple flex, Rubber, No stopper + Mass (40,5kg)

Simple flex Rubber 4	Box + 58,8kg	Sensors on box

Simple Flexible mounted – Rubber elements on box
Settings: 20mm/s   20-2Hz Sweep
Vertical

Simple flex, Rubber, both stopper

Vertical Simple flex Rubber 1	Box + 22,8kg	Sensors on box

Simple flex, Rubber, one stopper (longitudinal)

Vertical Simple flex Rubber 2	Box + 22,8kg	Sensors on box

Simple flex, Rubber, No stopper

Vertical Simple flex Rubber 3	Box + 22,8kg	Sensors on box

Simple flex, Rubber, No stopper + Mass (40,5kg)

Vertical Simple flex Rubber 4	Box + 58,8kg	Sensors on box

Double Flexible mounted – Steel elements on frame, Rubber elements on box
Settings: 20mm/s   20-2Hz Sweep (Added weight in frame 60 kg)
Longitudinal

Double flex, Steel-Rubber, both stopper

Double flex Steel-Rubber 1	Box + 40kg	Sensors on box
Double flex Steel-Rubber 1.2	Box + 40kg	Sensors on frame
Double flex Steel-Rubber 2	Box + 20kg	Sensors on box
Double flex Steel-Rubber 2.2	Box + 20kg	Sensors on frame
Double flex Steel-Rubber 3	Box	Sensors on box
Double flex Steel-Rubber 3.2	Box	Sensors on frame

Double flex, Steel-Rubber, one stopper (longitudinal)

Double flex Steel-Rubber 4	Box + 20kg	Sensors on box
Double flex Steel-Rubber 4.2	Box + 20kg	Sensors on frame

Double flex, Steel-Rubber, No stopper

Double flex Steel-Rubber 5	Box + 20kg	Sensors on box
Double flex Steel-Rubber 5.2	Box + 20kg	Sensors on frame

Double Flexible mounted – Steel elements on frame, Rubber elements on box
Settings: 20mm/s   20-2Hz Sweep (Added weight in frame 60 kg)
Vertical

Double flex, Steel-Rubber, both stopper

Double flex Steel-Rubber 1	Box + 20kg	Sensors on box
Double flex Steel-Rubber 1.2	Box + 20kg	Sensors on frame

Double flex, Steel-Rubber, one stopper (longitudinal)

Double flex Steel-Rubber 2	Box + 20kg	Sensors on box
Double flex Steel-Rubber 2.2	Box + 20kg	Sensors on frame

Double flex, Steel-Rubber, No stopper

Double flex Steel-Rubber 3	Box + 20kg	Sensors on box
Double flex Steel-Rubber 3.2	Box + 20kg	Sensors on frame

Double Flexible mounted – Rubber elements on frame, Steel elements on box
Settings: 20mm/s   20-2Hz Sweep (Added weight in frame 60 kg)
Longitudinal

Double flex, Rubber-Steel, both stopper

Double flex Rubber-Steel 1	Box + 22,8kg	Sensors on box
Double flex Rubber-Steel 1.2	Box + 22,8kg	Sensors on frame

Double flex, Rubber-Steel, one stopper (longitudinal)

Double flex Rubber-Steel 2	Box + 22,8kg	Sensors on box
Double flex Switch 2.2	Box + 22,8kg	Sensors on frame

Double flex, Rubber-Steel, No stopper

Double flex Switch 3	Box + 22,8kg	Sensors on box
Double flex Switch 3.2	Box + 22,8kg	Sensors on frame

Double Flexible mounted – Rubber elements on frame, Steel elements on box
Settings: 20mm/s   20-2Hz Sweep (Added weight in frame 60 kg)
Vertical

Double flex, Rubber-Steel, both stopper

Double flex Rubber-Steel 1	Box + 22,8kg	Sensors on box
Double flex Rubber-Steel 1.2	Box + 22,8kg	Sensors on frame

Double flex, Rubber-Steel, one stopper (longitudinal)

Double flex Rubber-Steel 2	Box + 22,8kg	Sensors on box
Double flex Rubber-Steel 2.2	Box + 22,8kg	Sensors on frame

Double flex, switched, No stopper

Double flex Rubber-Steel 3	Box + 22,8kg	Sensors on box
Double flex Rubber-Steel 3.2	Box + 22,8kg	Sensors on frame

Special Flexible mounted – steel elements on frame, Rubber elements on box
Settings: 20mm/s 20-2Hz Sweep (Added weight in frame 60 kg)
Longitudinal

Double flex, Rubber, one stopper (between shaker and box)

Double flex Special 1.	Box + 20kg	Sensors on box
Double flex Special 1.2	Box + 20kg	Sensors on frame

Special Flexible mounted – steel elements on frame, Rubber elements on box
Settings: 20mm/s 20-2Hz Sweep (Added weight in frame 60 kg)
Vertical

Double flex, Rubber, one stopper (between shaker and box)

