FreqNess

A vibration Analysis Application for Engine Mount Systems

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Abstract

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This thesis was commissioned by Novia University of Applied Sciences to improve the vibration force isolation of a laboratory engine (Sisudiesel 420 DWI). Excessive vibration is a major contributory factor of internal combustion engines (IC) inefficiencies impacting everything from increased structural fatigue to excessive noise.

The purpose of this thesis was to develop a program in MATLAB to analyze the vibration outcomes of a general engine mount system. The resultant application called FreqNess is able to perform a complete vibration analysis by the introduction of basic engine and vibration isolator specifications. Moreover, this intuitive and user-friendly application was found to be an appropriate tool for monitoring and offering vibration guidance applicable to a wide range of other engine mount systems. In this document, the instructions for future users on how to best operate FreqNess are presented, together with general guidance on vibration isolation theory.

With respect to the specific case and after an analysis performed by FreqNess, the application demonstrated that the current vibration isolators of the Sisudiesel 420 DWI system do not accomplish the customer vibration isolation requirements in a vast range of engine speed scenarios. The findings provide a starting point for the creation of software that can analyze any type of machine where vibration isolators are implemented.

Language: English Keywords: Engine mount system, Vibration isolator, MATLAB

Table of Contents

List of figures

List of tables

1 Introduction

Excessive vibration is one of the most relevant signs and issues of engines as it has a big impact on the quality and useful life of the different parts of it. In internal combustion engines (IC), Noria Corporation (2010) stated that it is not possible to avoid vibration by designing the engine correctly due to the fluctuating disturbances that have a place in it such as the cylinder motion. The consequences of excessive vibration can be an increase of forces in some parts of the engine, structural fatigue, energy dissipation, rapid wear of engine parts such as bearings or gears, or an increase in the noise done by the engine, resulting in excessive noise. These issues lead to a decrease in the performance of the engine, which will mainly depend on the engine and mounts structural characteristics. As indicated by Ramachandran (2012), it is difficult to change the character of the engine, therefore leaving the user with only one thing to adjust the engine mounts.

The mechanical engineer designs the engine to reduce imbalance, as imbalance is the reason behind vibration, while the structural engineer attempts to design the supporting structure to avoid the transmission of vibrations to a weaker part of the system, the application developed in this thesis focuses on engine mount systems. The purpose behind it is to test and analyse the different vibration behaviours, as frequencies and stiffnesses, of a certain system. However, the application analysis is based on the hypothesis that the most restrictive disturbance frequency coincides with the engine speed, which is demonstrated to be veritable in a large range of frequencies. Nevertheless, it needs to be clarified that the results might differ from reality as the application relies on assumptions that will be introduced later in this work. In this case, a more precise analysis needs to be made. Concluding, to be able to establish a certain level of safety an accurate and robust mount selection is needed, hence vibrations are the first topic to look at in the implementation process of an engine. FreqNess aims to teach and guide the customer through the selection process of the most suitable vibration isolators for the specified system.

Vibrations have been analysed and attempted to be reduced by engineers since its first apparition in engines. It has been an issue for a very large amount of time, and it is still generating problems, such as excessive noise or structural fatigue. Since the creation of the engine, the demand for highly efficient, silent, and acceptable vibration levels has been increasing. This entails an attempt to reduce vibrations, however, due to its functioning, it is not achievable to reduce this phenomenon by varying the intrinsic properties of the engine.

Therefore, it is needed to isolate the source of vibration, which in this case is the engine. This isolation can be done by the utilization of different techniques however, mounting systems attract all the attention.

As defined before, this is done by implementing what can be called mounts or vibration isolators which consist of an energy dissipator (damping) and a resilient member (stiffness) (Rao, 2011, pp.769–840). These can be passive or active, the difference being if external energy is needed for it to perform. As described by the company TMC (2016), this energy is usually electricity and is given by the response of multiple sensors to generate an opposite force in the needed direction, these are costlier because of sensors and maintenance conversely, they offer functions that a passive mount is not able to. Nevertheless, the type of vibration isolators that the stated case will focus on is passive mounts. There are many types, although this thesis will target the most frequently used in engine applications: elastomeric, metal springs, and air bellows that can be found in Paulstra Industry (2019) and Mecanocaucho (2021) catalogues. Each of these has different properties and there should be different solutions between these three types of mounts, in spite of this, the specific characteristics will be discussed in this paper in order to help the user to choose the most suitable one for his particular system.

To sum what has been stated before, vibrations are a really important factor to consider when building and mounting an engine and, they are even more significant if the source of vibration is constantly exposed to the public, as in the example that will be analysed throughout the work, due to the catastrophic scenarios that may occur as a result of structural fatigue.

1.1 Aims and objectives

There appears to be conflicting information when searching about vibration isolators. It is difficult to find reliable information if the objective is to choose properly between different types of isolators. Usually, the vibration isolator manufacturers attempt to perform that process for the user by making them fill a document where the user gives different sorts of information, hence the vibration isolator company decide which is the most suitable vibration isolator for the customer. However, this can be done by the user by knowing the same input values as in the paper given by the company. The application designed in the thesis aims to analyse an engine mount system, thus it is possible to test and compare engine mounts between different manufacturers. Moreover, if the customer desires to dig more into vibrations, this thesis will introduce, explain, and exemplify with practical examples the basics of vibration in these types of systems.

Nevertheless, the main aim of this thesis is to utilize the application as a tool, hence the users can analyse wisely between the most frequent vibration isolators to its specific engine case aiming for what the user isolation requirements are. The purpose behind it is that by introducing basic catalogue specifications from the engine and vibration isolators, and their specific disposition a degree of isolation analysis can be performed. These objectives will be possible through the help of the application software called MATLAB App Designer, in which an interface is created to graphically introduce data, perform calculations, and present the analysis. Also, an example will be executed by using the current Sisudiesel 420 DWI engine in Novia's engine laboratory specifications.

In order to meet these aims, the following objectives have been determined:

- To recognize and indicate both theoretical and practical aspects of engine vibration isolation.
- To develop an interactive and intuitive application using MATLAB App Designer.
- To illustrate efficiently the behaviours of the system and warn if there is any resonance situation.
- To exemplify the procedure to run FreqNess based on a Novia Laboratory engine.

1.2 Disposition

These are the chapters involved in this thesis and a summary of what they are about.

- Chapter 2 introduces the reader to general engine topics related to vibration.
- Chapter 3 brings into the text wide-ranging concepts about vibration, focusing on mount vibration analysis.
- Chapter 4 explains to the reader the principal differences between the most common vibration isolators moreover, practical advice is also introduced.
- Chapter 5 introduces MATLAB App designer and establishes the assumptions and considerations of the analysis.

- Chapter 6 and 7 guide the reader in the interface input and results tabs of the application and analyses the given example.
- Chapter 8 evaluates and reflects the results presented.
- Chapter 9 gives suggestions to the user based on the results and introduces the future analysis.

2 Vibration theoretical concepts

In this section, vibration concepts related to the thesis will be introduced. This is done to reference them in the following chapters and, give more information to the reader so it would be easier to follow the procedure performed. It is important to remark that there will be concepts that will not be used in the application development. However, these are needed to successfully understand and give some guidance to the performed assumptions and calculations.

2.1 Internal combustion engine

In this subsection, different types of concepts regarding a general internal combustion engine will be introduced. As the engine mount system will be the one on which the work will focus on an introduction of parameters related to the source of vibration such as frequency order and firing frequencies are imperative. This subsection will help the readers to have a wider perspective about different aspects of engine vibration behaviours.

2.1.1 Case-study engine

An engine can be used in many different fields, for example, to generate electricity. This thesis is done on a request to isolate vibrations from an engine to a frame. However, the real scope of this thesis is to give a first analysis of the vibration behaviour of any 4-mount internal combustion engine system. As an example, an engine of the Novia’s laboratory (Sisudiesel 420 DWI) will be tested by the final software. The type of designation stands for *4* is the number of cylinders, *20 D* is the basic type of engine, *W* designates that the engine is equipped with a by-pass turbocharger, and *I* stand for the intercooler, which in this case is air to water. It is a four-stroke, water-cooled, direct-injection in-line diesel engine. The engine also has a balancer unit, this unit has the function to even out the vibration forces exerted by the movement of the crank mechanism and the pistons explained by Sisu Diesel (n.d.) in their engine specification document. However, the effectiveness of the balancer is not enough to reduce the vibration, therefore the engine still vibrates at a high vibration severity. Thus, requiring a vibration severity improvement. The study engine mount system is shown in Figure 1.

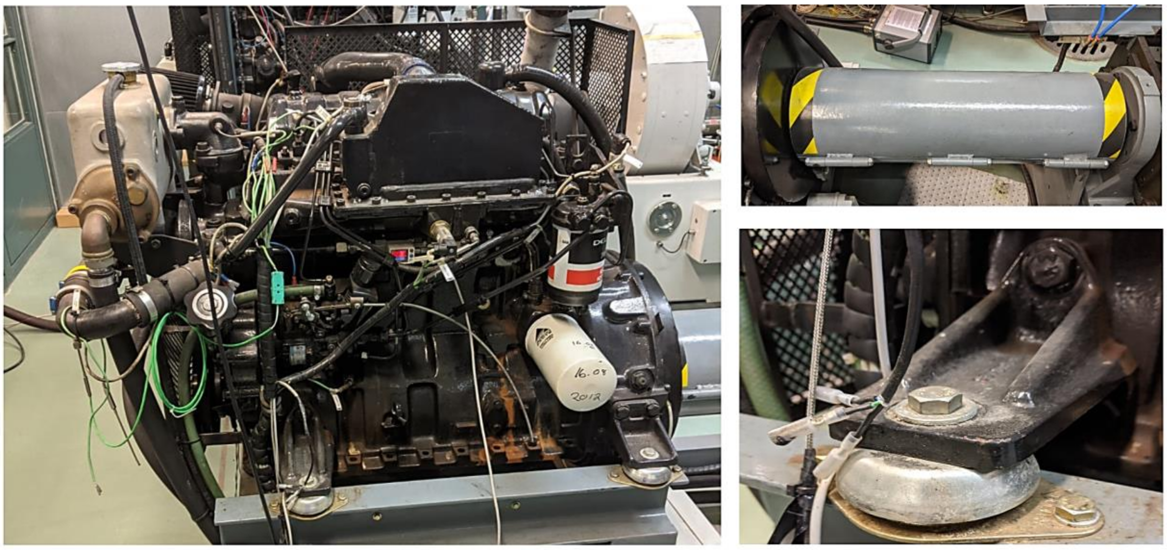


Figure 1. Sisu diesel 420 DWI study example engine configuration

Currently, as seen in the bottom right picture of Figure 1, there are elastomeric mounts implemented in the system. However, the vibrations transmitted thought do not fit the customer isolation request. As mentioned, the thesis aims to do an analysis of the vibration behavior of a system, hence the user can test and decide between different vibration isolators to improve the degree of isolation of a specific system, thus reducing the global vibration transmissibility in it. The parameter that will be crucial when deciding either if the system is acceptable or not is called degree of isolation. The stated parameter among more vibration concepts will be introduced in the following sections, with the aim of a correct understanding of the behavior of the system. In addition, the function of the system is to generate electricity in a Genset configuration.

This thesis attempts to analyse the system by considering that the engine speed is the most restrictive disturbance frequency, hence it is possible to examine the system vibration behaviours in a wide range of engine speeds. This is generalized to any engine by knowing basic inertia characteristics of them and the engine mounts. The data needed to perform the analysis can be found in their respective catalogues. As stated in the 1.1 Aims and objectives section, one of the objectives is to facilitate the analysis to the user, hence this work also aims to guide the user to find and calculate the parameters needed to perform the analysis. Once the input data is introduced into the software, this will give either an approximation or a detailed value of degree of isolation in both translational and rotational axes. The difference between a specific or an approximate analysis relies on how specific the introduced data is. Also, the program will perform a graphic analysis the speed of the engine, hence showing to the user which vibration behaviour the system is found to be (amplification, resonance, or isolation). Also, if the given case is found that is susceptible to resonance or amplification, it will warn the user. If the resonance or amplification case is found, the customer will have to either raise the engine speed, change the disposition of the mounts, or move to another type of mount. This is done to ease the way that this has been done and increase the security, comfort, and safety of wherever the application of the engine will be. The specific Sisudiesel 420 DWI system will be analysed by following the stated procedure.

2.1.2 Frequencies orders

As Lassfolk (2018) explained in his thesis, each component of an engine has its own excitation frequency. A way to categorize the sort of frequencies in an internal combustion (IC) engine is by using orders. Orders of frequencies are a range of frequencies depending on the engine's running speed. The formula to calculate in which order a certain frequency belongs is given by dividing the running speed of the engine by 60 (rev/min) and multiplying it with the order as shown in eq (2.1).

$$\chi \cdot \frac{n}{60}\ [Hz] \tag{2.1}$$

Where n is the engine running speed and χ is the order. An example of it is that if the engine is running at a speed of 1500 rpm, the first order would be $\frac{1500}{60}$ $[Hz]$ which simplified it would give $25\ [Hz]$. So, if there is any frequency of any part of the engine that would have the value of $\geq 50\ [Hz](25 \times 2)$ it would be considered a second-order vibration.

2.1.3 Firing frequency order

A form to describe the rate at which the cylinders fire in an engine is called firing frequency. However, there is a difference depending on the strokes of the engine. If it is a 2-stroke, the cylinders fire every revolution. Besides, in a 4-stroke engine, pistons will fire every second revolution (Wärtsilä, n.d.). The firing frequency order can be calculated by knowing the type of engine and the number of cylinders in a row as shown in eq (2.2).

$$Firing\ frequency\ order = \frac{Number\ of\ cylinders\ in\ a\ row}{n_s}\ where\ n_s = 1{,}2 \qquad (2.2)$$

Where n_s has the value of 1 if the engine is a 2-stroke engine and the value of 2 if it is a 4-stroke engine. These specials orders are used to identify in which range of frequencies the highest levels of torque variation will occur. Hence, when there is a variation of torque, simultaneously mean an amplitude peak, as can be seen in Figure 2.

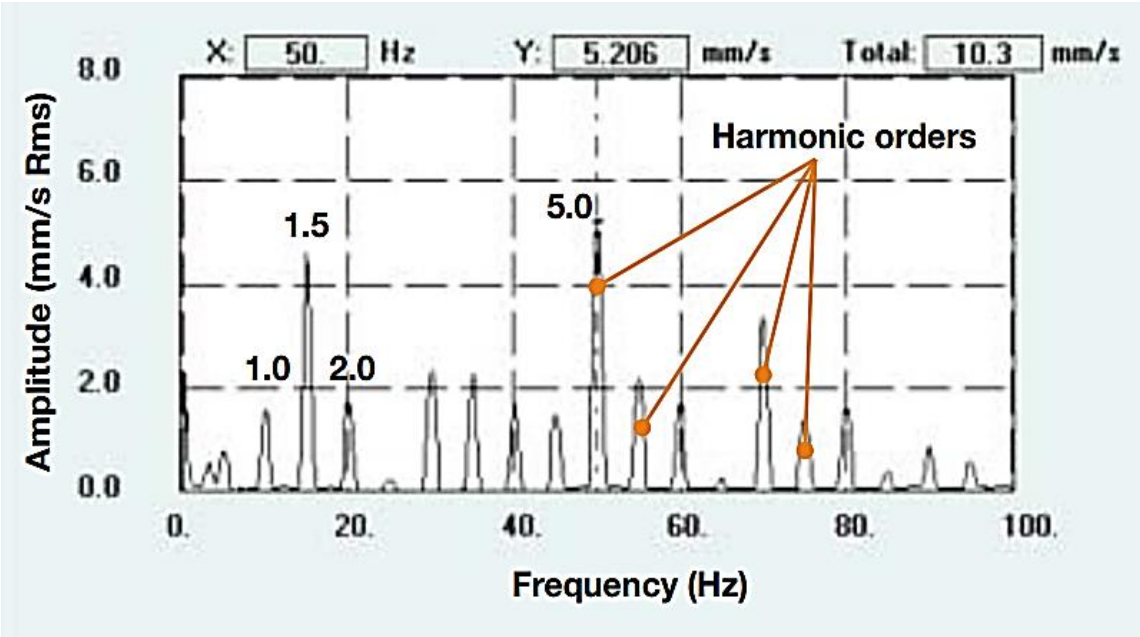


Figure 2. Vibration spectrum of a 4-stroke engine (Tienhaara, 2004)

As specified by Tienhaara (2004) in Wärstilä vibration guidelines, the results of Figure 2 were measured by a vibration modern sensor in a 4-stroke engine. As it can be seen the amplitude peaks coincide with the firing or harmonic orders of the engine, which as will be seen in the vibration theory sections, an amplitude peak denotes resonance. In addition, Tienhaara emphasizes half-firing orders, that take importance in 4-stroke engines as reflected in the figure. This analysis gives more reliability to the assumption of considering the engine speed as the only disturbance frequency, however, every case is different, and the assumption may not be veritable in some specific systems. Conversely, Lassfolk (2018) stated that the assumption indicated before decreases its reliability in the function of the firing frequency of the system is. As the firing frequency increases, the vibration behavior gets influenced by other engine comportments such as local vibration which decreases the dependability of the system with the speed assumption. Nonetheless, in a range of frequencies from 0 to 65 Hz, or in other words, from 0 to 4000 rpm, this assumption is considered valid, which is reflected in the slider implemented in the results sections of FreqNess, which will be detailed later in this work.

