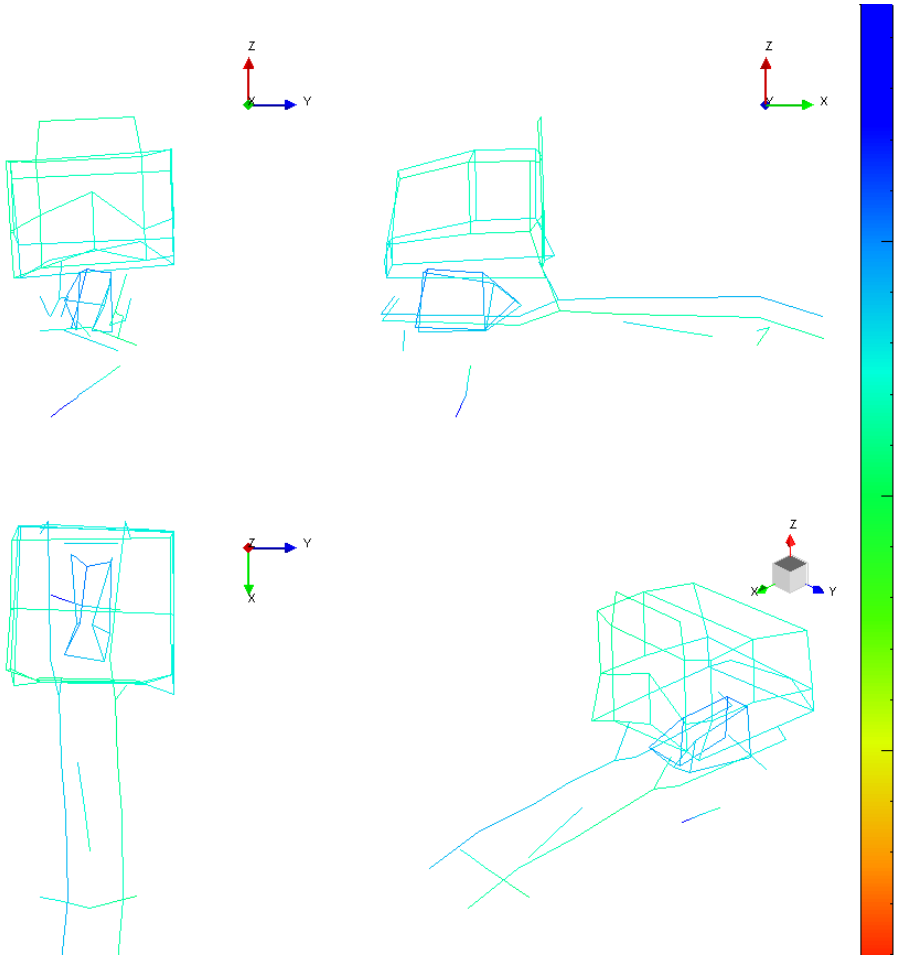
CHALMERS
UNIVERSITY OF TECHNOLOGY



Vibration characterisation of low frequency engine idle vibrations

Master's Thesis in the Master's Programme Sound and Vibration

ROBERT LARSSON

Department of Civil and Environmental Engineering
Division of Applied Acoustics
Vibroacoustics Research Group
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2016
Master's Thesis 2016:5

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Abstract

To deliver a product with high quality impression and a comfortable ride, the vibrations in a truck seat has to be well controlled and cannot disturb the driver. In order to characterise this, modal measurements and simulations are correlated with operational data. This thesis focuses on the low frequency engine idle vibrations, 0 to 20 Hz, and its behaviour in the driver seat, with emphasis on beating phenomenon occurring at engine idle rpm.

The operational measurements use the engine as the driving source and the truck placed on a concrete floor, which gives the correct boundary conditions for the beating phenomenon. In the modal measurements the truck's front left wheel is standing on a large hydraulic shaker to excite the complete truck. A setup of measurement positions is arranged to cover all vital parts of the truck. In order to make the results comparable the arrangement of measurement positions is identical for both measurement methods. The advantage of having both the operational behaviour alongside with the modal is that the driven behaviour, due to the engine excitation, can be separated from the pure modal behaviour.

A beating phenomenon could be found in the seat base of the driver seat and with the results from both the operational and modal measurement, a mode map of the complete truck was assembled. Two modulation frequencies were found and the measurements showed that the first engine order is very prominent and also difficult to attenuate along the transfer paths from engine to driver seat. The mode map consisted of several of the rigid body modes belonging to the engine and cab, as expected due to the low frequency range. Flexural modes in the chassis were also found. Comparing the mode map and the two modulation frequencies reviled that ten mode types can influence the beating phenomenon and out of those five match the modulation frequencies exact.

Keywords:
Vibration characterisation, Beating, Operational deflection shapes, Operational modal analysis, Modal analysis, Complete truck

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Last but not least I want to thank my family and especially my girlfriend for your support, it would not have been possible to achieve this without your help.

Nomenclature

Transmissibility	Response vs. Response (acceleration/acceleration)

Abbreviations

A-D	Analogue-Digital
BS	British Standard
DOF	Degrees Of Freedom
EO	Engine Order
EMA	Experimental Modal Analysis
FE-model	Finite Element Model
FRF	Frequency Response Function
ISO	International Standardization Organization
MIMO	Multiple Input Multiple Output
NTF	Noise Transfer Function
NVH	Noise, Vibration and Harshness
ODS	Operating Deflection Shapes
OMA	Operating Modal Analysis
OTPA	Operating Transfer Path Analysis
RBM	Rigid Body Mode
rpm	Revolutions Per Minute
SDOF	Single Degree Of Freedom
SIMO	Single Input Multiple Output
VTF	Vibration Transfer Function
WBV	Whole Body Vibrations

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1 Introduction

Volvo Trucks aim to sell vehicles in the premium segment of the truck market, which makes it important to deliver a product with a high quality impression. The truck has to fulfil the intended demands of the market and perform as good as, or better, than competitors. Regarding noise, vibration and harshness (NVH) it is important to analyse and evaluate design changes to fulfil the high demands of the quality impression.