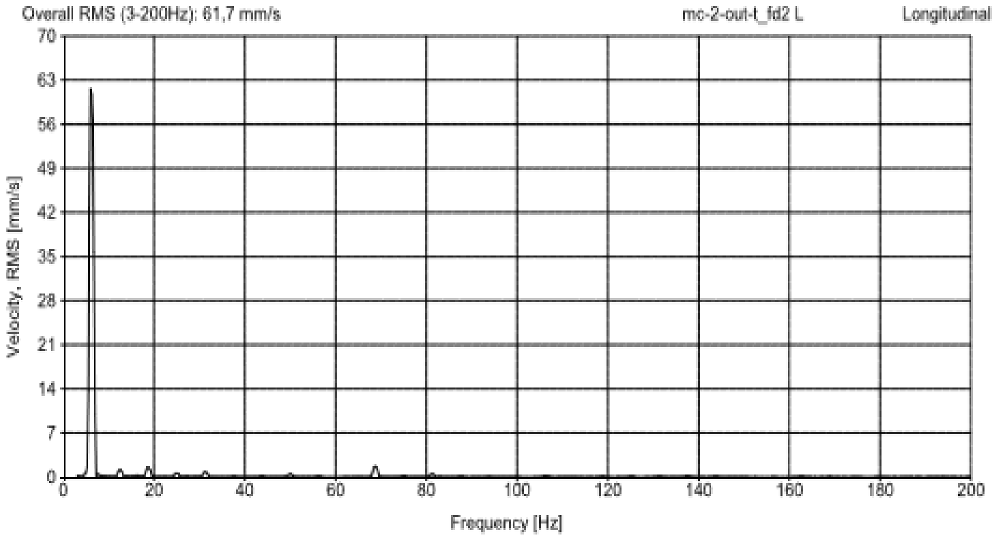
Double flex Special 1	Box + 20kg	Sensors on box
Double flex Special 1.2	Box + 20kg	Sensors on frame

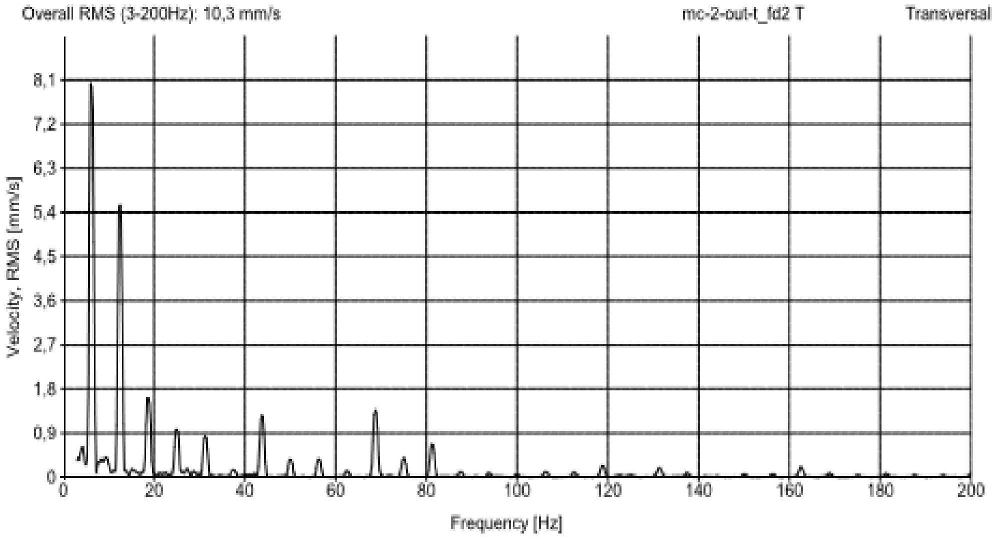
Appendix B

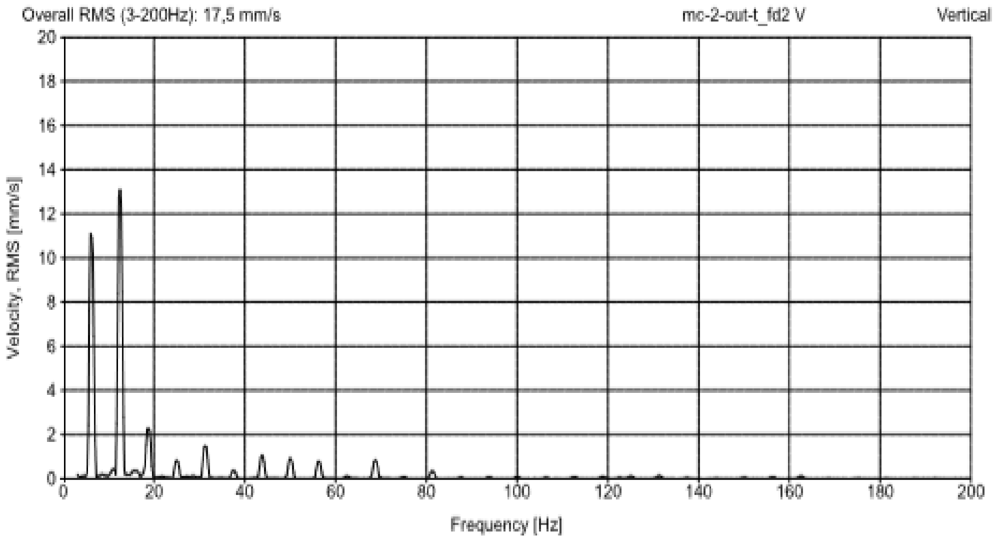
Installation: _ _ _ _
Engine type: W8L34DF
Engine no.:

VIBRATION MEASUREMENT
P = 4000 kW / n = 750 rpm

Point MC_2_out_top



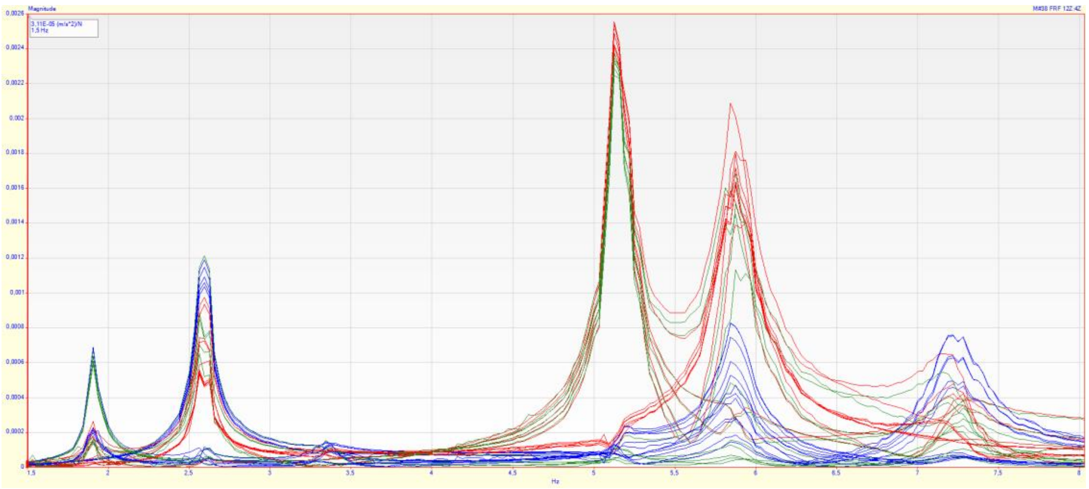




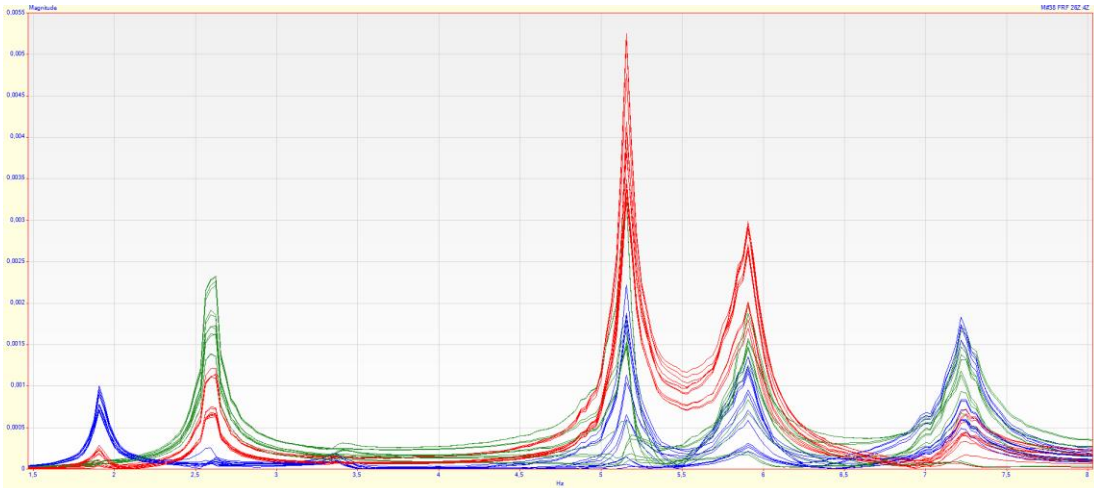
Measured by A Lassfolk 06.06.2017

Printed by AnVib v3.14 / 06.06.2017 ALA066

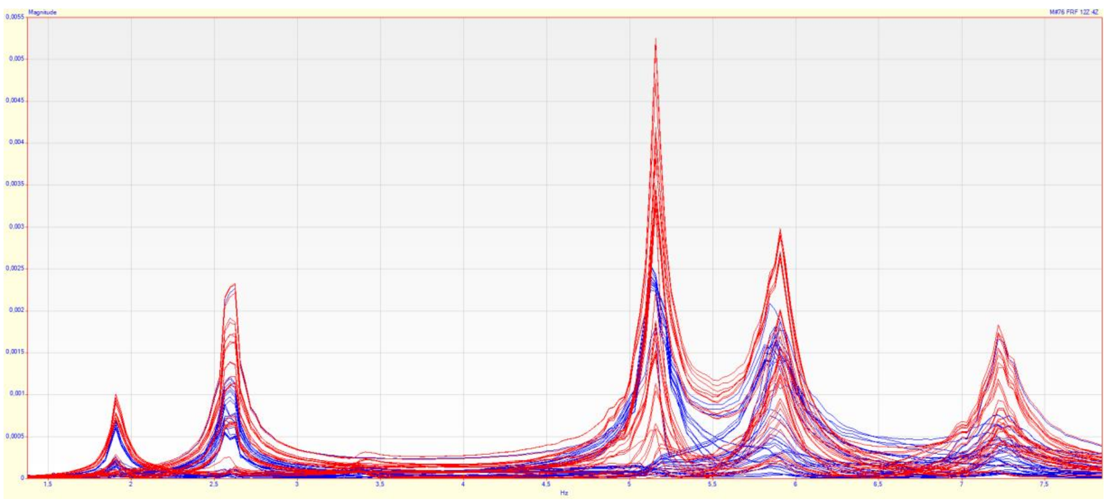