3 Vibration

Rayner (1995) defined that the reciprocating, oscillating, or another periodic movement of an elastic or rigid body or medium around a point of equilibrium is termed as vibration. This phenomenon occurs when a physical system is moved from its equilibrium state and is tolerable to specific forces to re-establish equilibrium, as indicated by Britannica.com (2021). Generally, when it is talked regarding vibration, it is usually referred to as a low amplitude and high-frequency movement that occurs in bodies or systems that allow the storage of kinetical and potential energy. In fact, in a vibration movement, there is a fluctuant trade between those two types of energy.

The simplest vibration system is the mass-spring system, in whose is restricted to one degree of freedom. A degree of freedom is the capability of a system to move or rotate on a certain axis. In the 2-D spring-mass example the movement can only be achieved linearly, the reason behind its one degree of freedom. However, in a real 3-D system, as in an internal combustion engine, the system is free on every axis. The system can move and rotate in every direction, giving as a result 6 degrees of freedom. These movements have been animated in Blender to help the user while processing the results of the analysis. These movements are shown in Figure 3.

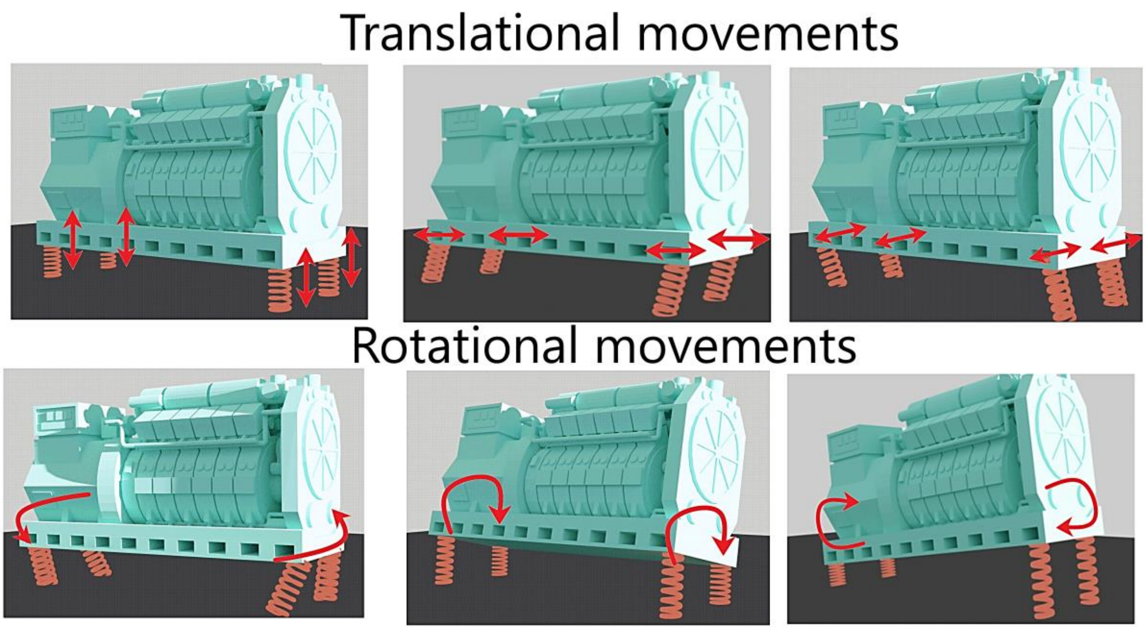


Figure 3. 6 DOF system movements animated in Blender

Returning to the spring-mass system, this one can be analysed by using the second Newton law. The system and its free-body diagram are shown in Figure 4.

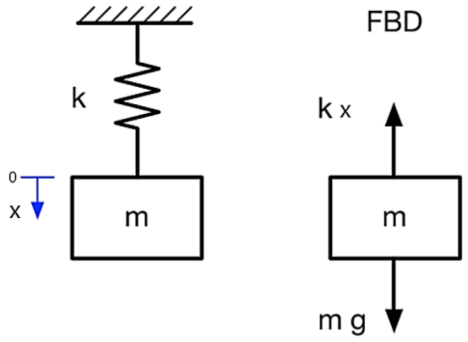


Figure 4. Spring-mass system and FBD analysis (Edited from Leancrew, 2014)

To not overcomplicate the analysis, the zero of the axes is when the system is in the static equilibrium position. It is important to state that the reference does not correspond to the not deformed length of the spring. The procedure in the numerical analysis is:

$$\sum F = m\ddot{x} \ \rightarrow -kx = m\ddot{x} \rightarrow m\ddot{x} + kx = 0$$

Where m is the mass of the body, k is known as the stiffness of the spring, and $\ddot{x}$ and x the acceleration and the displacement of the body respectively referred to as the reference system. This equation results to be a 2nd order ODE (ordinary differential equation), whose solutions are shown in the following equation.

$$x(t) = A\,cos(wt - \varphi)$$

Where $x(t)$ is the displacement of the spring depending on time, A is the amplitude of the system and has the same units as the displacement [mm], w is the frequency expressed in [rad/s], t is the time, and φ is the phase angle. Nevertheless, this system would oscillate in an infinite amount of time however, that is not realistic. Damping is responsible to stop the movement at some time. Then, it is important to redefine the system as illustrated in Figure 5.

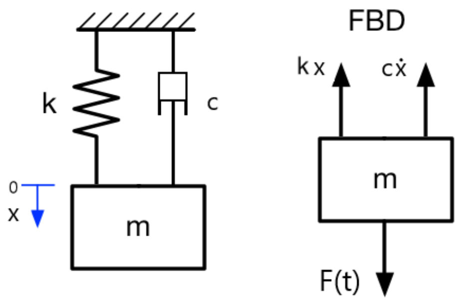


Figure 5. Damped 1 DOF system (Edited from Leancrew, 2014)

In this more realistic case, and following the same steps that in the first proposed system the resulted equation that governs the movement of the body is:

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

Where c is a constant of proportionality and $F(t)$ the excitation force and $\dot{x}$ the speed of the body. The damped force is usually proportional to the speed of the body, but it has the opposite way.

Scoping to the specified case, in engines the excitation forces and torques generate two types of vibrations: longitudinal and torsional. These excitation forces cannot be avoided in an engine as oscillating mass, imbalance, mechanisms of motion, and gas forces are essential processes. On one hand, longitudinal vibration is produced either in the engine block or in the mounts by for example, when the connecting rod (reciprocating movement) and the crankshaft mechanism transforms pressure into a rotating motion (Karan, 2019). On the other one, torsional vibration is the twisting motion due to the turn that, normally, a shaft receives in every combustion stroke due to the fluctuating combustion pressures (Costa, 2018). This type of vibration arises in the crankshaft of the engine mainly as a result of the tangential forces on it when is rotating. This section aims to introduce and explain vibrations to a point that both torsional and longitudinal vibration parameters can be easily understood in the results section of the App.

3.1 Concepts and parameters of a vibrating system

Inside the vibration field, some concepts must be introduced before analysing a vibration system such as stiffness, damping, and natural frequency. The concepts will be introduced to utilize them in the analysis and calculation sections, and also to explain to the customer the process that has been followed during the programming stage of FreqNess.

3.1.1 Stiffness

The resistance of an elastic body to bend in response to an applied torque or force is called stiffness. However, flexibility is the complementary concept, the more stiff and elastic body is, the less flexible it is. In the simplest case, linear vibration and with a statical force applied on the body, the static stiffness parameter can be calculated by eq (3.1).

$$k_s = \frac{F}{\Delta x} \tag{3.1}$$

Where F is the applied force on the spring and Δx its corresponding linear deformation or deflection. In a practical way of thinking, the statical stiffness defines the amount of statical force done by an object if it is compressed one meter, therefore being the units of the stiffness N/m. This formula is used when the spring or the elastic body is assumed to have no mass, this assumption can be done when there are big weight differences between the source of vibration and spring, as happens to be the case in engine mount systems. Nevertheless, when the system gets more complex dynamic stiffness appears. As stated in the Trelleborg (2021) anti-vibration portfolio, the stiffness of the amount changes when the force applied varies with time. However, as told by Trelleborg, the dynamic stiffness can be considered as an unchanged parameter between a range of frequencies from 5 to 80 Hz, this consideration comprehends the normal range of frequencies of vibration isolators (from 5 to 16 Hz) in most catalogues. A parameter to compare with more ease both static and dynamic stiffnesses is called the dynamic factor and is introduced in eq (3.2).

$$D_f = \frac{k_d}{k_s} \tag{3.2}$$

This parameter generally accomplishes the condition that $D_f \geq 1$, meaning that the dynamic stiffness is usually bigger than the static one. To sum what has been stated before, the importance of dynamic stiffness plays an important role in the analysis of the system, this is reflected in the FreqNess vibration isolator section.

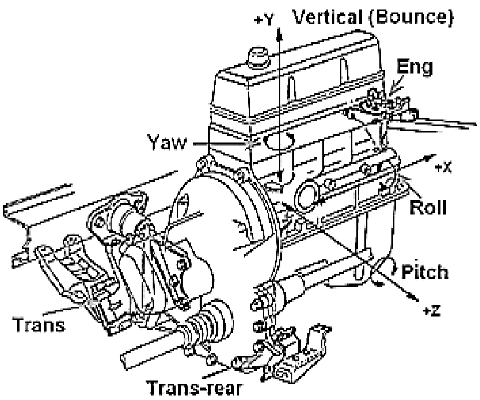
3.1.2 Natural frequency

The frequency in that the system resonates is called natural frequency, and it depends on two parameters: the stiffness of the spring and the mass of the object that is resonating. Natural frequency can be calculated using eq (3.3) when the disturbance is a force and eq (3.4) when the disturbance is a torque.

$$f_{ni} = \frac{1}{2\pi} \cdot \sqrt{\frac{k_{f,i} \cdot D_f}{m_{tot}}} \quad where\ i = x, y, z \tag{3.3}$$

$$f_{ni} = \frac{1}{2\pi} \cdot \sqrt{\frac{k_{t,i} \cdot D_f}{I_{tot}}} \quad where\ i = x, y, z \tag{3.4}$$

Where, f_{ni} is the natural frequency in Hertz (Hz), $k_{f,i}$ and $k_{t,i}$ (Newton/meter) is the stiffness due to a force and a torque respectively, D_f is the corresponding dynamic factor of the mount, and finally m_{tot} (kg) and I_{tot} (kg×m^2) is the mass of the body and the moment of inertia respectively. The various movements that an engine, or every rigid body, is able to perform are shown in the left picture of Figure 6.



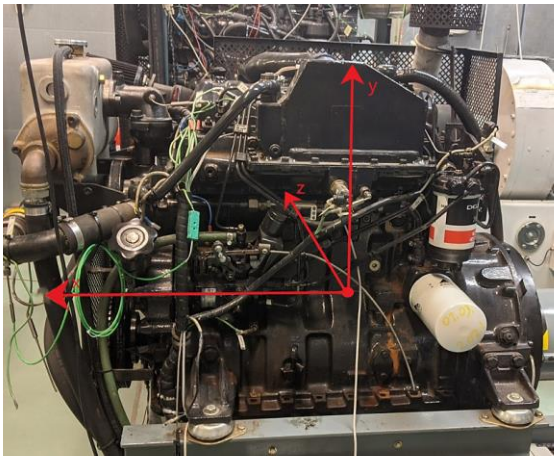


Figure 6. 6 DOF engine model (Edited from Ramachandran, 2012) and reference system on the example engine

As can be seen in the right part of Figure 6, the identification of the reference system in the example engine is done. This aims to help the reader or customer when this step needs to be performed as if his step is not done correctly, the results of the analysis will have a lack of coherence. In the force disturbance case, eq (3.3), i refers to the axes where the mass is oscillating (transversal, vertical or longitudinal force) and the stiffness is usually calculated as a ratio of the vertical stiffness of the mount, expressed by the vibration isolators companies as axial to radial or axial to transversal ratios. Besides, in the torque disturbance case, eq (3.4), i refers to the torque that causes the axis to rotate (pitching, rolling, or yawing torque) where the torsional stiffness depends on both the individual vertical stiffness and position of the mounts concerning the c.o.g of the body.

Every object of a system has its natural frequency, so it must be calculated for each of the receiver components (mounts) of the system, on the contrary as in the analysis the system will be transformed as one degree of freedom per movement the engine can perform (mode), there will be one output frequency per the mode of the engine. Usually, the vertical natural frequency can be calculated by knowing how much static load is in each mount by using specification graphics. There are as many natural frequencies as the degrees of freedom the system has. In a more precise analysis, the natural frequencies are coupled, on the other hand, in the analysis performed by FreqNess the coupling between frequencies is assumed to be negligible. The negligibility of this assumption can be done after contrasting results with a methodology that considers the coupling of frequencies. The acceptance and contrast were done and approved by the lead supervisor.

3.1.3 Resonance

It occurs resonance when the frequency of the excitation force approaches the natural frequency of the body. When this event takes place, the amplitude starts increasing as seen in Figure 7.

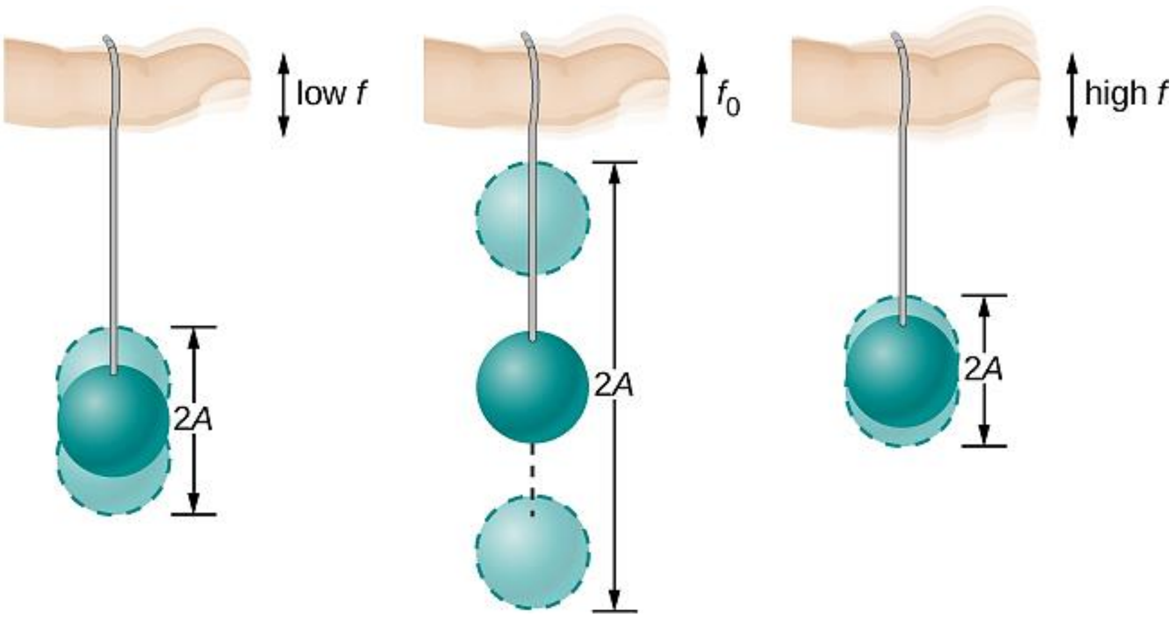


Figure 7. Finger-band-ball, practical example of resonance (Lumenlearning, 2016)

As seen in the figure, the amplitude increases dramatically when the finger does a force with the natural frequency (f_n) of the system: paddleball and a rubber band. In higher and lower frequencies, the energy transferred from the finger to the system is less efficient, which leads the system to a low amplitude vibration. Summarizing what has been stated before with the finger-band-ball example, an increase of amplitude carries the body to high vibration levels, which end up in a more abrupt vibrational motion.