Ride comfort is a crucial part of the premium impression of the truck since it affects the driver's body directly. Bad comfort can lead to muscle fatigue after a long work day due to that the muscles have to compensate for the vibrations and bumping of the seat to keep control of the body. Most of these compensations are done autonomously by the human body and the effect is detected later on when muscles are tired.

The project was initiated since a beating sensation was observed in a test truck in 2013 and initial measurements revealed that it might be possible to measure and characterise a beating in a physical manner and not only observed subjectively. At that time the beating varied depending on coordinate direction, from 0.5 to 2 Hz of modulation frequency. Resonances in the cab were suspected to be responsible for this possibly together with engine order one (EO 1).

1.1 Background

The truck can be decomposed into three major parts; chassis, engine and cab. Both the engine and the cab are connected to the chassis mechanically but not directly to each other except for weak couplings, e.g cabling. [6]. These three subsystems have their own material and structural properties leading to specific resonances of the three systems. In NVH-engineering it is important to be aware of and characterise all systems due to that the resonances affect the amplitudes of the vibrations going through the structures [13, 12]. One of several things that can happen due to the effect of resonances is beating phenomenon where one frequency is modulated with another creating a beating effect.

Beating is a phenomenon caused by two strong tones or vibrations with frequency close to each other giving the resulting tone an amplitude variation over time [13]. The caused amplitude variation over time is felt in the cab seats. Detailed explanation of beating is found in Section 3.3.

In 2013, the beating was observed having the truck idling standing still. From that, conclusions of two possible sources of the beating were outlined; EO1 and resonances. Since EO1 is suspected, the frequency range of interest is the region of rigid body modes. This light conclusions were based on the knowledge within the employees at the Noise and Vibration Laboratory at Volvo Group Trucks Technology. The engine is the only part of the truck providing an input force to the structure, since the truck is idling. Therefore the engine is involved in some way or another creating the beating. But since the engine and the cab are not connected directly the vibration energy from the engine has to pass through the chassis before reaching the cab and at the end the cab seats. This meant that the investigation

could not only include engine and cab but also the chassis. Idle rpm is equal to 550 rpm and the first engine order to 9.2 Hz and together with the prior knowledge of the modulation frequency between 0.5 to 2 Hz the frequency range of interest is therefore narrowed to 0 to 20 Hz.

1.2 Aim

The aim is to prove which resonance together with EO1 that creates the beating observed back in 2013. The study should also provide a detailed description of the low frequency engine idle vibrations between 0 to 20 Hz. Within this frequency range, the objective is to characterise the resonances and the rigid body modes of the major subsystems as described in Section 1.1. Another objective is to find suitable measurement methods to describe for the characterisation.

1.3 Limitations

The frequency range of interest 0 to 20 Hz, defined by the background knowledge, gives the possibility of having rigid body modes (RBM) influencing the vibrations. Since EO1 is believed to influence the beating the engine has to either be included in the measurements or exchanged by a controlled source with the ability to excite these low frequencies. Therefore the decision was to carry out measurements on the complete truck without dismounting any parts, as much as possible. The engine was used as source, i.e operational measurements, since this should provide the right excitation to the structure. A modal analysis was also performed for the complete truck, which has a frequency range limited to 5 to 20 Hz. The results are valid onto the used test truck and cannot be applied directly on another truck since the resonance frequencies are very dependent on the mounting and suspension parts and their ware and tare. However, rigid body modes and engine orders always exist in a truck and their interaction are possible but not always perceived as in this particular case.

2 The truck - a coupled system for vibrations

It is vital to understand the truck when investigating phenomena like beating sensation, the truck is a huge system coupled together in various points and including many rotating parts, such as gearwheels, fans and crankshaft [6]. To simplify the system and make it more understandable it can be decomposed into three major subsystems; engine, chassis and cab. The engine and the cab are not directly coupled, except for cabling, but both of them are connected the chassis; the engine through four engine mounts and the cab at seven points, six suspension points and one for the tilt function of the cab, to the chassis. The engine vibrations are transferred to the cab first through the engine mounts, to the chassis, then via one or more attachment points to the cab. Therefore it is important to have knowledge about all three subsystems when analysing engine induced vibrations in the cab. The wheel axles, both front and rear, are not two major subsystems but they do have sufficient mass to affect the vibration characteristics. Therefore they shall be included in the measurement not to miss any vital information.

2.1 Cab and its suspension

The suspension of the cab can be both mechanical and air controlled or a combination of both [6]. At the tested truck there are six connection points, two at the rear and four at the front, see Figure 2.1. The front suspension of the cab is displayed in detail in Figure 2.2 where one can see that it has both a shock absorber (2) and a mechanical spring (1). Figure 2.3 shows the rear suspension where both the shock absorber (2) and mechanical spring (1) are connected at the same point. The figures explain the schematics of the cab suspension but some changes have been made to the design. Today the cab is equipped with two lateral shock absorbers for better stability sideways; one at either side in the rear suspension set-up. Also at the front suspension some changes in the set-up of today's truck have been done. The shock absorber and mechanical spring have been put together like they are in the rear suspension but two bushings have been added instead in approximate the same position as the mechanical spring used to sit. The truck used in the measurements has only mechanical suspension and no air controlled suspension.