RESULTS HYDRAULIC SHAKER FRF **APPENDIX C**



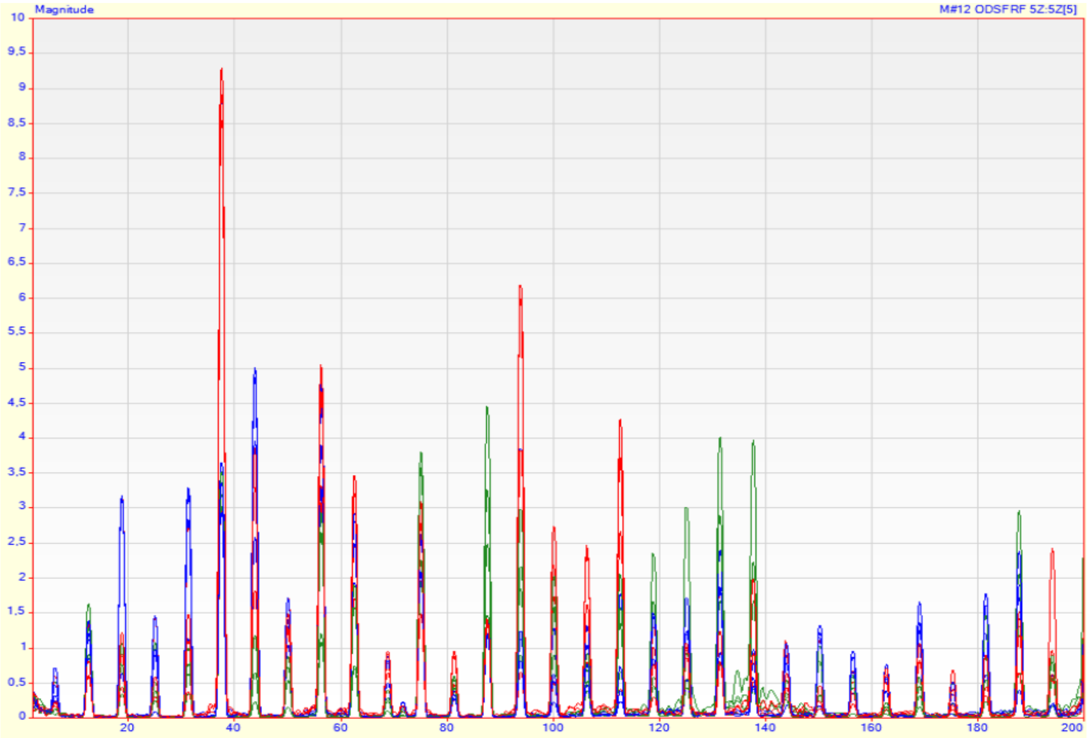
Engine block FRF from the hydraulic shaker measurement. The colors red green and blue represents the longitudinal, transversal and vertical directions.