3.1.4 Torsional vibrating system

For a torsional system, the movement is due to the rotation around an axis, as shown in Figure 8.

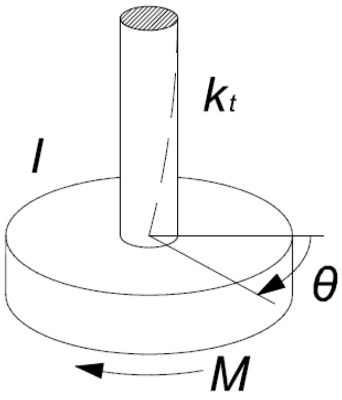


Figure 8. Torsional vibrating system (Roca, 2019)

The equations and procedure of this system are equivalent to the linear system discussed before. The analogy follows the next steps: linear coordinates changes to angular coordinates, force change to torque, and mass changes to the moment of inertia. In this case, the spring is treated as a rotational spring, and the torque that it does is proportional to the angular displacement as shown in eq (3.5).

$$M = -k_t\theta \tag{3.5}$$

The differential equation of movement, natural frequency, and movement equation is also equivalent to the ones in a linear spring-mass system.

$$I\ddot{\theta} + k_t\theta = 0$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_t}{I}} \tag{3.6}$$

$$\theta(t) = \Phi\, cos(\omega_n t - \varphi)$$

Where I is the moment of inertia, θ is the angular displacement, k_t is the torsional stiffness, and Φ the amplitude of the vibration movement. The calculation of the torsional stiffness for yawing, pitching, and rolling will be performed in this section, using as an example the mounted engine case. First, the analysis followed to find the rolling stiffness is shown. The other two analysis follow the same philosophy as the torsional or rolling one, the formula for the yawing and pitching stiffness will be directly given. First, a triangle geometrical relation is done.

$$tan(\Delta\theta) = \frac{\Delta y}{x}$$

It is assumed that $\Delta\theta$ is the increment of the angle of inclination in a rolling way, Δy is the vertical deflection due to the inclination of the body and x is the distance from the c.o.g of the engine to the MSA (middle section axis) of the mount. The tangent geometrical relation is done assuming that $\theta \to 0$, as the vibration movements have small paths, consequently the inclination curve can be treated as a straight line. Hence, the tangent relation can be applied to the case. By reformatting and doing an analogy of eq (3.1) it gives:

$$k_{roll} = \frac{T_{roll}}{\Delta\theta}, \qquad where\ T_{roll} = F_y \cdot z,$$

$$Hence\ k_{roll} = \frac{F_y \cdot z}{\Delta\theta}, \qquad being\ F_y = k_{vert} \cdot \Delta_y = K_v \cdot z \cdot \tan(\Delta\theta)$$

$$k_{roll} = \frac{z^2 \cdot k_{vert} \cdot tan(\Delta\theta)}{\Delta\theta}, \qquad defining\ \Delta\theta = 1\ rad$$

It is found that:

$$k_{roll} = 1{,}56 \cdot z^2 \cdot k_{vert}\ [N \cdot m\ /\ rad] \tag{3.7}$$

Analogously:

$$k_{pitch} = 1{,}56 \cdot x^2 \cdot k_{vert}\ \ [N \cdot m\ /\ rad] \tag{3.8}$$

$$k_{yaw} = 1{,}56 \cdot (x^2 + z^2) \cdot k_{vert}\ \ [N \cdot m\ /\ rad] \tag{3.9}$$

As seen, eq 3.10, 3.11, and 3.12 depending on the position of each mount. Therefore, every single mount will have a specified value, which is valid if the mount disposition is not symmetrical. As the springs are placed in parallel, these can be added linearly in each respective mode. Hence, the equivalent stiffness would consider each mount value, in contrast with the translational stiffnesses which have the same value. Consequently, in the translational modes the equivalent stiffness, in the specified four mount engine system, is four times the stiffness of one mount.

3.1.5 Damping

Vibration systems get isolated from an excitation force via a damper. The levels of vibration get reduced when using a damper. This is due to the decrement of energy that the damper implements. There is a clear difference between vibration isolation and vibration damping. In the vibration damping case, the objective is to reduce the global energy of the system by transforming the energy into heat, while the vibration isolation case tries to keep the natural frequency below the excitation frequency (Sorbothane, 2015). However, there are cases that a damper is needed, for example when the excitation frequencies cannot be shifted away from the natural frequency of the body, thus having a resonant system, a damping configuration is needed. This case is explained to help the reader to understand what can be done if its system is found in resonance. To be able to solve and understand a damped vibrating system it is considered that the excitation force $F(t)$ is zero, hence leading to the expression found in eq (3.13).

$$m\ddot{x} + c\dot{x} + kx = 0 \tag{3.13}$$

Eq (3.13) gets transformed by solving the ODE giving a result $x = Ce^{st}$, C being a initial condition constant. Substituting this solution to eq (3.13) it gives:

$$mCs^2e^{st} + cCse^{st} + kCe^{st} = 0 \rightarrow (ms^2 + cs + k)e^{st} = 0$$

Where s is the only value needed to solve the equation, which results to be a 2nd-grade equation whose solutions are:

$$s_1, s_2 = \frac{-c \pm \sqrt{c^2 + 4mk}}{2m} = -\frac{c}{2m} \pm \sqrt{\frac{c^2}{4m^2} - \frac{k}{m}}$$

Where depending on the s_1, s_2 values three cases are found:

- $if\ s_1, s_2 \in \mathbb{R}\ and\ s_1 = s_2$ it is called a critically damped system.
- $if\ s_1, s_2 \in \mathbb{R}\ and\ s_1 \neq s_2$ it is called an overdamped system.
- $if\ s_1, s_2 \in imaginary\ numbers$ it is called an underdamped system.

When it is said that a system is critically damped it means that the conditions stated before are accomplished. In addition to it, it is agreed that in this case $\frac{c^2}{4m^2} = \frac{k}{m}$. In this special scenario, the damping constant ζ can be calculated following the procedure:

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

Where c_c has a different value depending on in which case the vibration system is. In the one that it concerns, c_c can be calculated as:

$$c_c = 2\sqrt{km} = 2m\omega_n = c$$

Therefore, in this case, $\zeta = 1$. When this case occurs, the time that the mass returns to the equilibrium position is the shortest out of the three cases.

The over-damped case is given when the damping of the system is bigger than the critical damping, therefore it can be said that $\frac{c^2}{4m^2} > \frac{k}{m}$. To sum up, the under-damped case system is the opposite of the over-damped case, in this case $\frac{c^2}{4m^2} < \frac{k}{m}$. This is a very complex case where numerical imaginary theory is applied.

3.1.6 Creep rate

This parameter is intrinsic to vibration isolators. It defines time-dependent deformations when a persistent load is applied. In any elastic material, when a load is applied there is an increase of length in the stress direction. The creep rate is defined by eq (3.14).

$$t = \frac{x_1 - x_0}{x_0} \cdot 100\ [\%] \tag{3.14}$$

Defined by x_1 as the length of the body after the load application and x_0 as the initial length. The creep rate is given in percentage (%) and it depends on the geometry of the body and how it is worked (Mecanocaucho, 2021).

3.1.7 Transmissibility

This concept summarizes the special behaviours of a vibrating system. The transmissibility of a system is a ratio between the input force and the output force. In the engine-frame system, for instance, it would be the force that the frame receives through the isolators divided by the force generated by the source, the engine. In a flexible mounting system, the solution for the transmissibility factor is illustrated in eq (3.15).

$$\lambda = \frac{F'_M}{F_M} = \sqrt{\frac{1 + 4\zeta^2}{\left(1 - \frac{\omega^2}{\omega_n^2}\right)^2 + 4\zeta^2}} \tag{3.15}$$

In Figure 9 it can be seen the performance of transmissibility in front of the frequency ratio of different damping ratios.

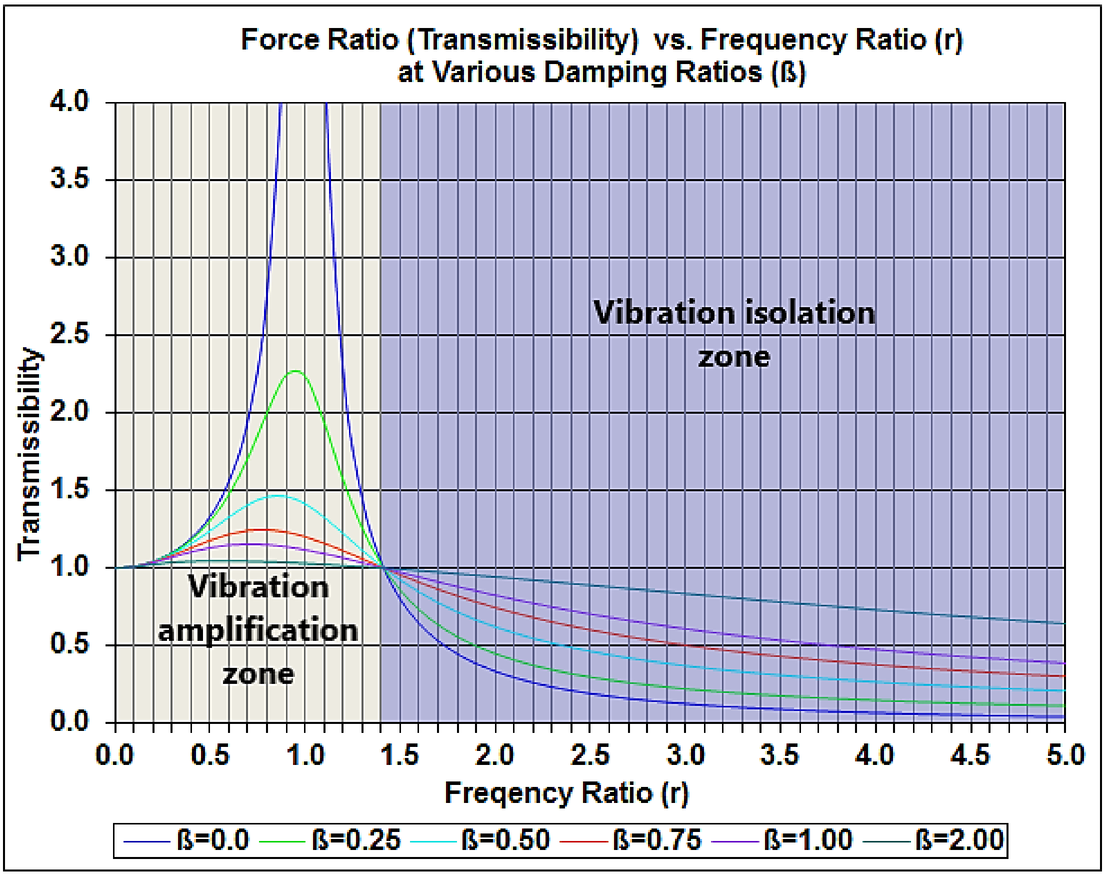


Figure 9. System transmissibility behaviours (Edited from Kane, 2012)

There is a peak of every damping ratio system, and it is when the frequency ratio (r) has the value of 1, which means that $f_f = f_n$. Also, when the force transmissibility is lower than 1, the zone is called the vibration amplification zone. Besides, when is higher than 1, it is called the vibration isolation zone and is the zone that this work aims for. Another important point is that as the damping ratio increases the isolation zone gets reduced, so if the objective is to have a big isolation range, the less damping the better. Also, isolation and transmissibility ratios are directly related, being one the opposite of the other. The damping coefficient will be the most relevant factor that differentiates one type of isolator from another in this type of analysis.

3.1.8 Degree of isolation

The degree of isolation or attenuation (a) describes how well isolated is the system. It is the most crucial parameter to consider when the comparison between different types of isolators is made and an analysis is performed. As transmissibility is already introduced, the degree of isolation is directly linked with it, therefore with a given induced frequency (ω), the degree of isolation would only depend on the natural frequency of the isolator. The formula is given in eq (3.16).

$$a = 1 - \lambda \, [\%] \tag{3.16}$$

In this work, the customer example establishes that outputs of more than 95% per cent of the degree of isolation will be accepted. Every other degree of isolation result $< 95\%$ will be declined. This will depend on the case, however, if it is needed to know which is the borderline in any specified case, it is recommended to read Table 1, where the vibration severity is regulated by ISO 12372.

4 Vibration isolators

Vibration isolators, or also called motor mounts create a mechanical link between the engine and the frame or object where it is attached. Depending on what type of vibration isolator it can be formed by rubber, metal, or other types of materials. The differences between three of the most typical engine mounts are explained later in this section. Knowing the differences between the different vibration isolators will help the user to know what type of isolator suits the most to its specific system. The practical differences and advice will be the ones that will make a difference when deciding the isolators, as the results in the analysis give a similar outcome when the frequency ratio is located in the isolation zone (see Figure 9). However, the composition of most vibration isolators is a relevant topic for the user. These are often disposed of three parts, as can be seen in Figure 10.

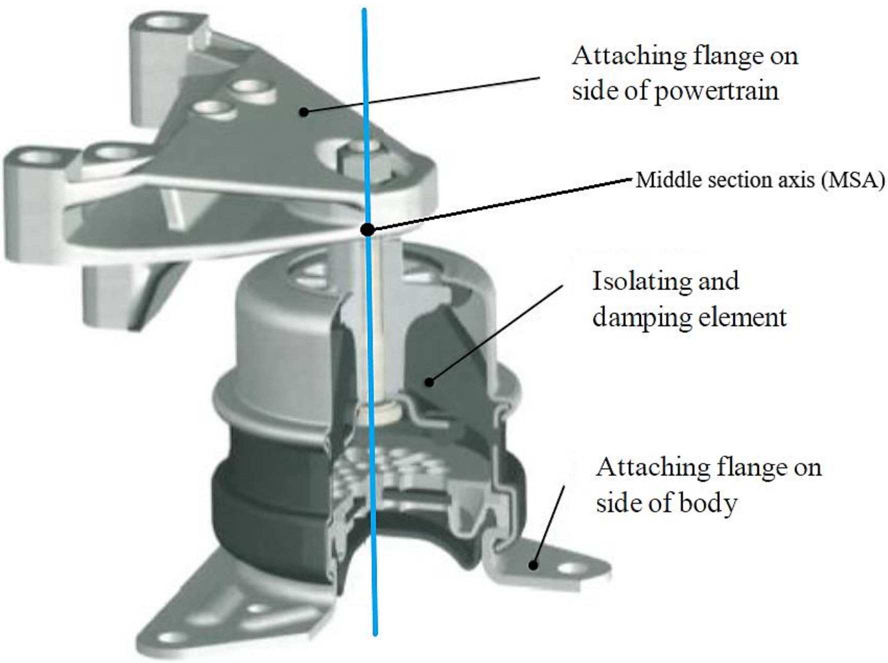


Figure 10. Engine mount composition (Edited from Hempel, 2002)

Where metal support is attached to the engine and conducts the forces to the vibration isolator, that in the function of its primary purpose the layout of the component can vary, this is usually found in the engine mount catalogues. Additionally, an attaching flange is implemented to connect the mount to a rigid body as described by Hempel (2002). Moreover, the MSI (Middle Section Axis) is shown in the figure, this is done to be referenced when the user needs to introduce the location of each isolator in the App.