In addition to these six points the tilt cylinder also connects the chassis and the cab together, but it does not provide any suspension when the cab is in its upright position. This is used for tilting the cab during maintenance. Ideal it shall disconnect when the cab is put in upright position and not provide any connection between the cab and chassis.

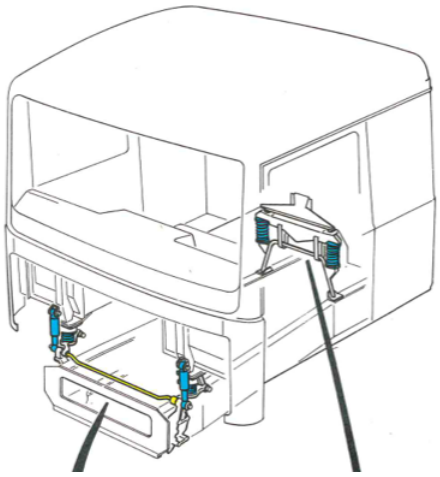


Figure 2.1: *The six points of cab suspension [6]*

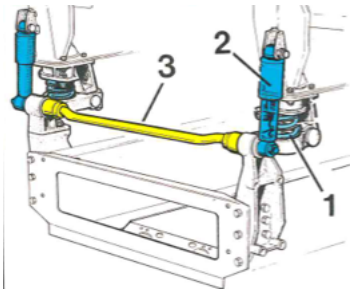


Figure 2.2: *Front suspension of the cab [6]*

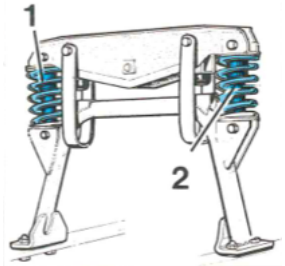


Figure 2.3: *Rear suspension of the cab, without lateral shock absorber [6]*

2.2 Engine and chassis

The chassis is the mainframe of the truck and most larger parts are connected to it [6]. Simplified; the chassis consists of two lengthways oriented steel beams as seen in Figure 2.4, to where engine and cab are connected. The chassis shall not only bear the weight of the whole truck and its load, it shall also be flexible to absorb the unevenness of the road together with suspension. The chassis design vary a lot between the type of truck, both lengthways but also in the way that wheel axles are connected, this is however left out due to the vastness of possible configurations.

The in-line six cylinder engine is oriented lengthways of the truck. It is mounted to the chassis in four connection points; engine mounts. The engine mounts are designed to isolate the engine as much as possible so that the vibrations generated due to the combustion and the rotating parts should not propagate to the rest of the truck. In Figure 2.4 the engine mount positions on the frame are marked by number 3. The front and rear wheel axles are mounted in position 4 via the suspension.

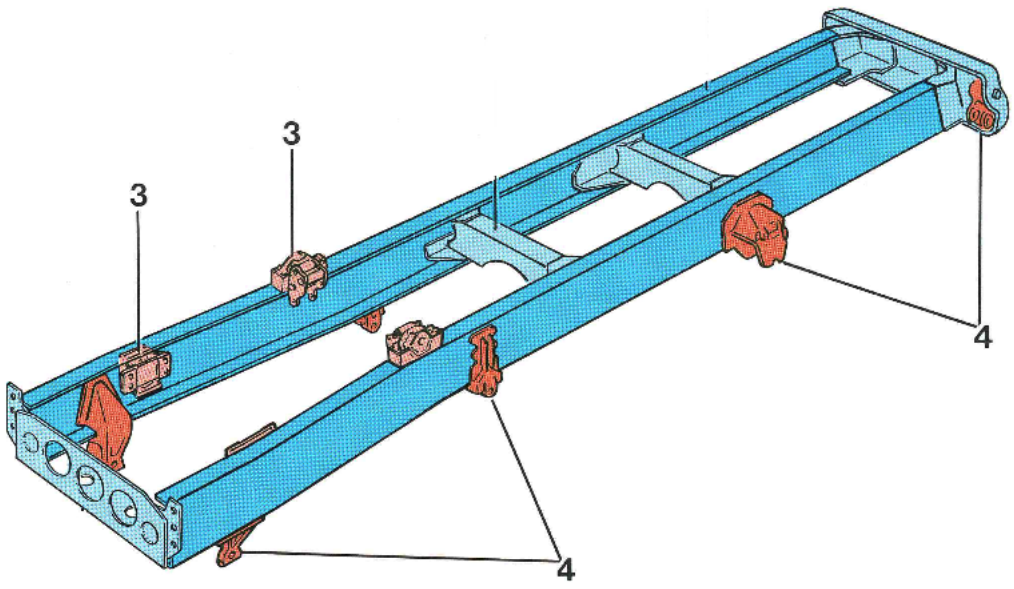


Figure 2.4: *The main frame of the chassis [6]*

3 Theory

Investigating the low frequency vibration characteristics with interest in beating phenomenon requires, in this case, measurements of the truck. A literature study of different measurement techniques was performed. The sensation of beating was also studied in order to describe and understand the beating further. The study of measurement methods together with the enhanced knowledge of beating served as the knowledge for deciding how to measure and investigate the phenomenon.