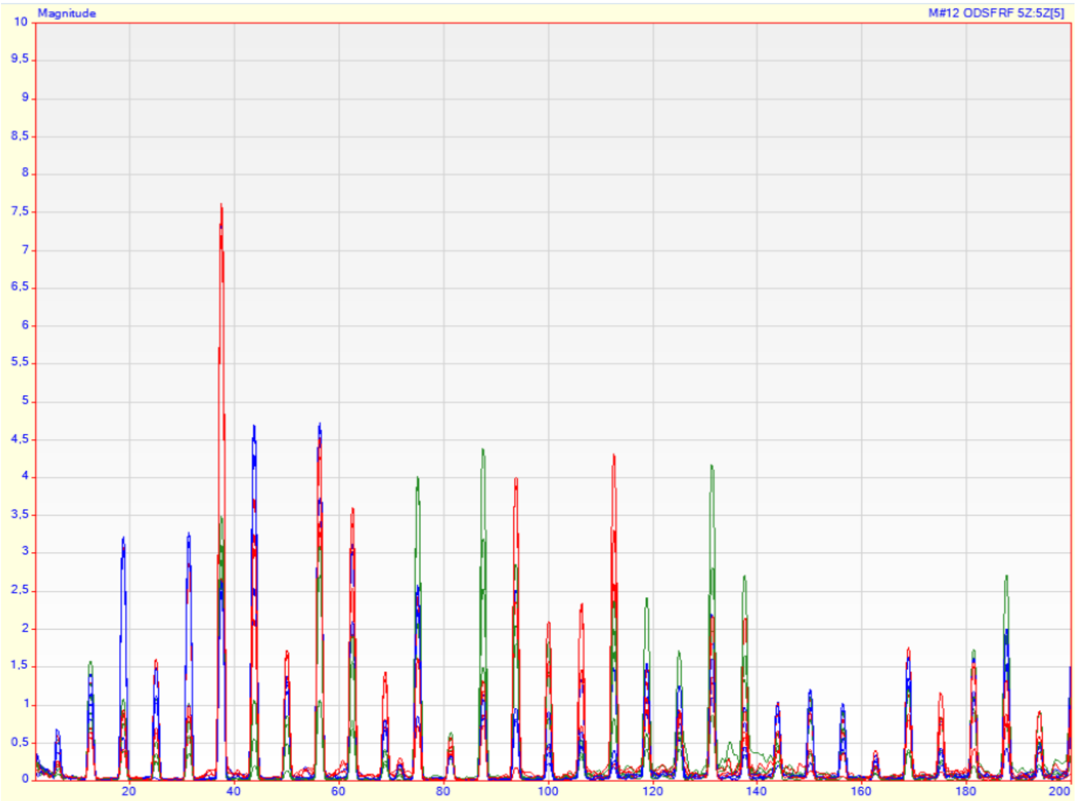
Cabinet and brackets FRF from the hydraulic shaker measurement. Also in this picture the colors represent the directions of longitudinal, transversal and vertical.



Overlapping FRF spectrum, engine block is presented as blue and cabinet as red.



Original bracket ODS Results



Reinforced bracket ODS Results

APPENDIX E

Simple flex rubber 1	Frequency [Hz]	Amplitude [mm/s]
x	11.4	58.1
y	15.0	20.6
z	11.6	28.8

Simple flex rubber 2	Frequency [Hz]	Amplitude [mm/s]
x	10.9	119.2
y	11.9	11.9
z	11.0	43.4

Simple flex rubber 3	Frequency [Hz]	Amplitude [mm/s]
x	7.0	99.1
y	7.3	2.3
z	7.1	22.7

Simple flex rubber 4	Frequency [Hz]	Amplitude [mm/s]
x	5.5	97.5
y	5.8	2.1
z	5.6	15.2

Vertical simple flex rubber 1	Frequency [Hz]	Amplitude [mm/s]
x	18.6	34.6
y	19.2	13.7
z	19.7	66.2

Vertical simple flex rubber 2	Frequency [Hz]	Amplitude [mm/s]
x	17.8	50.2
y	19.3	12.9
z	19.8	64.3

Vertical simple flex rubber 3	Frequency [Hz]	Amplitude [mm/s]
x	15.3	15.6
y	6.6	7.5
z	18.8	135.4

Vertical simple flex rubber 4	Frequency [Hz]	Amplitude [mm/s]
x	12.6	21.9
y	13.1	11.2
z	11.7	111.4

Double flex Steel-Rubber 1	Frequency [Hz]	Amplitude [mm/s]
x	3.1	58.7
y	3.4	9.6
z	3.1	18.1

Double flex Steel-Rubber 2	Frequency [Hz]	Amplitude [mm/s]
x	3.3	55.0
y	3.4	9.3
z	3.4	17.4

Double flex Steel-Rubber 3	Frequency [Hz]	Amplitude [mm/s]
x	4.0	73.0
y	4.0	18.1
z	4.0	18.6

Double flex Steel-Rubber 4	Frequency [Hz]	Amplitude [mm/s]
x	3.3	75.8
y	3.3	14.1
z	3.3	24.2

Double flex Steel-Rubber 5	Frequency [Hz]	Amplitude [mm/s]
x	3.2	67.5
y	3.4	13.5
z	3.4	18.5

Double flex Steel-Rubber 1.2	Frequency [Hz]	Amplitude [mm/s]
x	3.1	55.6
y	3.1	12.5
z	3.2	22.9

Double flex Steel-Rubber 2.2	Frequency [Hz]	Amplitude [mm/s]
x	3.4	60.6
y	3.4	15.0
z	3.4	24.7

Double flex Steel-Rubber 3.2	Frequency [Hz]	Amplitude [mm/s]
x	4.0	67.2
y	4.0	22.3
z	4.1	19.5

Double flex Steel-Rubber 4.2	Frequency [Hz]	Amplitude [mm/s]
x	3.3	62.1
y	3.5	18.0
z	3.4	19.5

Double flex Steel-Rubber 5.2	Frequency [Hz]	Amplitude [mm/s]
x	3.2	53.6
y	3.4	14.7
z	3.4	17.4