For the given example, a first market search has been done. As many mounts can give accepted and similar vibration isolation values, the real difference lies in how the mounts act in the different directions of excitation. The intrinsic stiffness behaviours suitable for Genset configurations are listed:

- Primarily axial loading
- Axial and radial flexibility
- Axial and radial flexibility with anti-rebound

To clarify, these types of isolators are the first version and are recommended based on Paulstra Industry Genset vibration isolation guidance. In this case, the isolators needed may have different stiffness behaviours, which will be seen in the procedure section. As stated on pages before, there is a wide range of isolators that are differentiated mainly by how they are built and the mechanism that utilizes to isolate vibrations, therefore each of them has different applications and properties. This mount function allocation process will have to be done by the user before analysing the system.

4.1 Elastomeric mounts

Elastomeric mounts are the most commonly used vibration isolator in engine mounts systems (Vahdati and Saunders, 2002), this is reflected in the given example where these types of mounts are implemented. Engel (2013) states that there is a current conflict between the damping and elasticity (noise-reduction) of this type of mount. Nonetheless, the elastomeric mounts are the most matured and therefore economic vibration isolator as it is widely used since the 1930s. These have been through an evolution process that ended up transforming the mount into a vibration isolator that performs a proper function in all directions as detailed by Alkhatib (2013). This can be reflected in Table 2, where the elastomeric mounts stand out among the others due to their high stiffness ratios.

Every mount model is different, even if it is denominated in the same category. As there are a large number of models in the market, it is not possible to generalize and recommend one type of mount over another. Nevertheless, the parameters involved in every type of mount coincide, being more appropriate to guide the user by the utilization of the general vibration isolator parameters, as damping and stiffness. If the customer needs to isolate low vibration at low frequency, a high damping or stiffness rate would fit his requirements. Conversely, the vibration isolator for that customer will not suit a high-frequency system, where the mount functioning might be insufficient. Alkhatib also mentioned that if the specified system might work in a wider range of frequencies, low damping mounts are recommended, this fact is reflected in Figure 9.

Conversely, the stiffer the mount is, the more efficient the loads, especially high, are handled. It is infeasible to generalize between types of isolators, however, the works conditions of each differ from the others, as the material used in their construction is different. For elastomeric mounts, it is needed to be under normal type of conditions due to its material: rubber, which means that the environment must be under 70°C and there should not be corrosive liquids or gases. Another property of elastomeric mounts is that as they are loaded during large periods of time, creeping and permanent deformation are unavoidable. Usual values of the creep rate of rubber mounts can be as much as 25% as stated by Mecanocaucho (2021). The environment customer conditions are a crucial factor to consider when deciding the type of isolator, as the performance of it is directly related to this factor.

4.2 Steel mounts

These vibration isolators, due to be built in steel, have considerable mechanical acceptance to high loads, and also when shock appears in the system, the damping characteristics of it will help tremendously to the safety and workability of the engine. As explained by Cfm-schiller (2015), the steel spring vibration isolators are commonly used when the disturbance frequency is above 3 Hz, and up to 60 Hz as specified by Lassfolk (2018). However, they also state that in a system where impulse excitations can be found, steel mounts have a great ability to withstand shocks.

The most important part of a steel mount is called a steel cushion. These are made from woven stainless steel compressed into a geometrical object. This element from the mount is responsible for the damping characteristics of the vibration isolator as stated in Metallic-damper (2021). Compared with the elastomeric mounts, the steel mounts are unaffected by corrosion due to the thermal treatments on the steel which they are made of, also they provide stability characteristics maintaining the height under load constant over time, while in other types of mounts the height might reduce due to constant deformations. Another advantage of this type of isolator is that it can be easily produced, which makes them economical affordable. The environment temperature range of the steel mounts is very wide, as these isolators are able to withstand temperatures from -70°C to 300°C as specified by Paulstra Industry (2019). The steel spring mounts are recommended to the customer either if the environment is not suitable for elastomeric mounts, or if the stresses are out of the rubber mount range.

Rubber mounts are not able to isolate loads as heavy as the steel spring isolators, thus the implementation of the type of mounts this section focus on is mainly in heavy constructions. In addition, their natural frequency oscillates between 1 and 6 Hz (Wasberg, 2011), in the function of their static deflection. This fact restricts the general implementation of this type of mount, as it gets restricted by the disturbance frequency of the system. As a meant of comparison, the elastomeric natural frequency range goes from 6 to 20 Hz (Mecanocaucho, 2021), making them more adaptable to different disturbance frequency ranges.

4.3 Air springs

Air springs, or also denominated air bellows, are made from elastomeric reinforced material. These are simple action pneumatic actuators, which can deliver different stiffness properties by changing the pressure inside the isolator. This property is called system reveling and makes the isolator adapt to various load capacities by altering the pressure within the springs. Gavriloski et al. (2014) asserted that air springs are suitable to a system where the suspension is a key factor to consider, meaning that the load transmission will be softer than in other types of isolators. Gavriloski also states that this type of isolator is very interesting in frequency domains from 0 to 20 Hz. Sun et al. (2017) stated that these types of isolators are generally used in commercial vehicles, nevertheless, in more recent times their implementation in higher classes of vehicles is starting to grow. This fact is reflected in the selection of these types of isolators in Novia student's thesis, as seen in the Långbacka (2012) work.

To be able to determine which type of air below might be needed in a system, an introduction of the categorization and composition of these isolators needs to be made. Each bellow, in its ends connected to bead rings that contain air inlet ports and mounting spaces. These rings are built in steel, and it has the capability to be crimped or removed from the isolator. These are usually categorized by the output stroke, the higher the stroke is the less force can the isolator withstand. Some examples could be rolling sleeves, triple convoluted or single convoluted mounts, organized from the highest to the lowest stroke. The air is compressed inside rubber bellows, which does not support the isolation their only function is to maintain the gas trapped. The layering of the elastomeric part of the air below is specified in Figure 11.

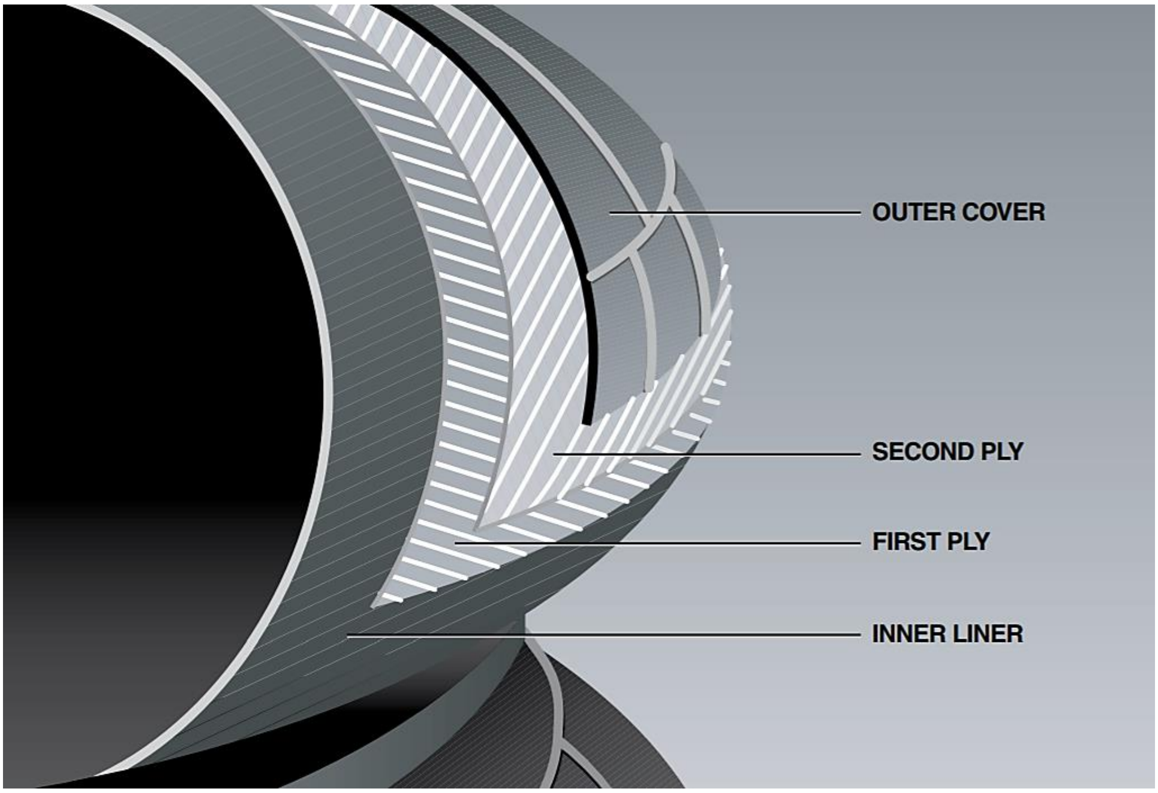


Figure 11. Air spring material layering (Firestone, 2021)

The Firestone (2021) air springs catalogue states that the load range of each air spring is from 0,45 to 575 kN. This type of isolator is a good alternative to the isolation of low frequencies if the steel spring mounts do not suit the system. This is since the isolator can withstand system disturbance frequencies from 1 Hz, specified also by Firestone. Also, this type of mount is the most efficient when there are constant load changes in the system, as its natural frequency is mostly constant during the variation of stresses. If the customer desires to expand its knowledge on this new type of isolators, the Firestone (2021) air spring manual that can be found in the References chapter digs deeper into this topic.

4.4 Mount disposition

As there are six degrees of freedom in the modelling of the engine mount system, one mode per DOF will be generated for the engine inertia being three rotational and three translational. In this work, each vibration mode will be treated separately, just as if it were an one degree of freedom system, called axis decoupling. Nevertheless, there is always parametric relation between every mode. In order to a correct decoupling, Timpner (1965) stated that the centre of gravity of the engine had to coincide with the elastic centre of the mounts. Being the elastic centre a point of an axis in where if a force is applied, the body would move entirely as a translative movement without rotation. Adhau and Kumar (2013) stated that this point is always below the intersection of the vibration isolator. If this assumption was real the mounts must be installed inside the volume of the engine, which is not a possible solution.

Nevertheless, it is possible to achieve a simultaneous location of both centre of gravity of the engine and the centre of elasticity of the mounts without having to do infeasible implementations. When the alignment of these centres is achieved, the assumed error decrease. However, the mount disposition designed should have been defined before the application analysis. This aspect is not inside the scope of the thesis, nevertheless, the information given before should help the user if that is the case. What is inside the scope is the assumption regarding the decoupling topic, in the analysis, it is assumed that as vibration is a low-path movement, the movement in one mode will not affect other modes. And therefore, the main and only analysed movement will be the one in the direction of the mode, nonetheless in reality a small movement in other modes will happen.

4.5 Practical considerations in vibration isolators systems

A general engine is able to vary its speed, calculations, as the firing frequency orders, need to be made with the lowest speed of the engine as it is the most restrictive one due to the resonance effect seen in Figure 9 (lower natural frequency leads the system to the amplification zone). Also, when turning on the engine it is known that while speed on it is increasing, there is a certain time where the system is in the amplification zone, and thus in resonance. The customer, by performing the first analysis will have to find which is the resonance border speed, as in the application results section the speed of the engine can be modified by the user. Once this speed is known, it is needed to ensure the passage through lower speeds is at utmost quickness and that the system is adequately damped to bear the maximum force transmitted.

Besides, it is assumed that the system is only connected with its surrounding by the mounts, nevertheless, in real-life problems, the engine is connected to other components, as an alternator in a Genset configuration. The connection between the engine and other external components must be by the utilization of flexible linkages, as can be seen in the top left picture of Figure 1, where the shaft is connected with an alternator through an elastic connector. For example, in the Genset configuration stated before, the engine is also coupled with the alternator, thus the coupling of the engine with the alternator should be flexible to prevent damage to the components of the system.

4.5.1 Shock

When the engine is turned on, or turned off, for a certain period the mounts are under an impulsive excitation. This will happen due to the changing of the speed of the engine because of positive or negative acceleration, and consequently, the mounts are subjected to a force. For the good functioning of the mount, it is important to compare the natural frequency of the equipment in contrast to the length of the period in which this phenomenon is applied due to the certain reaction-time of each vibration isolator.

4.5.2 Severity of vibration

It is important to determine and benchmark the specific amount of allowed vibration for each system. This will only apply if the system is located in the vibration isolation zone (see Figure 9). In this zone, the degree of vibration isolation will be greater than zero, and thus the vibration isolators will reduce the transmitted force by that value. However, determining how high will the degree of isolation needs to be is very important, this fact depends on the engine application, and it is a key factor when deciding which isolator will be implemented in economic and safety-wise thinking. This is standardized in ISO 12372 (10816) to help the user to identify the severity grade desired in the function of the application. This standard is based on the vibration velocity or other parameters, that need to be calculated or estimated by the user using proper equipment, the size of the engine, and the type of foundation. The range of vibration, type of machine, and foundation analysis are shown in Table 1.

Table 1. Range of vibration severity for different classes of vibration machines (ISO 12372)

<table>
<tr><th colspan="2">Range of Vibration Severity</th><th colspan="2">Maximum Values</th><th colspan="4" rowspan="2">Class of Vibration of Machine</th></tr>
<tr><th rowspan="2">Range Classification</th><th rowspan="2">Effective Velocity: RMS (mm/s)</th><th rowspan="2">Vibration Velocity (mm/s)</th><th rowspan="2">Vibration Displacement (µm)</th></tr>
<tr><th>Class I</th><th>Class II</th><th>Class III</th><th>Class IV</th></tr>
<tr><td>0.28</td><td>0.28</td><td>0.4</td><td>1.25</td><td rowspan="3">Good</td><td rowspan="4">Good</td><td rowspan="5">Good</td><td rowspan="6">Good</td></tr>
<tr><td>0.45</td><td>0.45</td><td>0.63</td><td>2</td></tr>
<tr><td>0.71</td><td>0.71</td><td>1.0</td><td>3.15</td></tr>
<tr><td>1.12</td><td>1.12</td><td>1.6</td><td>5</td><td rowspan="2">Acceptable / Allowable</td></tr>
<tr><td>1.8</td><td>1.8</td><td>2.5</td><td>8</td><td rowspan="2">Acceptable / Allowable</td></tr>
<tr><td>2.8</td><td>2.8</td><td>4.0</td><td>12.5</td><td rowspan="2">Improvement Required</td><td rowspan="2">Acceptable / Allowable</td></tr>
<tr><td>4.5</td><td>4.5</td><td>6.3</td><td>20</td><td rowspan="2">Improvement Required</td><td rowspan="2">Acceptable / Allowable</td></tr>
<tr><td>7.1</td><td>7.1</td><td>10</td><td>31.5</td><td rowspan="6">Not Acceptable</td><td rowspan="2">Improvement Required</td></tr>
<tr><td>11.2</td><td>11.2</td><td>16</td><td>50</td><td rowspan="5">Not Acceptable</td><td rowspan="2">Improvement Required</td></tr>
<tr><td>18.0</td><td>18</td><td>25</td><td>80</td><td rowspan="4">Not Acceptable</td></tr>
<tr><td>28.0</td><td>28</td><td>40</td><td>125</td><td rowspan="3">Not Acceptable</td></tr>
<tr><td>45.0</td><td>45</td><td>63</td><td>200</td></tr>
<tr><td>71.0</td><td></td><td></td><td></td></tr>
</table>

As it can be seen, once the vibration velocity, vibration displacement, or effective velocity is defined, it is possible to classify the vibration machine among a range of vibration severity. Once the vibration severity is known, an identification of the class of the machine needs to be done. To summarize, Ugechi et al. (2009) categorizes the classes by their size and output power. Class I is defined as small machinery, and it can have an output power of up to 15 kW. In the second class (Class II), medium-size machinery is identified with a power output range from 15to 75 kW without special foundations, however, if the machinery is mounted in rigid or special foundations the output power can go up to 300kW. In the Class II machinery, large rotating machines are found especially mounted on stiff foundations, which can be electric motors that have an output power of 300 kW or above. And last, the Class IV machinery is defined as large machinery that is flexible in the vibration measurement direction, an example that could fit into this class could be a turbomachine.