3.1 Investigation of suitable measurement methods

3.1.1 Operating Deflection Shapes

Operating deflection shapes (ODS) is a measurement technique that visualise the deflections of a structure subjected to vibrations [9]. Acceleration of the structure is measured in multiple positions. The word *operating* in the name means that the measurements are done during operating conditions of the structure, e.g the trucks engine is running. This also means that the input force to the structure is unknown during the measurements. Therefore method relies on so called transmissibility measurements, i.e acceleration over acceleration . All measured responses on the structure is related to one single response, the so called reference position. This position serve as a phase reference in order to keep track of the phase difference in all points on the structure. The analysis and visualisation of the deflection shapes can be performed both in time- and frequency domain. The amount of measured positions gives the resolution in the visualisation. It is also easier to detect false responses, i.e error in mounting of the accelerometer or local resonances in a position that does not yield for the whole structure, if having a higher amount of positions.

3.1.2 Modal Analysis

Modal analysis is a vital tool in the field of NVH-engineering [10, 2]. It is used to correlate simulations, e.g Finite Element Models (FE-models), and to explain the properties of the structure subjected to vibrations. The method uses an excitation of the structure such as modal hammer or a shaker where the input force is known. The response of the structure is measured with accelerometers in one or multiple positions and gives the frequency response functions (FRFs) that is used in the analysis. The FRFs are then used to create a synthesised model of the truck, i.e mathematical model that should represent the reality. The modes are represented by the stable poles within the measured FRFs. A stable pole is located inside the unit circle in the complex plane of the Z-transform of the FRF. Putting all measured signal together and the found stable poles they shall, ideally, be located at the same frequencies which are those where the modes of the structure are found. The analysis can be performed both in time and frequency domain. The model is improved by fitting the synthesised FRFs to the measured ones in all positions until the error is low between the synthesised and the measured FRFs. This is done by changing the number of modes, changing the resonance frequencies and also selecting the right amount of damping in each mode.

3.1.3 Operational Modal Analysis

Operational modal analysis (OMA) is a version of the previously explained modal analysis, Section 3.1.2 [11, 4, 14]. Instead of using an excitation of the structure with a known force, OMA uses the original source, e.g the engine of the truck, where the source excites the structure with an unknown force. Force estimation is often a hard and time consuming task in NVH-engineering, as explained in Section 3.1.4. Therefore it is attractive to save this time and try to evaluate the modal behaviour of a structure without the need of input force knowledge.

Similar to the measurement done in an ODS study, explained in Section 3.1.1, OMA uses transmissibility measurements [11, 4, 14]. One accelerometer is used as reference response to which all others are related to. To find modal properties in OMA measurements, the measured responses have to be processed more than in an ODS. This can be compared to the modal analysis where resonances in the structure are located via finding the stable poles in the measured signal. Having found the stable poles and the corresponding resonance frequencies the modes can be visualised.

3.1.4 Transfer Path Analysis

Transfer path analysis (TPA) is mainly used to evaluate and refine already existing structures [8, 15, 12]. A typical task involving TPA is to fine tune engine mounts at the end stage in the development when dimensions of the surrounding structures are defined. TPA is also used to correct and fine tune calculation models such as FE-models in order to improve the precision and the imitation of reality in models.

The basics of TPA requires that the structure can be divided into an active and a passive structure [8, 15, 12]. Meaning that the active structure provides a force/vibration, e.g an engine, and the passive structure transmits the vibrations to a receiver, e.g the driver in a truck. Consider an engine connected to a chassis via a number of engine mounts. The engine generates an input force which is transmitted through the engine mounts and are defined as input paths to the passive structure. They represent the inflection point between the active and passive structure. The chassis transmits the vibration to the receiver which can be the driver but also the seat, steering wheel or the door panel. A basic sketch of the structure and the different parts is seen in Figure 3.1 and the general equation in classical TPA is seen in Equation 3.1. y_k represents the magnitude of the noise or the vibration at the receiver, VTF stands for vibration transfer function and F_i is the input force. The methods require that one finds both the VTF and the input force, which is further explained later on in this section.

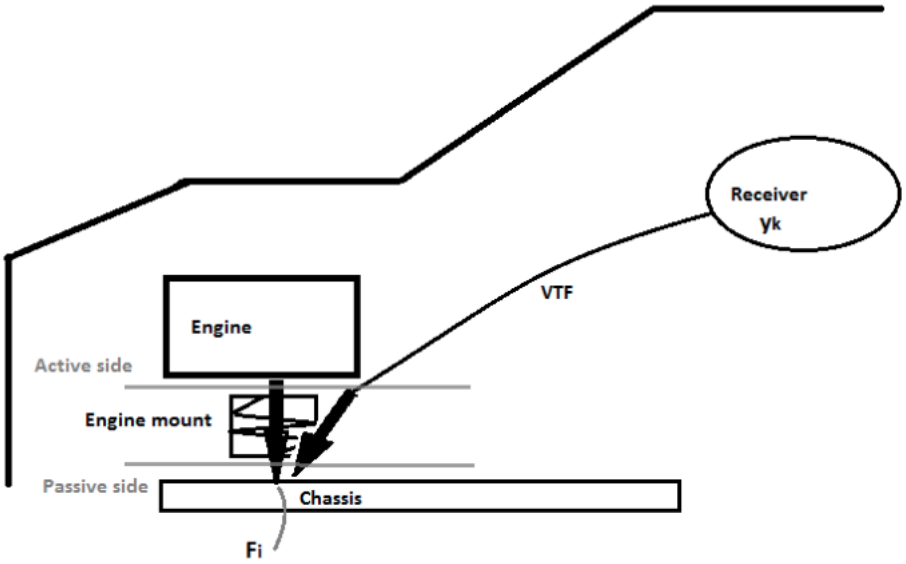


Figure 3.1: *Basic sketch of a TPA setup [8]*

$$y_k = VTF \cdot F_i \tag{3.1}$$

Finding the VTF

The VTF is measured by exciting the input path at the passive side and measuring the response at the receiver position. The excitation is done so that the input force can be measured, e.g with a modal hammer. There exists as many VTF as there are input paths between the active and passive side. In Figure 3.1 the VTF is the long arrow between the receiver and the underside of the engine mount, which is the point of excitation.