Vertical double flex Steel-Rubber 1	Frequency [Hz]	Amplitude [mm/s]
x	5.9	6.6
y	4.3	7.4
z	7.1	62.1

Vertical double flex Steel-Rubber 2	Frequency [Hz]	Amplitude [mm/s]
x	8.0	7.7
y	6.0	7.4
z	7.2	64.9

Vertical double flex Steel-Rubber 3	Frequency [Hz]	Amplitude [mm/s]
x	7.8	11.2
y	6.0	7.1
z	7.7	62.3

Vertical double flex Steel-Rubber 1.2	Frequency [Hz]	Amplitude [mm/s]
x	7.1	15.7
y	7.4	11.6
z	7.0	61.3

Vertical double flex Steel-Rubber 2.2	Frequency [Hz]	Amplitude [mm/s]
x	7.1	15.5
y	7.4	12.1
z	7.0	59.8

Vertical double flex Steel-Rubber 3.2	Frequency [Hz]	Amplitude [mm/s]
x	7.0	10.1
y	7.6	11.8
z	7.0	54.4

Double flex Rubber-Steel 1	Frequency [Hz]	Amplitude [mm/s]
x	3.8	88.3
y	4.4	11.5
z	4.0	20.9

Double flex Rubber-Steel 2	Frequency [Hz]	Amplitude [mm/s]
x	3.8	92.8
y	4.6	11.0
z	3.9	21.9

Double flex Rubber-Steel 3	Frequency [Hz]	Amplitude [mm/s]
x	3.8	89.8
y	4.6	10.2
z	3.9	23.3

Double flex Rubber-Steel 1.2	Frequency [Hz]	Amplitude [mm/s]
x	3.8	66.7
y	4.6	11.8
z	4.0	16.5

Double flex Rubber-Steel 2.2	Frequency [Hz]	Amplitude [mm/s]
x	3.7	68.3
y	4.6	11.6
z	4.0	18.1

Double flex Rubber-Steel 3.2	Frequency [Hz]	Amplitude [mm/s]
x	3.7	62.6
y	4.6	8.8
z	3.9	11.1

Vertical double flex Rubber-Steel 1	Frequency [Hz]	Amplitude [mm/s]
x	10.3	11.8
y	11.6	39.3
z	9.7	133.6

Vertical double flex Rubber-Steel 2	Frequency [Hz]	Amplitude [mm/s]
x	10.9	16.0
y	10.4	32.1
z	9.8	134.1

Vertical double flex Rubber-Steel 3	Frequency [Hz]	Amplitude [mm/s]
x	9.9	5.5
y	10.4	40.5
z	9.9	121.4

Vertical double flex Rubber-Steel 1.2	Frequency [Hz]	Amplitude [mm/s]
x	9.2	6.7
y	12.2	13.8
z	9.6	80.8

Vertical double flex Rubber-Steel 2.2	Frequency [Hz]	Amplitude [mm/s]
x	9.2	8.0
y	10.4	13.8
z	9.7	80.0

Vertical double flex Rubber-Steel 3.2	Frequency [Hz]	Amplitude [mm/s]
x	9.2	12.7
y	10.8	17.0
z	9.7	74.8

Double flex special 1	Frequency [Hz]	Amplitude [mm/s]
x	5.9	60.4
y	6.4	14.6
z	6.5	36.1

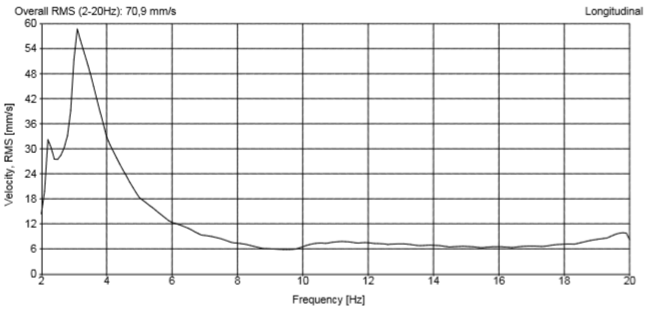
Double flex special 1.2	Frequency [Hz]	Amplitude [mm/s]
x	6.0	84.4
y	6.3	16.2
z	6.6	36.6

Vertical double flex special 1	Frequency [Hz]	Amplitude [mm/s]
x	15.8	27.5
y	6.8	8.7
z	7.4	70.1

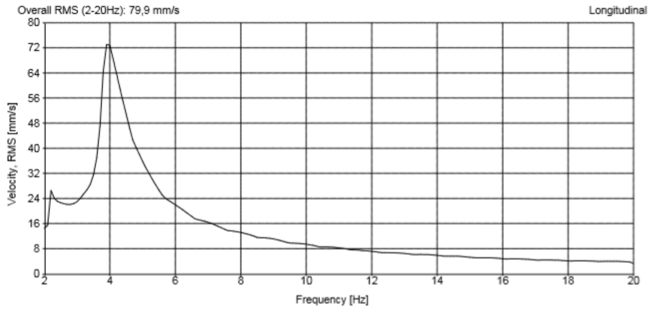
Vertical double flex special 1.2	Frequency [Hz]	Amplitude [mm/s]
x	7.7	21.0
y	7.5	11.7
z	7.3	80.8