As an example, Örn (2014) commented that in Wärstila engines the overall vibration values must be kept below the 18 severity grade. This could be used as a reference for the user, Örn also stated that the typical vibration velocity values can go from 10 mm/s to 15 mm/s in engine blocks and common base frame engines which means a range of RMS from 3 to 200 Hz.

5 Application guidelines

In this section, an introduction of the program used to develop FreqNess will be done, Moreover, the procedure, assumptions, justifications, advice, and general explanations of some parts of the application will be also introduced.

5.1 MATLAB App Designer

For the purposes of this thesis, App Designer, which is a feature of MATLAB, allows MATLAB programmers to develop desktop or web applications. It is based on MATLAB language, and it uses a graphical interface (GUI) to integrate functionalities as buttons, plots, tables by simply dragging them to the editing application space. MATLAB is a computing environment built by MathWorks, which is mainly based on a matrix and vector analysis. It is available for 119 euros annually for personal use, however, there are other pricing depending on the software applications (Mathworks, 2021). FreqNess was developed in the R2021a MATLAB release. The author of this work selected this software due to two main reasons. The first one, and the most relevant, is that the programming language was already known by him, hence making it easier to implement functions and perform calculations. The second one is that most of the engineering background people know about MATLAB, as the number of users that utilize it is more than 4 million (Mathworks, 2017). To conclude, FreqNess can be easily installed by the customers via the MATLAB webpage.

5.2 Procedure justifications and assumptions

The stage that will be analysed by the application is called static force analysis. First, it is needed to establish the reference system used in the App, which is specified in Figure 12.

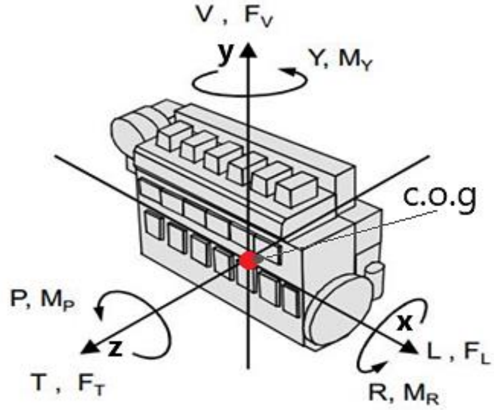


Figure 12. Mode forces and directions of an engine (Edited From Wärstila, n.d.)

The three-dimensional space is defined by the three-axis (x,y,z). These axes can also be called longitudinal, vertical, and transversal axis, respectively. The forces that have the same direction as the axis are nominated as F_L, F_v, F_T, and the torques are specified as M_R, M_Y, M_P where the subindex stans for rolling, yawing, and pitching. This will be the reference system used in the whole work, and it will be referenced in the next sections. As it can be seen in Figure 12, these axes can be easily recognized by only identify the shaft of the engine, parallel to it, and crossing the centre of gravity the longitudinal axis is placed. Once the longitudinal axis is identified, the other two are perfectly defined as the transversal axis is orthogonal to the longitudinal and in the same vertical coordinate.

After the system of reference is well-defined, then the procedure followed to face the problem will be introduced. This methodology follows the superposition rule, analysing every mode independent from the other ones, as briefly introduced before in the Mount disposition subsection. The independence of the modes carries a frequency output error from 0 to 2 Hz. This error is assumed to be acceptable, and it was measured by comparing results with Rintanen (personal communication, 15.05.2021). This approach studies every mode by considering that the most relevant disturbance frequency coincides with the speed of the engine. Also, for the stiffness analysis, in each mode is considered that the four mounts can be simplified into one DOF problem seen in Figure 13. As the mounts are placed in parallel, the stiffness of the equivalent vibration isolator is stated in (5.1).

$$k_{eq,j} = \sum k_{i,j} \tag{5.1}$$

Where $k_{eq,j}$ is the equivalent stiffness in the j mode, and $k_{i,j}$ is the particular stiffness of the mount i, in the mode j. As a result, FreqNess will calculate one stiffness per the mode of the system by using eq 5.2, hence six different stiffnesses will be determined. Consequently, applying eq 5.3 and 5.4, the different natural frequencies for each mode are evaluated. The programming procedure to perform what has been stated before is shown in Appendix 1: Programming analysis implementation. Also, the damping coefficient is assumed constant in the space, this supposition was done by the lead supervisor. However, each type of mount has a different damping coefficient, as stated in the Vibration isolators section. The assumed damping coefficient values can be seen in Appendix 2: Table of assumed values.

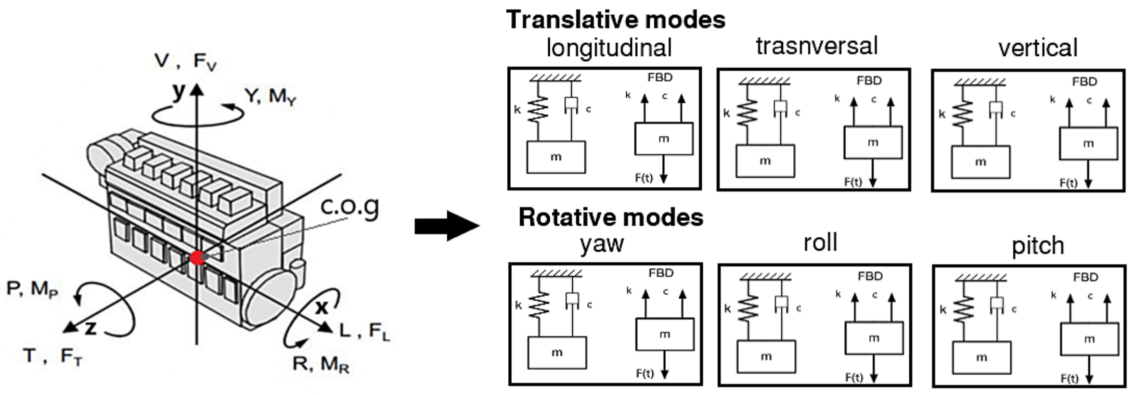


Figure 13. Superposition procedure (Edited from Wärstila, n.d. and Leancrew, 2014)

It is important to remark that without the *modus operandi* followed in the next step, the rotative modes stiffnesses and frequencies and the translative frequencies would not be possible to be determined. As for the calculations in the rotative modes the position and vertical stiffness are required, and for the translative ones the different axes stiffnesses are needed. In Figure 14, the user is able to let the program know the position of the mount referring to the established reference system in Figure 12.

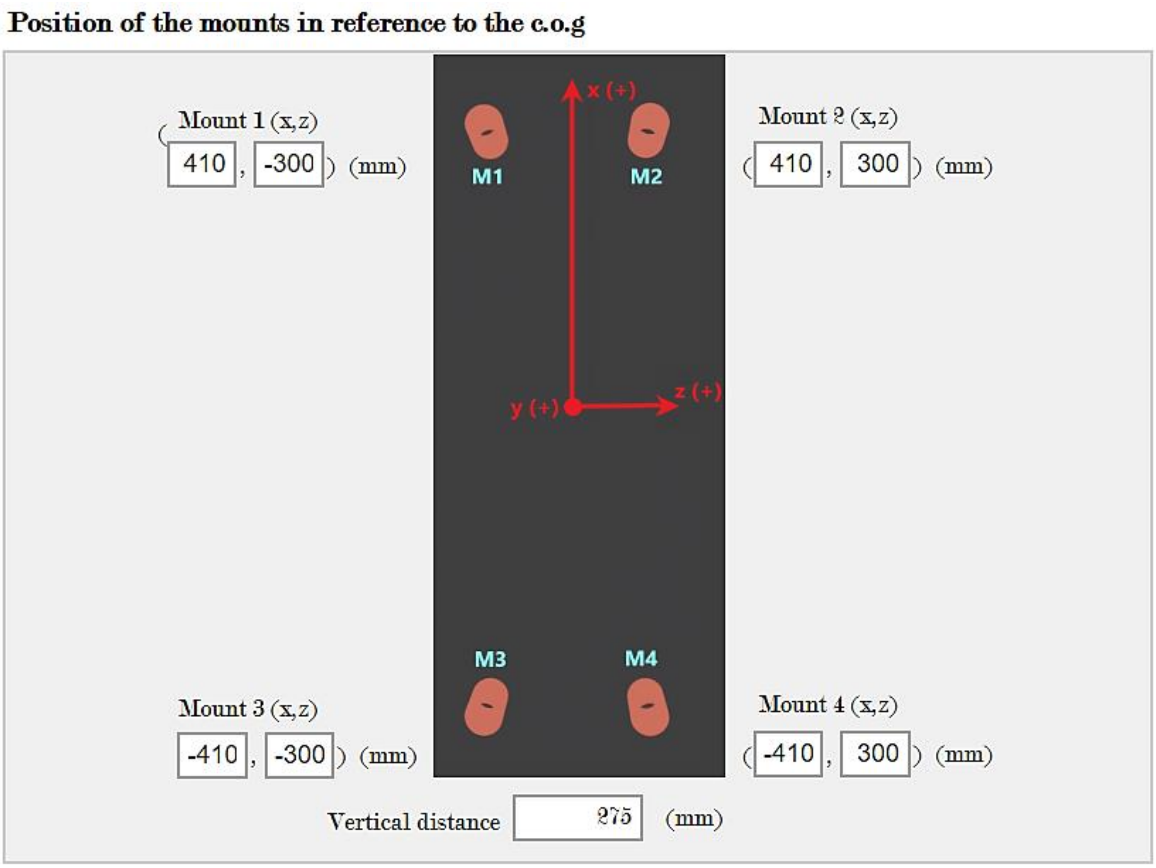


Figure 14. FreqNess mounting disposition section design

As it can be seen, each mount is placed in the analysed space based on the centre of gravity and the axis stated before, the c.o.g can be known by contacting the manufacturer or by doing an approximation. It is assumed that every mount has the same vertical coordinate, and therefore is located at the same height level which is a very typical case as stated by Zhang et al, (2014). Hence, there is only the need to identify the height of one mount, Nevertheless, in the longitudinal and transversal axis, each mount must have different coordinates, in every longitude input data it must be measured in millimetres. Also, the point of measurement of each mount is defined in Figure 10 as the Middle Section Axis (MSA), a vertical axis that constrains the longitudinal and transversal coordinates.

Continuing, in the vibration isolator properties section of the App, the static stiffness of the mount is asked to the user to be introduced. These values can be found in the specification catalogue of the mount. The procedure to obtain the static stiffness will be stated later in the procedure section. The application design for this section can be seen in Figure 15.

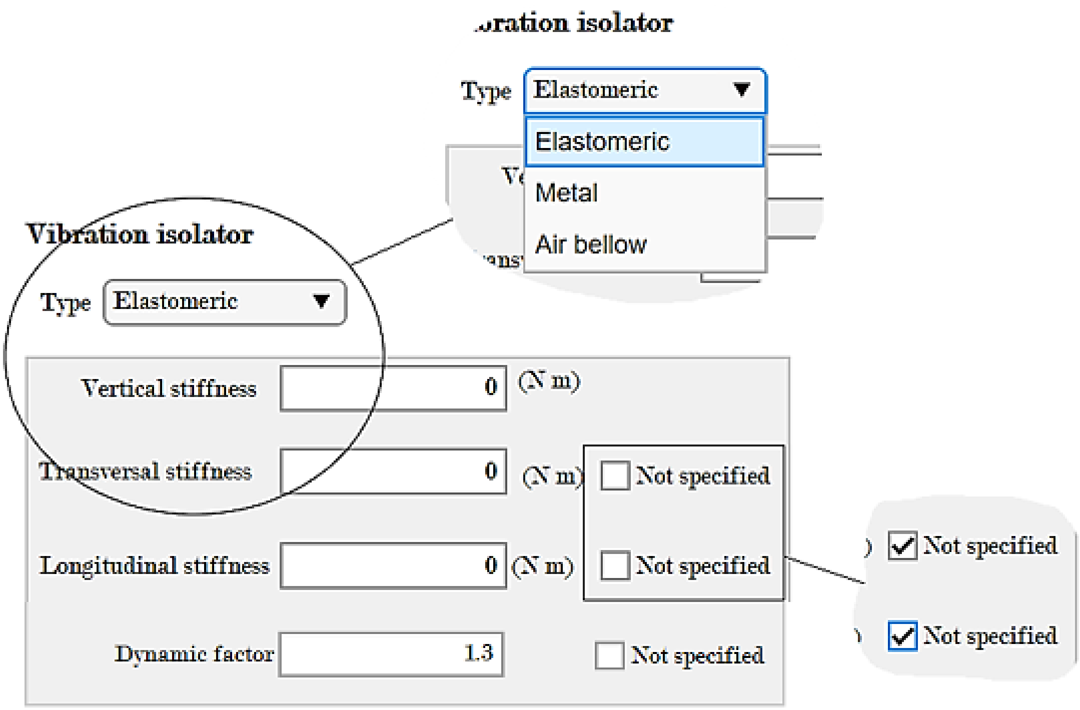


Figure 15. FreqNess vibration isolator properties section

In this section, the user is able to specify the type of isolator that will be implemented in the system between the elastomeric, metal spring, and air below options as is seen in the top part of Figure 15. Also, a global dynamic factor can be introduced for the calculation. In the supposed case where this value is not given, then the customer must check the box on the right. This will be assumed, stated in Appendix 2: Table of assumed values, and approved by the lead supervisor.

As advice for the user, this dynamic factor depends on the hardness of the mount, it is recommended to contact the manufacturer in case this information is not given in the catalogue. The values need to be introduced by the user in SI units, as in this section of the application talks about stiffness the units must be N/m. In this stage of the input data tab, there is the option to mark as not specified regarding the transversal and longitudinal stiffness. These values usually are not given by the manufacturers, see Paulstra Industry (2019) or catalogues. These characteristics usually need to be calculated by the user by using the geometrical characteristics of the mount. However, in some mount catalogues such as the Trellcan (2021) where some mounts an axial to radial or transversal stiffness ratio is given. If this is the case the user can specify the exact transversal or longitudinal stiffness, if not an approximation for every type of mount is done. The approximate stiffness ratios were settled after an email discussion with Rintanen (personal communication, 15.05.2021) and are shown in Table 2, and in Appendix 2: Table of assumed values. To complement this approach, in Figure 15 located at the bottom right part, the checked boxes of "not specified" are shown.

When every needed parameter is correctly introduced, the program will perform one last calculation, this one is to calculate the most relevant parameter in the App, the degree of vibration isolation. As stated before, as there is one frequency per mode, there will also be a degree of isolation per mode. This parameter, once the calculation is performed, will only be a function of the disturbance frequency, as formulated in eq (3.15). The disturbance frequency can be modified by the customer either moving the slider (see Figure 21) up or down or by typing the exact number. This function will be introduced later on in Section 7, however, an example of the procedure applied for the calculation of the degree of isolation is showed in Appendix 1.4.

6 Input data procedure

This section will help the reader to understand how the software should be used. Also, where to find crucial information about engines and mounts that might need to be introduced in the App. Engine, mount, and mount disposition sections of FreqNess will be introduced while the input data is collected. Examples, approximations, and assumptions that the customer can perform are also introduced. In this section, specific parts of the application will be illustrated as figures however, in Appendix 3: FreqNess screenshots complete captures of the application can be found.

6.1 Engine information

The engine information needed to perform the analysis is, in general, the inertia and rotational inertia of the body. More precisely, the mass and the moment of inertia in all three axes. The design to introduce these properties of the engine can be seen in Figure 16.

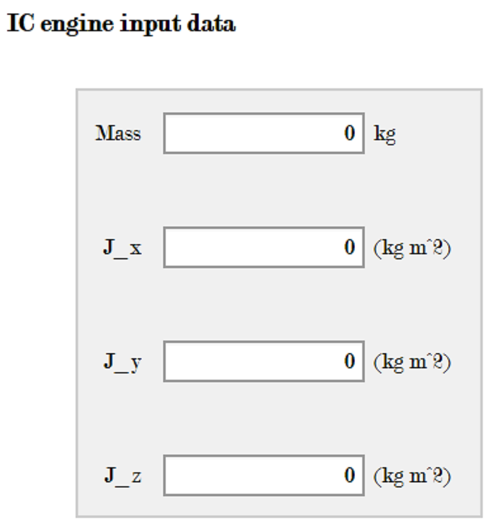


Figure 16. FreqNess engine properties section

The mass, expressed in kilograms, can be easily found in the first pages of the engine specification document as is shown in Figure 17.