Load/source identification

Load identification is often the hardest part of a TPA [8, 12, 15], due to the design properties of modern structures and that engine mounts are very well defined these days. Modifications are often impossible or would change the properties too much. The three most common load identification methods are:

1. Direct method
2. Mount stiffness method
3. Inverse matrix method

Table 3.1: Load identification methods

Ranked in order of measurement effort, low effort for the first and highest for the third. Using the direct method requires placing of a force transducer between the active and the passive

structure to measure the input force directly [8]. This is in most cases impossible due to lack of space and that it would destroy the often well defined properties of the mounting.

The mount stiffness method requires knowledge about the stiffness (K_ω) of the actual engine mount [8] and the acceleration of the mount, which the difference between the acceleration of active and passive side of the engine mount ($a(\omega)_{active} - a(\omega)passive$). The input force ($F(\omega)$) is then calculated with Equation 3.2. However, the required data for the stiffness, $K(\omega)$ might not be available and if so with low accuracy. Therefore the mount stiffness method is sometimes unusable.

$$F(\omega) = K(\omega) \cdot \frac{a(\omega)_{active} - a(\omega)_{passive}}{-\omega^2} \tag{3.2}$$

The third classical method described here is the inverse matrix method [8, 15]. This method requires measurements in operating conditions and also with a known source, e.g a modal hammer. The input forces, F_i, through all input paths are estimated by Equation 3.3. F_i are the input forces to the structure, FRF-matrix is the frequency response function of all transfer path in the structure and a_i are the indicator accelerations. The force/load identification is done in two steps, first the indicator accelerations are measured during operating conditions, i.e with the engine as the source. Next step is to measure the FRFs in the matrix, which is done with exciting the structure with a known force at the passive side of the source and measuring the response in several positions of the structure, this step is done with the original source removed from the structure. One excites one input path at the time and measuring in all response positions. Equation 3.3 often requires an over-determination by a factor two to gain numerical stability, i.e twice the number of indicator responses compared to excitations, according to [12]. Therefore this is the most time consuming method of the three presented load identification methods. The response of the structure is low at the anti-resonances and those FRFs have a low signal to noise ratio, in order to eliminate this, single value decomposition is often used [1].

$$[F_i] = [FRF : s] \cdot [a_i] \tag{3.3}$$

3.1.5 Operating Transfer Path Analysis

A variant of the traditional TPA described in the text above is the Operating Transfer Path Analyis (OTPA) [15, 1]. As the name implies the method involves measurement during operational conditions. The original source, e.g the engine of a car or truck is used as source which provides an input force to the structure. No estimation of the input force is done and the method relates the response at the passive side of the input path to the response at the receiver, as described in Section 3.1.1. This is called transmissibility measurements and it is a fast version of TPA since the load identification process is removed. However, the drawback is that the method hardly can predict what a change in a transfer path could lead to, since it is depending on the load that is unknown. OPTA should be used to evaluate an existing structure where it can diagnose strong transfer path and weaknesses. The interpretation of the result has to be done carefully and the analyst has to be aware of the weaknesses when drawing conclusions.

3.2 Amplitude- and frequency modulation

Amplitude modulation is an effect of having one strong tone, called the carrier frequency, and two less strong tones, one below and one above in frequency [13]. This creates a amplitude variation in time domain of the strong carrier frequency.The classical and simplest form of amplitude modulation is also known as beating, explained in Section 3.3, where only two strong tones create an amplitude modulation over time.

Frequency modulation maintain the same amplitude over the whole signal but varies the frequency over time instead [13]. It oscillates around a center frequency symmetrically and with a certain modulation frequency, the step in frequency is the modulation.

3.3 Beating

Beating is a phenomenon that occurs when for example two sound waves with frequencies close to each other but not the exact same interact with each other and create a modulation in the amplitude over time [13]. The most simple example is two sine tones with equal amplitude, the first at 100 Hz and the second at 90 Hz and the sum of the two tones is displayed in Figure 3.2. As the figure illustrates in the time domain, the interaction of the two tones create a beating effect where the amplitude is periodic. The frequency difference between the two tones corresponds to the beating frequency, in this case 10 Hz or 0.1 seconds as seen in Figure 3.2. Equation 3.4 describes the beating phenomenon in a mathematical way, where f_1 and f_2 represents the two tones.

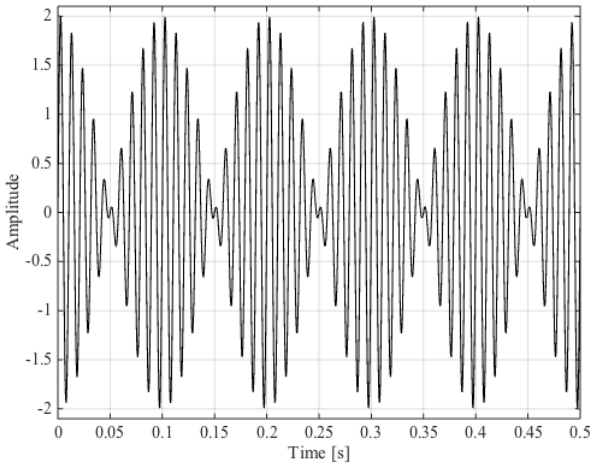


Figure 3.2: *Two sine tones with equal amplitude, $f_1 = 100$ Hz and $f_2 = 90$ Hz, the amplitude is normalized in the graph. A beating effect is created.*

$$\sin(f_1 \cdot t) + \sin(f_2 \cdot t) = 2\cos(\frac{f_1 - f_2}{2} \cdot t) \cdot \sin(\frac{f_1 + f_2}{2} \cdot t) \tag{3.4}$$