APPENDIX F

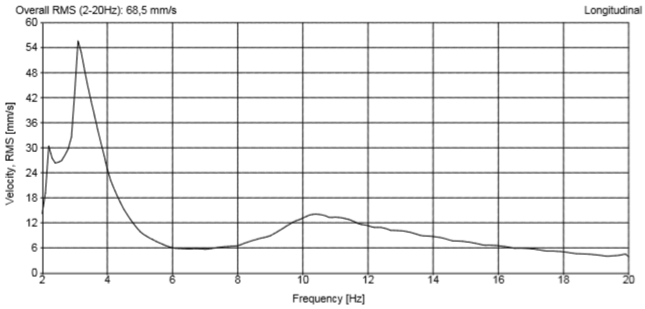
Point : Double flex Steel-Rubber 1



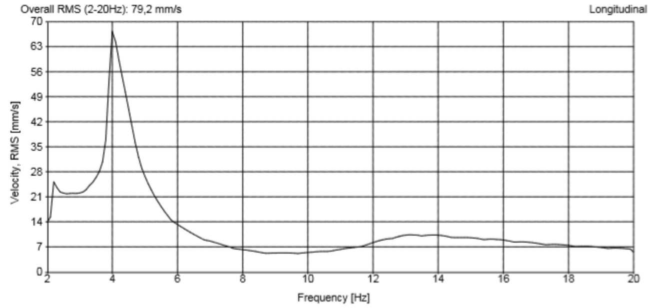
Point : Double flex Steel-Rubber 3



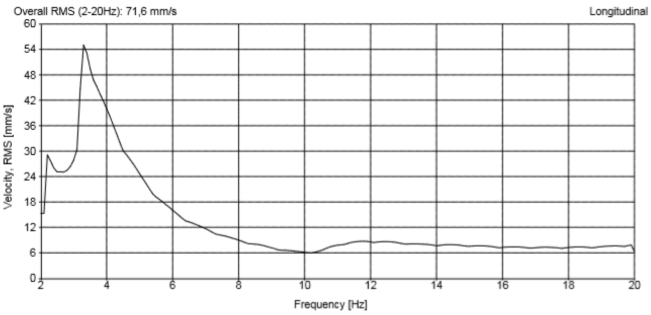
Point : Double flex Steel-Rubber 1.2



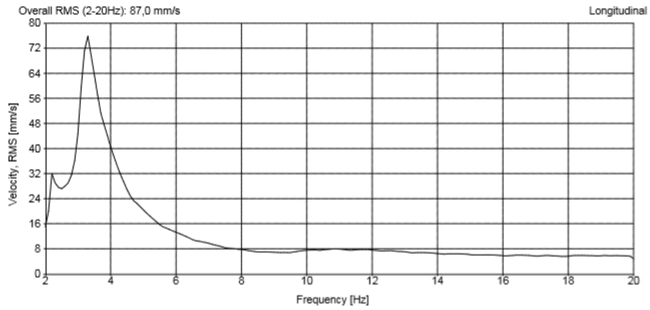
Point : Double flex Steel-Rubber 3.2



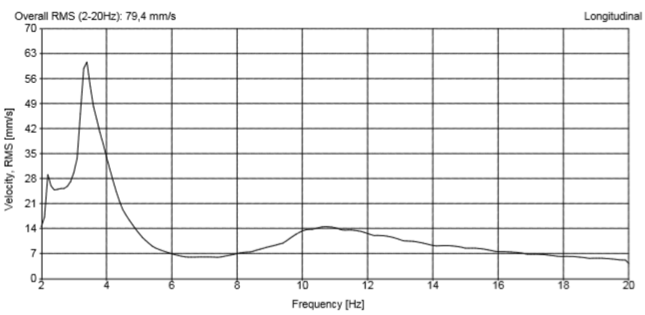
Point : Double flex Steel-Rubber 2



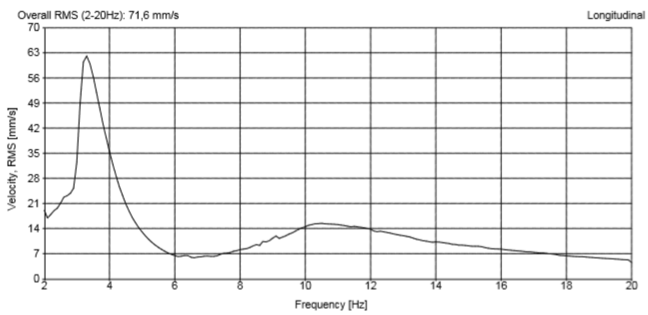
Point : Double flex Steel-Rubber 4



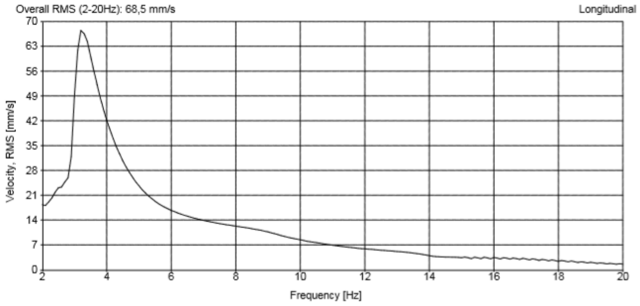
Point : Double flex Steel-Rubber 2.2