SisuDiesel — Engine Specifications — 0–3

ENGINE SPECIFICATIONS

Study engine: 420DWI

Engine type	320D	320DS	420D	420DS	420DW	420DWI	620D	620DS	634DS
Number of cylinders	3	3	4	4	4	4	6	6	6
Displacement (dm^3)	3,3	3,3	4,4	4,4	4,4	4,4	6,6	6,6	7,4
Cylinder bore (mm)	108	108	108	108	108	108	108	108	108
Stroke (mm)	120	120	120	120	120	120	120	120	134
Compression ratio				16,5/18,5:1					
Combustion				direct injection					
Firing order	1–2–3			1–2–4–3				1–5–3–6–2–4	
Compression pressure bar[1]				24					
Weight kg[2]	275	280	335	340	340	345	500	510	515
Direction of rotation from the engine front				clockwise					

[1]) Minimum value at operating temperature and starting revs. Max permitted difference between cylinders 3,0 bar.
[2]) Without flywheel and electrical equipment.

Figure 17. General engine specifications (Edited from Sisu Diesel, n.d.)

Highlighted in green the mass of the given engine example is found. The program will use this mass for the vertical, longitudinal, and transversal frequency calculation by using eq (3.3) and showed as programming code in Appendix 1.1. The stiffness parameter of the stated equation will be introduced later in the mount input data subsection. Once the blank of the mass of the engine is filled, the user will have to find information about the moment of inertia of the engine if the desire is to know how stiff every mount is while rolling, pitching, and yawing. Needs to be calculated by the customer or contact the manufacturer to get more information about the engine. Nevertheless, a first estimation can be done by assuming that the engine is a rectangular prism, illustrated in Figure 18.

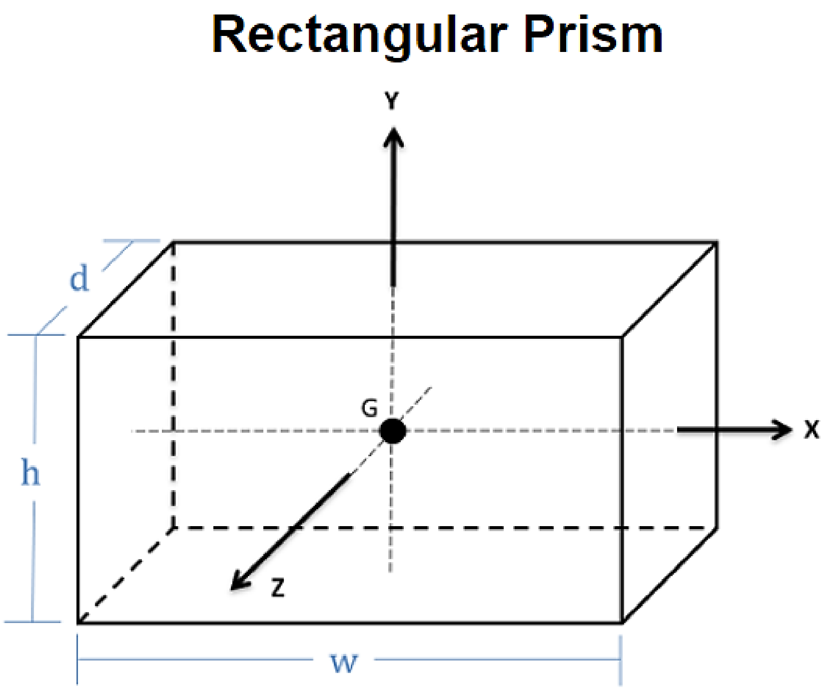


Figure 18. Rectangular prism system approximation (Psu.edu, 2021)

Once the general measurements of the figure are made (d, h, w see Figure 18), thus to calculate the moment of inertia in each axis the equations (6.1), (6.2), (6.3) need to be applied.

$$J_x = \frac{1}{12} \cdot m(h^2 + d^2) \tag{6.1}$$

$$J_y = \frac{1}{12} \cdot m(w^2 + d^2) \tag{6.2}$$

$$J_z = \frac{1}{12} \cdot m(h^2 + w^2) \tag{6.3}$$

In these equations, the desired units are $kg \cdot m^2$, therefore the weight must be calculated in kilograms and the longitudinal parameters in meters. Once the approximated values are known, then the program will have an approximation value of the moment of inertia and a rotational analysis will be performed. The frequency analysis will be based on it; however, this is done to have an estimation of what range of rotational frequencies the system will vary. Conversely, it is very recommended to contact the manufacturer or calculate each moment of inertia by using any 3-D physical analysis software, like SolidWorks or Creo Parametric. Appendix 4.1 shows a summary of this section measured values of the engine input FreqNess data in the 420 DWI engine.

6.2 Vibration isolator information

In this section, the user will need to fill in the type and stiffness of the vibration isolator used. This will be based on Figure 15, which is the design of the introduction of data for this subsection. First, it is needed to know which type it is between elastomeric, metal spring, and air below. This is very crucial to know by the program due to the last assumption stated in section 5.2, where a generalization in the stiffness ratios is done, the estimation procedure is illustrated in Appendix 1.1. Also, the type of isolator defines its average damping coefficient, discussed in a Microsoft Teams meeting with the lead supervisor. The assumed data is shown in the table in Appendix 2: Table of assumed values. Once the type of isolator is defined, then the vertical stiffness needs to be introduced by the user. Generally, the catalogues do not give the static stiffness as a specification, however by using Eq (3.1) and knowing the load that the mount is stressed with, it is possible to distinguish the vertical stiffness. The procedure to determine this parameter is done by the utilization of a graph given in the mount catalogue (see Figure 19). Assuming an arbitrary load in one mount of 80 kg, the deflection of an, also arbitrary, the mount can be found.

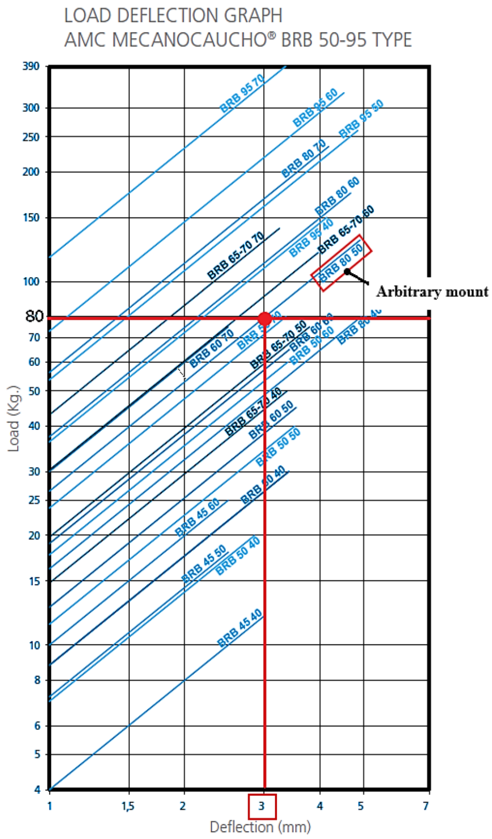


Figure 19. Arbitrary mount deflection of a specific load (edited from Mecanocaucho, 2021)

As seen in Figure 19, the deflection of a BRB 80 50 elastomeric mount under a stress of 80 kg is 3 mm. Once the deflection is defined, the stiffness can be calculated by following the next steps:

$$\Delta y = 3\,mm, F = load \cdot g = 784{,}80\ N$$

$$k_{vert} = \frac{F}{\Delta y} = \frac{784{,}8}{3 \cdot 10^{-3}} = 2{,}62 \cdot 10^{5}\ N/m$$

In this case, no horizontal or transversal stiffness ratios were given, so the user should check the “not specified” boxes. This will assume stiffness ratios for every type of isolator, the values were discussed with the lead supervisor and are shown in Table 2.

Table 2. Stiffness ratios of different vibration isolators

Type of isolator	Vertical to horizontal ratio	Vertical to transversal ratio
Elastomeric	1:0,70	1:0,50
Metal spring	1:0,60	1:0,40
Air bellow	1:0,50	1:0,30

In this table, it can be seen that the elastomeric type of mount is the one that is less rigid in the horizontal and transversal axis. This helps to transmit the forces, and therefore the vibration, more fluent than the air below type of isolator, which is less stiff in those directions. However, depending on the load scale, different stiffnesses might be needed. This will be discussed more deeply in the result analysis section.

6.3 Mount disposition information

This is a step where the user will have to know an idea of where the mounts should be installed beforehand. The disposition designing part of the mounts does not enter in the project scope, however, in subsection 4.4 crucial information and considerations in relation to this topic are given. However, as stated in Figure 14, and highlighted in the Procedure justifications and assumptions section, the coordinates of every mount are referred to as the center of gravity of the engine. And the point where to measure the distance on the mount is defined as the MSA intersection with the plane formatted with the horizontal and transversal axis (see Figure 10). As stated in the lasts sections, every parameter introduced by the user must follow the International Unit System, hence in this section of the application the values need to be introduced in millimeters, first the longitudinal value, and then the transversal value for every one of the four mounts (see Figure 14). The rotational calculations performed by FreqNess rely mainly on the position of the isolator, this can be seen in Appendix 1.2 and Appendix 1.3. Where first, in Appendix 1.2 every mode rotational stiffness is calculated, in the appendix, the calculation is only performed in one mount. Nevertheless, Appendix 1.2 shows a guidance example, this calculation is performed for each mount, as each mount can have different coordinates. Where in Appendix 1.3 the equivalent system rotational stiffness is calculated, and thus every mode rotational frequency. When every parameter is introduced correctly into the program, the user should click the "Calculate" button located in the bottom part of Figure 20.

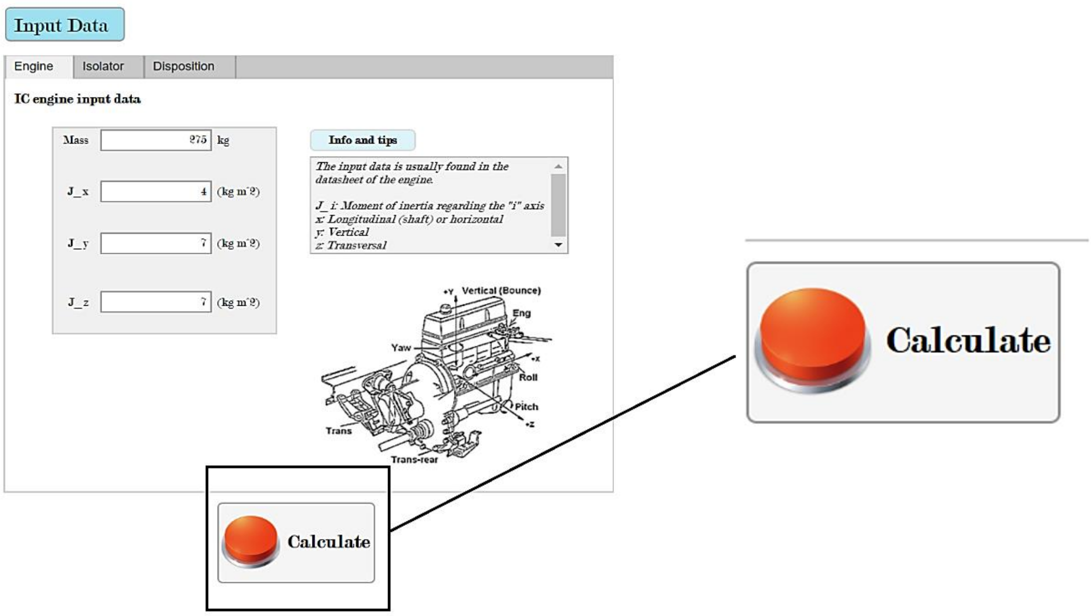


Figure 20. Screenshot of the input tab of FreqNess

This button will perform the calculations stated before and, it will show the results to the user in a simplified form, hence the user can decide whether implement or discard the specified amount. The result analysis will be performed in the following sections.

7 Results

In this section, the result tab of FreqNess will be explained, so the user can interpret the result values of the analysis. The result section tab can be divided into three main components: The amplification, resonance, and vibration isolation graph and the engine speed slider function. In addition, at the end of the section, a step-by-step analysis of the Sisuengine 420 DWI will be performed, with the aim to guide the user through the App.

7.1 Engine speed slider

This function in the result section was implemented with the objective to let the user comprehend the vibration behaviours of the engine in the function of the most restrictive disturbance frequency of the system: the engine speed. The slider starts with a value of 500 rpm as can be seen in Figure 21.

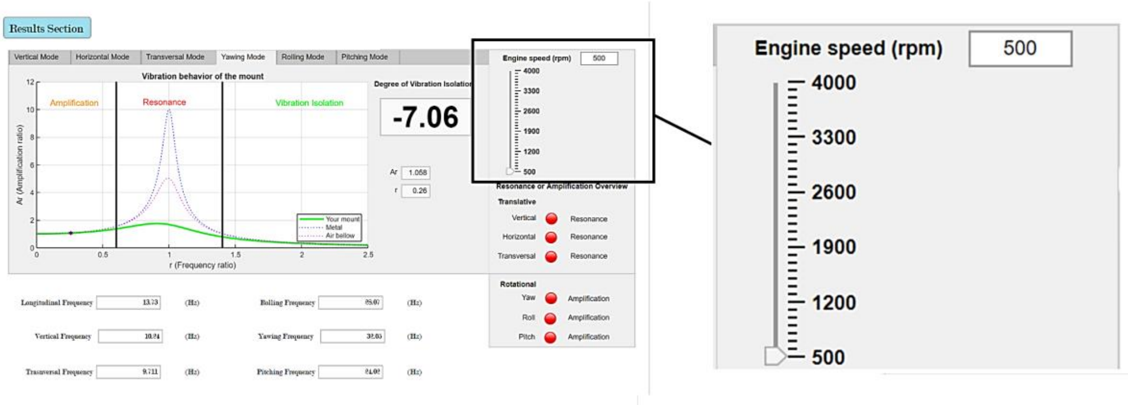


Figure 21. Engine speed slider

The engine speed can vary between a range of 500 to 4000 rpm. This range of speed was selected based on general frequency behaviours of 4-stroke engines, as stated in subsection 2.1.3. The functionalities of the slider are to discern in which disturbance speed the system goes through the resonance-vibration isolation borderline and to distinguish which engine speed adequate to the customer vibration isolation requirements. By identifying the first stated functionality, and as stated in subsection 4.5, it is recommended to cross the defined vibration borderline, hence avoiding resonance to occur. It is important to remark that neither the natural frequencies nor stiffnesses of the mounts are not influenced by the disturbance frequency of the engine. This parameter will only impact the system's vibration behaviour.

7.2 Degree of vibration isolation results

The main parameter on which this subsection will be focused is the degree of vibration isolation. Nevertheless, in the results tab other numerical and graphical parts will be also introduced. These, are additional tools to help the customer to understand the analysis while focusing on the main analysed parameter. The main component of the results tab is shown in Figure 22.