3.3.1 Beating in a single degree of freedom system

A version of the classical beating, which is explained above, is when one tone and a resonance interact with each other and create a beating [3]. Mathematically it is the multiplication of the two sinusoids in equation 3.4 instead of addition. This is illustrated in Figure 3.3 and 3.4 where a single degree of freedom (SDOF) system with a resonance frequency of 159.2 Hz is exposed to a force with a fixed frequency. In the left figure the frequency of the force is equal to 155 Hz and to the right equal to 40 Hz. The results show that the modulation frequency also in this case is equal to the difference between the tone and the resonance, i.e in the left figure 4.2 Hz and in the right 119.2 Hz.

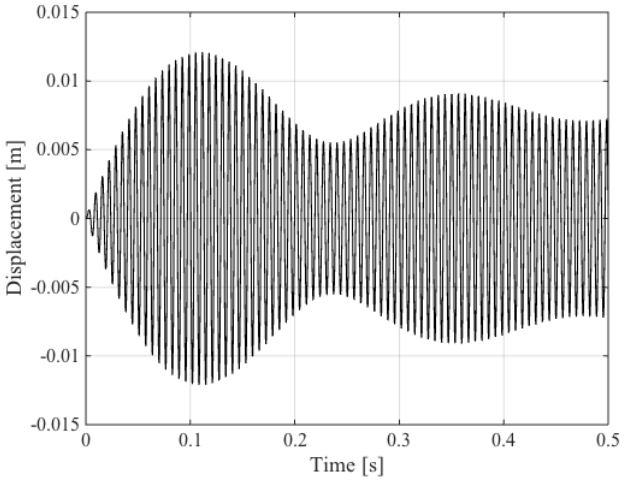


Figure 3.3: *Modulation frequency 4.2 Hz*

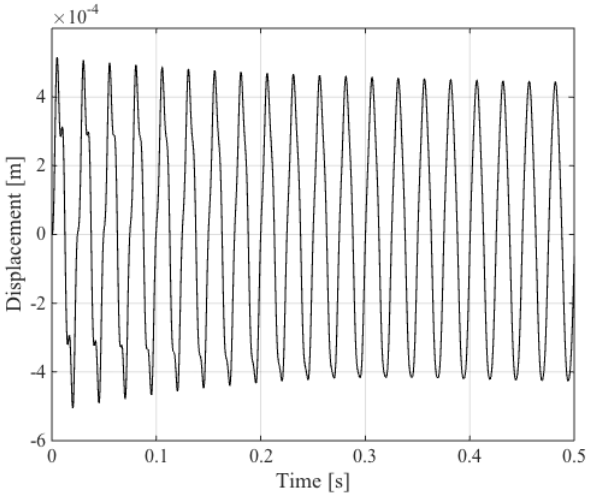


Figure 3.4: *Modulation frequency 119.2 Hz*

3.4 Fluctuation strength and Roughness

The psychoacoustic parameters Fluctuation Strength and Roughness are two parameters that can be used to describe the beating phenomena. They are related to each other but are used in different frequency ranges [13]. Fluctuation strength from 0 to 20 Hz and Roughness from 15 to 300 Hz. This is due to the perceived sensation is different in these two frequency regions.

Both parameters are described by their unique unit and their range is 0 to 1. [13] Fluctuation strength has the unit *Vacil* which is equal to one (maximum) at the modulation frequency 4 Hz. *Asper* is the unit for roughness which is equal to one at 70 Hz modulation frequency. [13]

The sensation is created by the explained beating or modulation and is dependent on the modulation frequency but the carrier frequency can be both low and high [13]. The sensation Fluctuation strength is explained as a fluctuating sound, imagine the sound from helicopter wings. For Roughness the sensation is best explained by the sound having a raw and rough tone.

3.5 Whole body vibration

Whole body vibration (WBV) appears in the frequency range of 0.5 Hz to 100 Hz and the effects of it can be divided into five categories [7].

1. Degraded comfort
2. Interference with activities
3. Impaired health
4. Perception of low-magnitude vibrations
5. Occurrence of motion sickness

Table 3.2: WBV categories, [7]

The human body reacts differently to vibrations due to their coordinate direction and to which part of the body the vibrations affect [7]. Coordinate direction for the measurements of WBV related to the human body are shown in Figure 3.5. Muscles stabilise the body and the composition of them give the ability to be more or less good in stabilisation depending on the direction of vibration. In the horizontal direction, both x and y-direction, the body loses control at about 1 Hz and starts to sway. Increasing the frequency up to about 3 Hz the body has a hard time maintaining control. However, the help of a backrest makes the body resist this swaying in most cases. The backrest will introduce a large contact area, i.e the whole spine, which will lead to more vibrations reaching the body. It is mostly high frequency vibrations that will increase due to the backrest. In the vertical direction, z, the human body has a resonance frequency of ∼5 Hz and it is also at this frequency that human perceives the highest discomfort when subjected to vibrations.