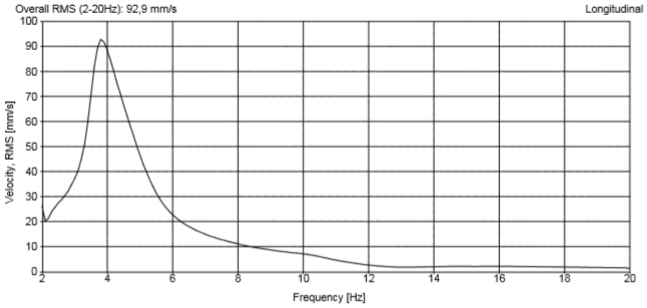
Point : Double flex Steel-Rubber 4.2



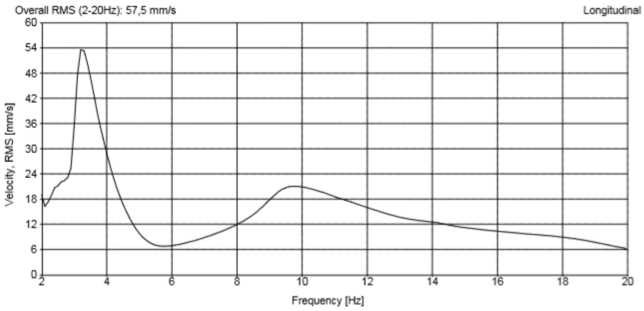
Point : Double flex Steel-Rubber 5



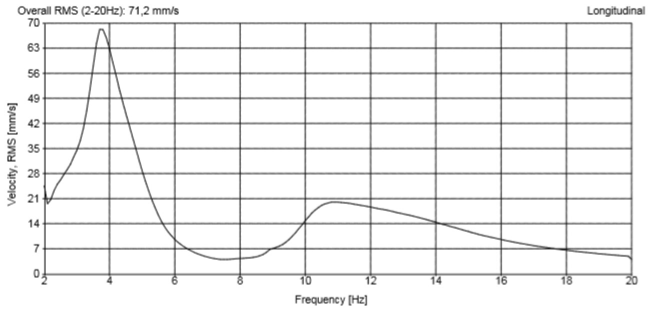
Point : Double flex Rubber-Steel 2



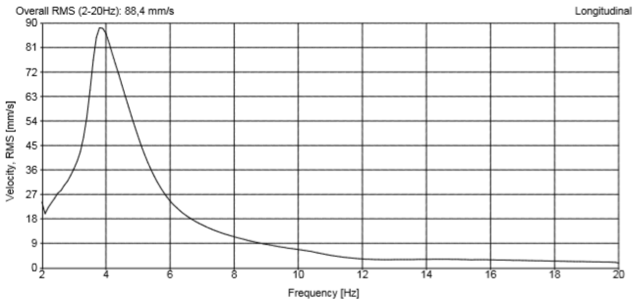
Point : Double flex Steel-Rubber 5.2



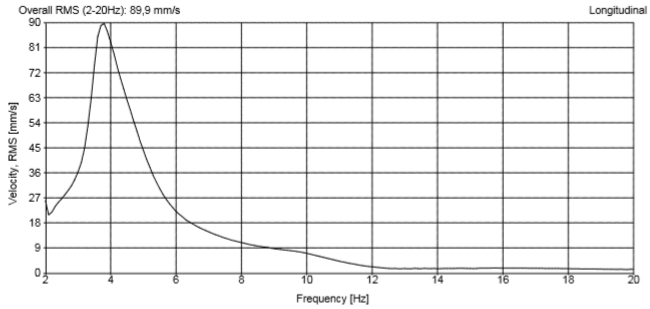
Point : Double flex Rubber-Steel 2.2



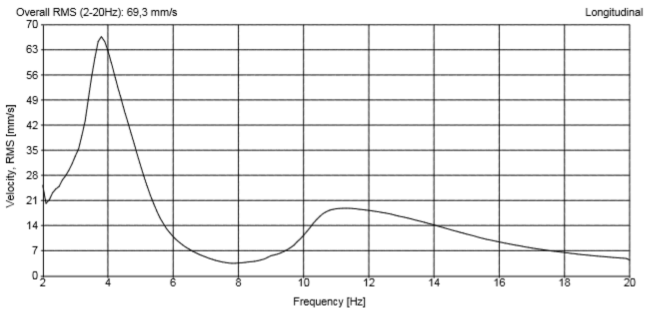
Point : Double flex Rubber-Steel 1



Point : Double flex Rubber-Steel 3



Point : Double flex Rubber-Steel 1.2



Point : Double flex Rubber-Steel 3.2