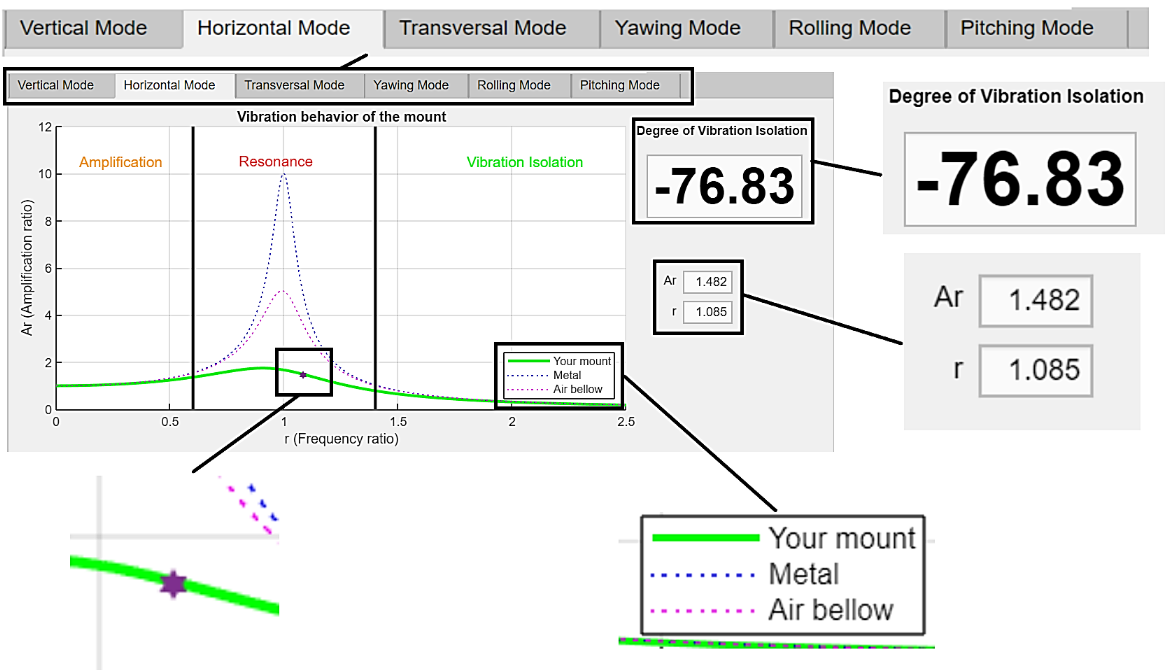


Figure 22. System vibration isolation result performance

On the top part of the figure, every mode is represented by a tab. In each tab the results are shown using the same structure however, each component is calculated separately. As it can be seen, the vibration behaviour graph compares the amplification ratio (Ar) in front of the frequency ratio (r), whose values are printed in the middle-right part of the tab. In this plot, various curves are illustrated. These, compare the amplification comportment mainly in the resonance zone, as in the other two zones the curves overlap each other, of the three types of mounts, as seen in the legends on the bottom-right part of the graph. These legends are also programmed to let the user know which type of mount he has chosen, and then plot it with a wider line, the other two types are plotted in dotted lines. The vertical black lines indicate the borderlines between the settled vibration behaviours.

As in this work, resonance is a crucial factor to consider the general vibration amplification zone illustrated in Figure 9 has been divided into a normal amplification zone called

"Amplification" and a high amplification zone called "Resonance". The decision of the borderline values was done by the lead supervisor, in where the vibration isolation zone begins when an acceptable degree of isolation value for engines is acquired, and not at a null degree of vibration isolation. Also, the damping ratios statements detailed in section 4 are reflected in this graph. Nevertheless, the main objective of this graph, and as affirmed before, is to locate the behaviour of the system between the amplification, resonance, and vibration isolation zones. This is done by plotting a purple pentagram, which is seen in the top-left part of the figure, that tells the customer in which vibration zone the system is found. This pentagram will move to the right if the user speeds up the engine by using the slider, or to the left if contrary. This function is very useful to determine the vibration borderlines, as will be seen in the next subsection. Moreover, in every tab, a Blender animation of the engine movement in every mode has been implemented. The aim of these animations is to link the real-life vibration movement to the graph, and the degree of isolation parameters. Lastly, the degree of vibration isolation is printed in the top-right part of the interface. This value will vary as the user modify the engine speed, the program is settled to recalculate this value every time the speed is adjusted. As a way to facilitate and give more information to the customer, the components in Figure 23 were implemented.

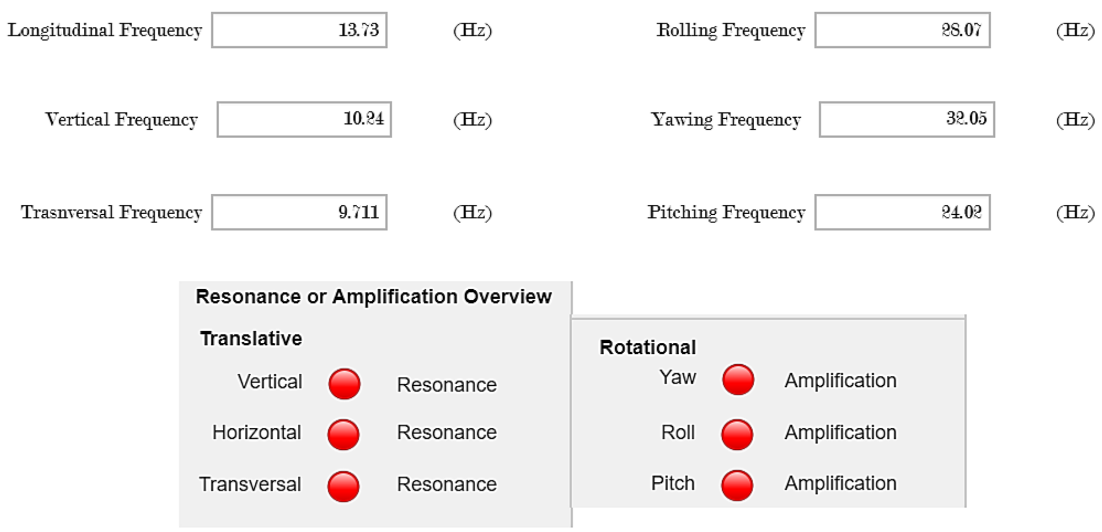


Figure 23. Additional graphical tools for the customer

The information collaged on the top part of the figure summarizes the system natural frequencies in each mode. If the user has an idea of what these frequencies are, then it is easier to select the proper engine mount that suits the frequency range established by FreqNess. On the other side, at the bottom of the figure, a general overview of the frequency behaviour is done. This will print by means of a lamp in which zone of vibration every mode is. Hence, the customer will be aware of the weak modes and act in consequence. To sum and process what has been stated before, subsection 7.3 will give a step-by-step analysis of how to run FreqNess.

7.3 Sisu engine analysis

In this subsection, the example engine (Sisuengine 420DWI) will be analysed by FreqNess. This process will consist of three main steps, the first one being the input data identification, then followed by the introduction of that information into Freqness, and finally, the result reflections. Hence, analysing the given case, and reinforcing the customer guidance with a real example.

In the engine input data introduction, the mass of the engine has been known in the example given in subsection 6.1 (see Figure 17). Consequently, leaving behind the moment of inertia parameters of the engine. To be able to find these values, the estimation explained in subsection 6.1, assuming that the engine is a rectangular prism (see Figure 18), will be performed. First, it is needed to measure the engine general d, h, w lengths. Values of the dimensions for the example engine are seen in Figure 24.

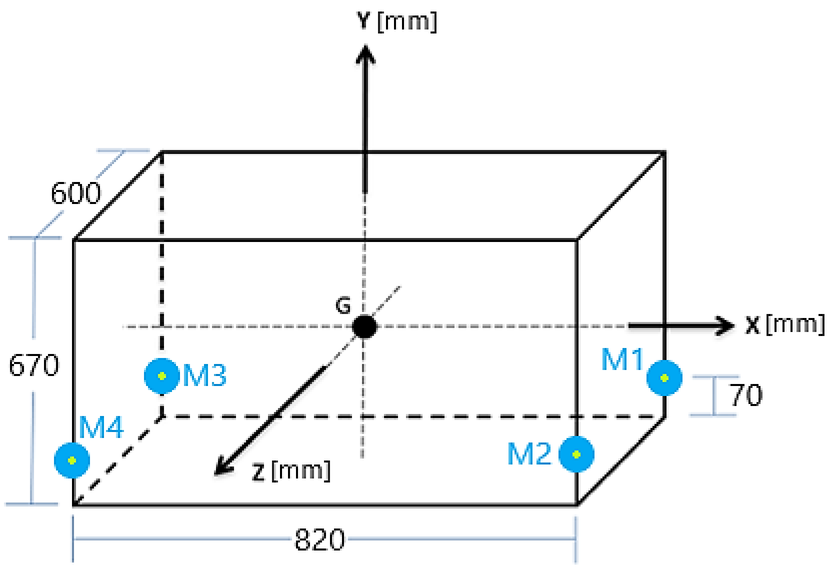
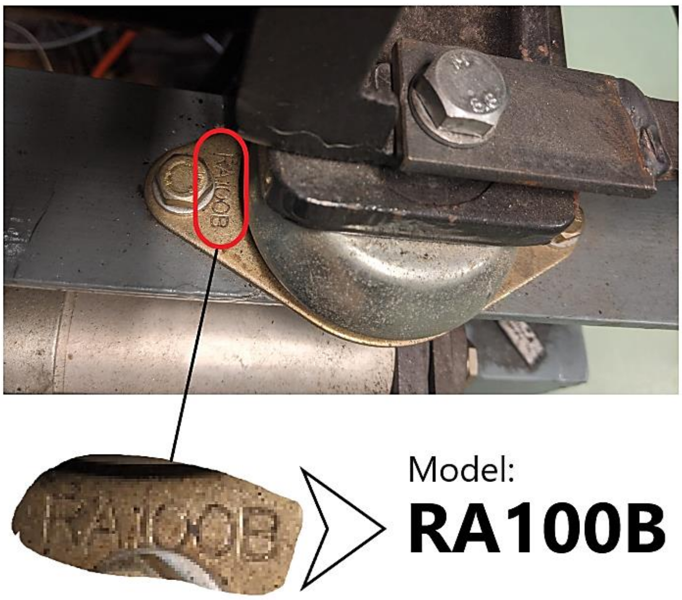


Figure 24. Approximate volume for the example engine (Adapted from Psu.edu, 2021)

In the figure, the disposition of the mounts in the x-z plane can also be seen. In addition, the measure from the bottom of the engine until the point of measure is illustrated, as the distance from the point of measurement of the mounts (M1, M2, M3, and M4) to the centre of gravity of the engine is a value that is introduced in this FreqNess section. Consequently, eq (6.1), (6.2) and, (6.3) need to be applied. Knowing that the m parameter coincides with the mass found before in the explanation, the moments of inertia in the three axes for the example engine are:

$$J_x = 23{,}26\ kg \cdot m^2, J_y = 29{,}68\ kg \cdot m^2, J_z = 32{,}24\ kg \cdot m^2$$

After the engine values identification, the following is step is the mount specification recognition. In the laboratory, there was barely any information about the implemented elastomeric mounts. Nevertheless, the vibration isolator specifications could be easily found by following the next phases. First, as seen in Figure 25, the model of the elastomeric was found.



Type
RA 100/M10
RA 100/M12
RA 200/M10
RA 200/M12
RA 350/M12
RA 350/M16
RA 500
RA 800
RA 1200
RA 1800

Figure 25. Identification of the example case mounts and Ra & Fail Safe EF types (Wilrep.com, 2021)

As it can be seen in the illustration, there are ten different types of the isolator model. To be able to discern between them, the isolator nomenclature was the determinant factor to know which of those is implemented in the engine. Nevertheless, still, there were two types of isolators with a similar nomenclature (the first two in Figure 25.b). By measuring the width of the screw, it was possible to determine that is an M10 and consequently, the RA100/M10 was found to be the implemented elastomeric mount. Once the identification of the mount is done, the next stage is to identify the load per mount. It is recommended to measure the distance shown in Figure 26 to determine if either the mounts are symmetrically mounted, which tends to be the most common case, or if not. As if the mounts are symmetrically implemented, then the measured distance in Figure 26 must be very similar.

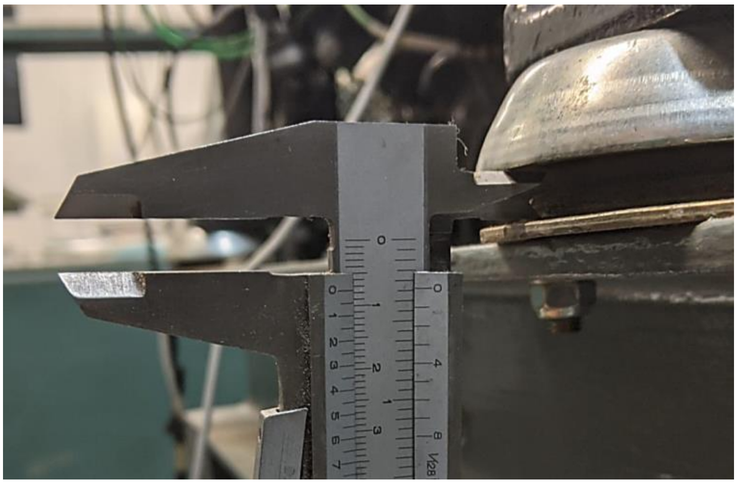


Figure 26. Methodology to verify a symmetrical mount disposition

This procedure was applied to each mount and, in conclusion, the distance of each was identical with an error that can be neglected. Thus, it can be stated that as this measured distance is identical for each mount, then it can be affirmed that the deflection of each mount is also identical. This statement leads to the conclusion that each mount is equally loaded, and therefore a symmetrical mount disposition can be assumed. The procedure to distinguish the vertical stiffness of the isolator is similar to the one made at the end of subsection 6.2. Nevertheless, the specific deflection procedure for the example engine can be seen in Appendix 4.. When the deflection is identified, by applying eq (3.1) it is possible to calculate the specific vertical stiffness of the mount. This is possible, as the load per mount is known and the deflection is distinguished. Concluding then, that the vertical stiffness of the vibration isolator has a value of $47{,}96\ kN/m$.

As the manufacturer do not give any information neither about the stiffness ratio in the other two axes or the dynamic factor of the mounts, the not specified boxes (see Figure 15) will be checked. For the disposition part, and to simplify the measurements done, it is assumed that the mounts are placed in the concerns of the rectangular prism. However, as seen in Figure 1, the isolators are not below the engine. This vertical difference will be considered. The input data screenshots from FreqNess can be found in Appendix X. However, a general vibration overview can be seen in Figure 27.

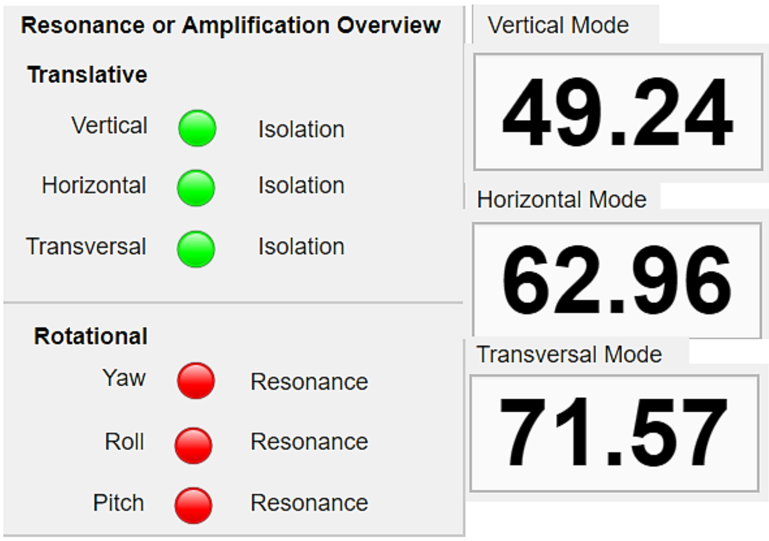


Figure 27. Sisu 420 DWI vibration overview

In the illustration, it is seen that at the lowest frequency (500 rpm), the vibration behaviour of the system differs between translative and rotative modes. In the translative, the mounts isolate the vibration forces nevertheless, on the right side of the figure the respective degrees of vibration isolation are shown. The values are not what is expected from an engine mount, isolating approximately half of the forces in the vertical mode. As stated in earlier sections, the customer requirements for vibration isolation were that at least 95% of the forces were isolated. However, the frequency (rpm) is variable, so there might be a certain frequency where the customer requirements are accomplished.

Moving to the translative modes, it can be seen that even having the system high moments of inertia, the resonance phenomenon is not avoided. As a recommendation by the author, and as told in the practical consideration sections, the engine must go through the 500 rpm to the 800 rpm (rotative isolation zone borderline) as fast as possible, due to resonance in the rotative modes. In a different type of mount, but with similar properties to the implemented one, the resonance would be a more dangerous topic, due to the damping factor of the other types of isolators as the amplitude of the movement would be a risk for the system. This statement is reflected in Figure 22. To conclude what has been stated before, the current mounts do not accomplish the user degree of isolation requirements and the system leads to low isolation in the rotative modes. Therefore, it is recommended to change the model of the implemented vibration isolators to another one that can meet the user's necessities.