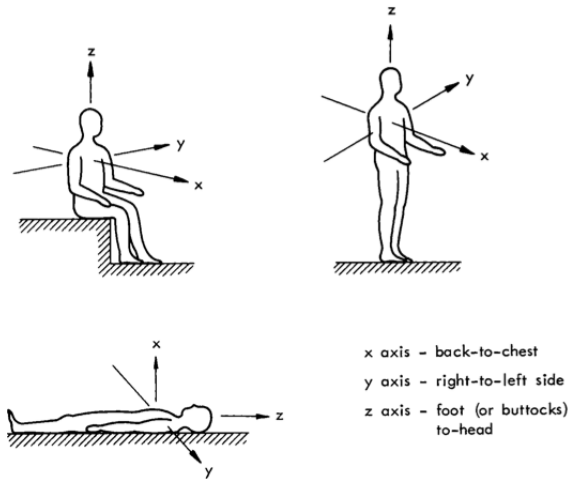


Figure 3.5: *Definitions of axis according to ISO 2631 [7]*

Two major standards exist when measuring WBV in vehicles, ISO 2631 and BS 6841 [5]. The difference between them both is the interpretation of the analysis. The BS 6841 defines strictly which axis that shall be evaluated and averaged, but in ISO 2631 the axis of evaluation are chosen by the analyst to a greater extent, either the worst case axis or the frequency weighted acceleration of all axis can be chosen.

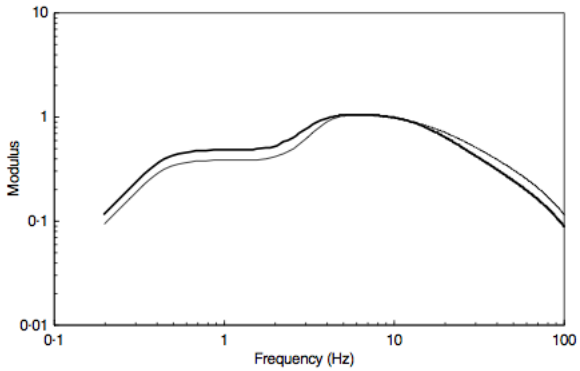


Figure 3.6: *Weighting curves, BS 6841 - thick line and ISO 2631 - thin line [5]*

To compensate for the properties of the human body, such as resonance frequency, inertia etc, the analysis of WBV has to be filtered [5]. Figure 3.6 displays the vertical filter curves for the two mentioned standards. Modulus, unit on the y-axis, represents the factor of which the acceleration at a specific frequency is multiplied with. As one can see the highest modulus, equal to one, is located at around 5 Hz where the resonance frequency of the human body is located.

4 Choice of methods for the investigations

From the gathered knowledge in the theory section, Section 3, the plan for the rest of the project was outlined. The characterisation of the vibrations was performed by a characterisation of the movement of the structure, instead of a characterisation of the transfer paths, like a TPA. The decision is based on two facts, one that a TPA often requires a large effort in establishing the input force. Secondly a characterisation of the movements can be done both in operating condition and with a known input force, without too great effort. In the available time frame the characterisation of the movement is likely to deliver more useful information about all different subsystems than a TPA would do. In this case the aim is to prove if a resonance together with the first engine order creates the observed beating, and therefore the resonances has to be located. Operational deflectional shapes, operational modal analysis and modal analysis shall be used to characterise the movement.

It was decided to gather information regarding the movement from simulations of the complete truck as well as the engine itself; in order to have a better understanding of what to expect during the measurements and the analysis.

At first, before any measurements, a sufficient amount of good measurement positions on all major subsystems had to be identified, together with a repeatable measurement procedure, i.e controlling temperatures, fan- and engine speed. After that operational measurements and the one with a known input force could be performed.

5 Finite element simulations

Theoretical knowledge of the rigid body and flexural modes was gathered to serve as an initial knowledge for the measurements since the modal behaviour of the truck is believed to influence the beating. The truck can be dealt with as one complete system or it can be divided into the three subsystems as described earlier in the report. It was chosen to do a simulated modal analysis for the engine itself to find its six rigid body modes, along with that a simulated modal analysis of the complete truck was done, including the engine. All simulations focused on the frequency range 0 to 20 Hz. The simulations were done in the FE-program NASTRAN by Volvo AB employees and used a complete setup of the truck in the simulation, including all equipment fitted on the real measurement truck.

5.1 Results of finite element simulations

Rigid body modes are directly linked to the six degrees of freedom (DOF) that a solid suspended mass have; three translational DOF in the coordinate directions and three rotational DOF around the coordinate axles. Other modes are so called flexural modes, where the structure itself bends. Rigid body modes were expected to be found for the engine and the cab itself. Some flexural chassis modes were also expected to be fund in the frequency range 0 to 20 Hz.

Table 5.1 describes the type of motion and the corresponding name of the six rigid body modes.

Surging	Movement in X
Sway	Movement in Y
Bounce	Movement in Z
Roll	Rotation around X
Pitch	Rotation around Y
Yaw	Rotation around Z

Table 5.1: Definition of the six rigid body modes

5.1.1 Engine

All six rigid body modes were found within the 0 to 20 Hz range and several of them placed close to the 9.2 Hz EO1 frequency. The frequency of the rigid body modes is set by the suspension and mass of the engine. The suspension consists of four engine mounts manufactured by rubber with a chosen stiffness to provide isolation of the engine from the rest of the structure, so that as little vibrational energy as possible is transferred from the engine into the chassis. The front engine mounts are positioned on a subframe and in the circular radius of the crankshaft in order to handle the rotating forces from the engine. The rear mounts are positioned in the rear corners of the engine.