8 Discussion

In this chapter, a discussion of the partial or full delivery of the initial aims and objectives will be made. In addition, other published work, results, limitations, consistencies, and inconsistencies will be also debated.

With respect to the established aim and objectives, it is considered that the most crucial goal has been achieved by the creation of a didactic and transparent App. This allows the user to compare, test, and analyse different vibration scenarios of a four-mount engine system by introducing general specification parameters. Hence, obtaining a complete analysis in every mode of vibration for a variable disturbance frequency, which is the final deliverable of this work. The application is considered to be transparent as every calculation, MATLAB programming procedure, assumptions, and instructions to run the application were fully explained in this paper, which leads to a user-friendly tool that can be used by anyone that has installed MATLAB.

Also, different examples and alternatives of the process of collecting the specification data either for the engine or the mounts were given. This thesis was structured as a manual for future users that desire to analyse their specific systems. As a consequence of this fact, animations, practical advice, and considerations were implemented to FreqNess with the previously stated aim of developing an interactive and intuitive App.

Regarding other published works, Lanbacka (2012), Lassfolk (2018), and Orn (2012) previous thesis gave the author a clear path to follow and helped to enlarge his knowledge about vibrations and engine mounts. Moreover, the work done by Ramachadan (2012) was crucial to the development of the understanding and development of the vibration theory chapter.

With respect to the results in the Sisuengine 420 DWI test that has been carried out in the previous chapter, remarkable data has been found. It has been demonstrated that analysis in every mode of vibration is needed, as the rotational modes are as important as the translative modes.

It has been found that all three rotative modes are resonant while in the translative modes there is poor isolation in a big range of disturbance frequency. Consequently, the user will need to reduce the system's natural frequency, so the frequency ratio is bigger, by modifying the isolator properties or increasing the disturbance frequency. Increasing this parameter connotes an increase in the engine speed, which as stated by Hanafi et al. (2015) leads to higher brake-specific fuel consumption, therefore, being the engine less economically efficient. Nevertheless, and as it will be introduced in the inconsistencies part of the discussions, a more specified moment of inertia approach would be greater to make as the approximation done might help the resonance result. However, the procedure is done to achieve the results it can be used as a manual for the customer, therefore, accomplishing the second stated objective.

Corresponding to the limitations of the thesis, the pandemic COVID-19 affected greatly the development of a more complex and specified App. Due to this pandemic, the meetings with the thesis supervisors were very limited, as a consequence of the constant lack of time due to the university work, they had been through. It is important to take into consideration that the whole work was based on books and articles theory, which the author tried to illustrate in the structure of this thesis, which is a summary of the collected knowledge to program the App.

With respect to the inconsistencies, the most relevant part that slowed down this work was the lack of experience in engines and vibration isolators, leading to an analysis that can be considered basic. However, this simple analysis helped to accomplish the objective to make a friendly-user App. Also, the author had to go through a learning process to use the tools: MATLAB App Designer and Blender. MATLAB App Designer was used to program and create the interface of FreqNess while Blender was used for the engine vibration animations. Another inconsistency would be the needing for MATLAB, a paid software, to be able to run FreqNess. A further work of this thesis would be the code conversion into, for example, Java, hence expanding the range of use of the App. Moreover, a sensorial vibration analysis would be useful to compare and settle the committed error of FreqNess. Nevertheless, the application results were compared with the lead supervisor manual calculations for a specific engine and were accepted to be reliable.

As regards the consistencies, the knowledge acquired in the courses, regarding engines and programming, made easier the development of the App. Moreover, the guidance and advice coming from Kaj Rintanen (lead supervisor) were essential to programming FreqNess. The guidance is reflected all over this work, and mostly in Appendix 2: Table of assumed values.

9 Conclusions

The findings of this thesis suggest that being able to test a system before its construction is a crucial step that most experiments and studies where vibration is present require, as resonance can induce catastrophic scenarios. The lack of information given by the vibration isolator companies does not allow the customer to test, understand and compare the behaviours of a specified system. This bachelor thesis presents an application that solves this problem in a general four-mount system, whose results revealed that every mode is equally important and, in every case, both translative and rotational modes needed to be tested.

FreqNess allows the users to see how their specific system behaves by varying the engine speed. Moreover, when deciding which vibration isolator will be implemented in their system, the thesis application will help to compare the different degrees of isolation in every mode for most of the market mounts. Which will save the customer time and money instead of asking for analysis from the mount manufacturers. As a way to exemplify this and based on the acquired results, the author would suggest testing other vibration isolator models. As the one currently implemented does not satisfy the customer's degree of isolation requirements in any engine speed.

Future research could be the conversion from the MATLAB programming language into another that can be downloaded freely. Moreover, a more specified vibration analysis could be done by assuming that the relation between the frequencies, leading to a more realistic scenario. Also, FreqNess could be used in any vibration machine. This can be done by redesigning the application and therefore allowing the user to choose the number of mounts, and shape of the machine. In addition, the implementation of a double mounting scenario can also be added.

This project has allowed the author to enter into the mechanics behind vibration, related to engines and mounts. The programming and designing process of an application was a useful learning experience and may prove beneficial for the educational delivery at the university, as it will allow the laboratory staff to quickly analyse their current engine mount systems.

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Appendix 1: Programming analysis implementation

Appendix 1.1

```
app.k_z.Value=app.k_y.Value*app.EMpctg_z; % Estimated transversal stiffness calculation
app.k_x.Value=app.k_y.Value*app.EMpctg_x; % Estimated longitudinal stiffnes calculation
freq_x= 1/2/pi()*sqrt(app.k_x.Value*4/app.Mass.Value); % Calculation of the frequencies in the
three axis
freq_y= 1/2/pi()*sqrt(app.k_y.Value*4/app.Mass.Value);
freq_z= 1/2/pi()*sqrt(app.k_z.Value*4/app.Mass.Value);
app.f_x.Value=freq_x; % Prompt of the frequencies in the App
app.f_y.Value=freq_y;
app.f_z.Value=freq_z;
```

Appendix 1.2

```
% Calculation of the rolling, pitching and yawing stiffnesses for each mount
k1_roll=(app.k_y.Value*(app.m1x.Value/10^3)^2*tan(app.angle));
k1_pitch=(app.k_y.Value*(app.m1z.Value/10^3)^2*tan(app.angle));
k1_yaw=(app.k_y.Value*((app.m1x.Value/10^3)^2+(app.m1z.Value/10^3)^2)*tan(app.angle));
```

Appendix 1.3

```
% Translation frequencies calculation
k_roll=k1_roll+k2_roll+k3_roll+k4_roll; % Total rolling stiffness
k_yaw=k1_yaw+k2_yaw+k3_yaw+k4_yaw; % Total yawing sitffness
k_pitch=k1_pitch+k2_pitch+k3_pitch+k4_pitch; % Total pitching stiffness
% Rotational frequency calculation
app.f_roll.Value=1/2/pi()*sqrt(k_roll/app.X_MomInt.Value);
app.f_yaw.Value=1/2/pi()*sqrt(k_yaw/app.Y_MomInt.Value);
app.f_pitch.Value=1/2/pi()*sqrt(k_pitch/app.Z_MomInt.Value);
```

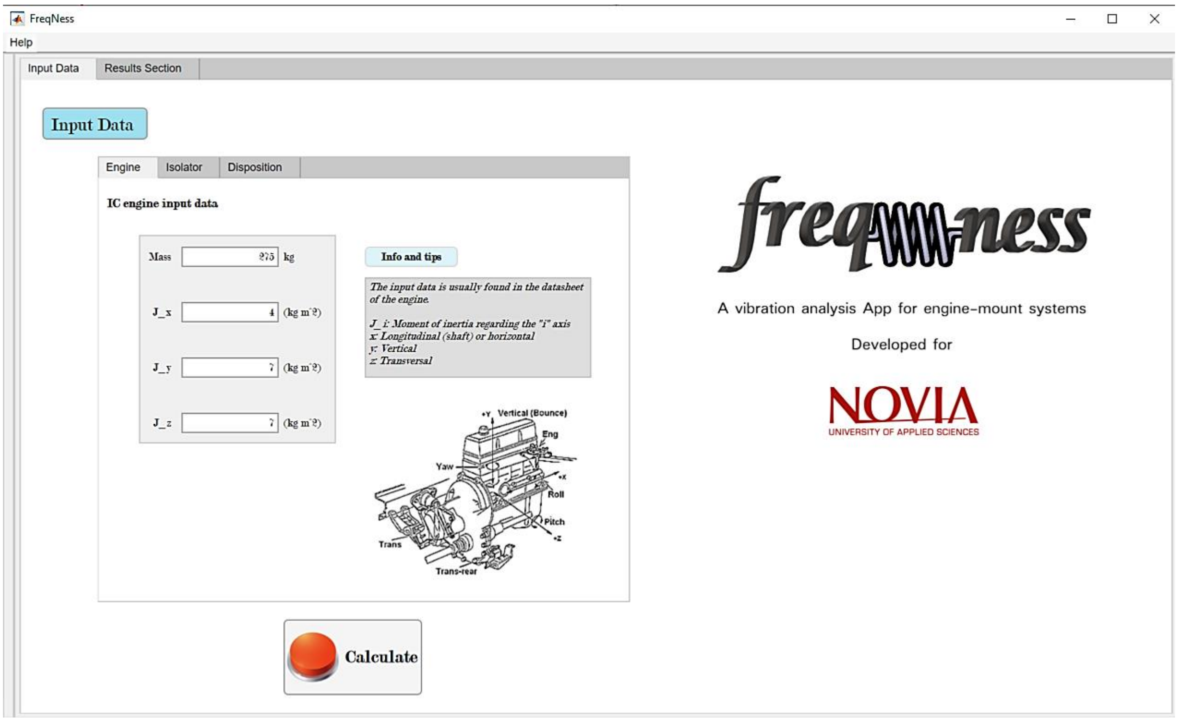
Appendix 1.4

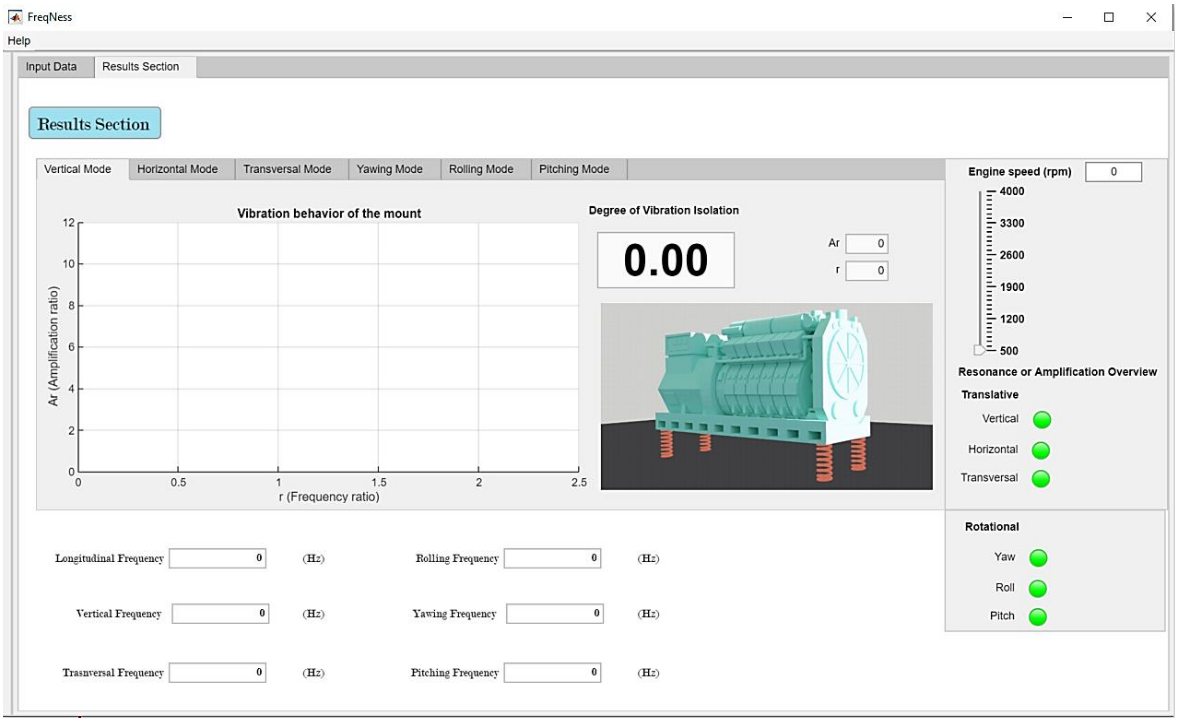
```
rax2y= rpm/60/app.f_y.Value; % Conversion into the frequency factor in vertical axis
ampy=1./sqrt((1-rax2y.^2).^2+(2*app.Mdmpc*rax2y).^2); % Amplification ratio calc
% Degree of vibration isolation
app.doiy.Value=(1-sqrt((1+(2*rax2y*app.Mdmpc)^2)/((1-rax2y^2)^2+(2*rax2y*app.Mdmpc)^2)))*100;
```

Appendix 2: Table of assumed values

Variable name	**Assumed value**	**Description of the variable**
$r_{y,z,E}$	0,5	Vertical to transversal stiffness ratio of an elastomeric mount.
$r_{y,x,E}$	0,7	Vertical to longitudinal stiffness ratio of an elastomeric mount.
$D_{f,E}$	1,3	Stiffness dynamic factor of an elastomeric mount.
$c_{c,E}$	0,1	Damping coefficient of elastomeric mounts.
$r_{y,z,M}$	0,4	Vertical to transversal stiffness ratio of a metal spring mount.
$r_{y,x,M}$	0,6	Vertical to longitudinal stiffness ratio of a metal spring mount.
$D_{f,M}$	1	Stiffness dynamic factor of a metal spring mount.
$c_{c,M}$	0,05	Damping coefficient of metal spring mounts.
$r_{y,z,A}$	0,3	Vertical to transversal stiffness ratio of an air bellow mount.
$r_{y,x,A}$	0,5	Vertical to longitudinal stiffness ratio of an air bellow mount.
$D_{f,A}$	1	Stiffness dynamic factor of an air bellow mount.
$c_{c,A}$	0,1	Damping coefficient of air bellow mounts.

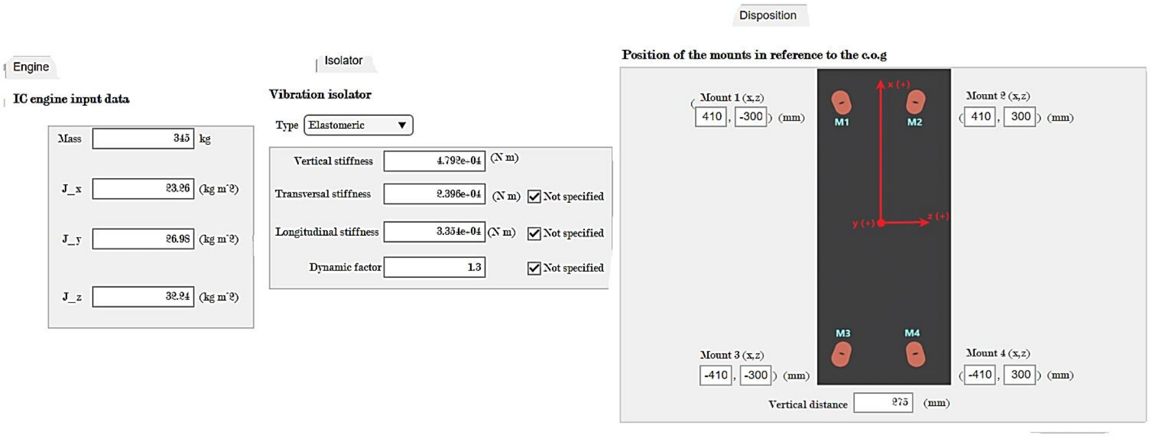
Appendix 3: FreqNess screenshots





Appendix 4: Sisuengine 420 DWI analysis

Appendix 4.1



Appendix 4.2