5.2 Complete truck

The simulation for the complete truck gave a larger picture of how the system ideally behaved when all parts were connected since they depended on each other. In the frequency range 0 to 20 Hz more than 20 modes were found where many of them had similar behaviour and shape. However, some clear differences existed, such as flexure modes on the large chassis beams, both sideways and vertical flexure. Also the wheel axles proved to have clear rigid body modes. For the cab it was harder to distinguish its modes much due to that it was a heavily damped system with many connection points and a non symmetrical point of gravity, which lead to a rotational behaviour in the horizontal plane. Although some of the ridged body modes could be located.

6 Equipment and measurement layout

This section includes the practical setup of the measurements, i.e measurement positions, software and hardware.

6.1 Measurement Equipment

A limiting factor for all measurements using accelerometers is the range of the equipment. In this particular project it is the lower frequency limit that is the crucial one. The different models used have a lower frequency limit between 0.6 Hz to 2 Hz. The upper limit varies between 3000 to 6000 Hz. They also have different sensitivities, 10 mV/g and 100 mV/g, which is good because of the rather high amplitudes of acceleration at some positions. A couple of the PCB accelerometers are also high temperature ones suitable for mounting on the engine. All used accelerometers are triaxial, i.e measuring in x-,y- and z-direction. Depending on the measurement position, three different mounting solutions were used, magnet, glue gun and cyanoacrylate. They all have advantages and disadvantages but all satisfies the frequency range of interest.

Type of Equipment	**Brand**	**Serial No.**
Accelerometer 1, Type 4506	Brüel & Kjær	2267126
Accelerometer 2, Type 4506	Brüel & Kjær	2267284
Accelerometer 3, Type 339A31	PCB	6519
Accelerometer 4, Type HT356A02	PCB	80918
Accelerometer 5, Type HT356A02	PCB	80916.
Accelerometer 6, Type 356A02	PCB	596558
Accelerometer 7, Type HT356A02	PCB	105495
Accelerometer 8, Type HT356A02	PCB	79853
Accelerometer 9, Type M356A02	PCB	20410
Accelerometer 10, Type 339A31	PCB	6599
Accelerometer 11, Type 4506	Brüel & Kjær	2328797
Accelerometer 12, Type 4506	Brüel & Kjær	2267285
Accelerometer 13, Type 4506	Brüel & Kjær	2328805
Accelerometer 14, Type HT356A02	PCB	80913
Accelerometer 15, Type HT356A02	PCB	76106
Accelerometer 16, Type 356A02	PCB	59865
Accelerometer 17, Type M356A02	PCB	20413
Accelerometer 18, Type HT356A02	PCB	20412
Frontend Scadas III	LMS Test.Lab	41024609
Frontend Scadas Mobile	LMS Test.Lab	55113110
Seat Accelerometer, Type 2560	Endevco	10010
Computer		Running measurement software
Computer		Running the truck and logging data

6.2 Software

LMS Test.Lab 14A were used as both acquisition software and analysis software in the project. The operational measurements used the Signature testing - Standard module. When performing the FRF measurements the Multiple Input Multiple Output (MIMO) Sweep & Stepped Sine Testing module was used, but to preform a Single Input Multiple Output (SIMO). For the analysis the following major add-ins were used: Operational Deflection Shapes, Operational modal analysis, PolyMAX - Operational modal analysis, Modal analysis, PolyMAX - Modal analysis.

Volvo Group Trucks Technology uses a software named ATI Vision to control the truck remotely. It gives the possibility to manually set the rpm, fan speed etc, and also log CAN data. A prefixed script with settings was made so that all the measurement runs were preformed as similar as possible regarding engine mode, engine speed, fan speed etc.

6.3 Truck specification

Volvo ID	FH-1552
Reg.no	DWG 699
Chassis.no	A-74 88 23
Year	2013
Type	FH 4x2T (Thereof two wheels driving)
Wheel base	3700 mm
Engine type	D13K460
Bhp	460 Hp

Table 6.1: Truck specification

6.4 Measurement positions and geometry

In order to perform two runs of measurements with different methods a layout of suitable measurement positions was created. It serves as a blueprint for the geometry setup created in Test.Lab and ensures that the same positions are measured in both runs. The positions have to be connected to the rigid structure of the cab, chassis etc, in order to locate the rigid body modes and the large flexure ones of the different subsystems.

In total 71 positions were selected to represent the structure. This includes points on all major subsystems of the truck such as the cab, engine and chassis, but also on both wheel axles since they proved to have their rigid body modes in the frequency region of interest when running simulations on the truck. For all positions where the different subsystems connect to each other, measurement points were selected on both sides of the couplings, i.e on both the chassis and the engine for all engine mounts. To identify a rigid body mode a minimum of nine DOF has to be measured, three in each coordinate direction, three measured positions with x-, y-, and z-direction in each. For most of the measured subsystems more than three positions were

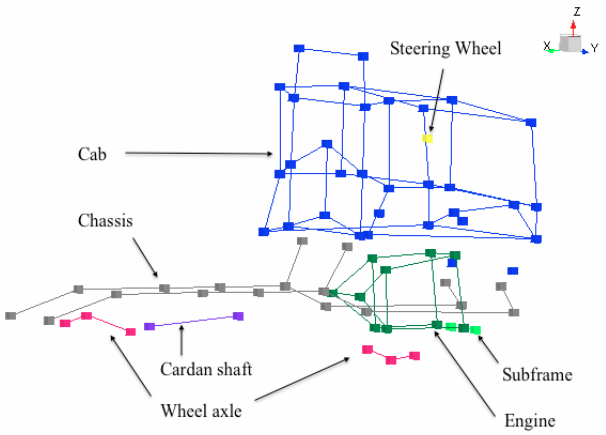


Figure 6.1: *Complete display of all measurement position on the truck*

measured, because it gives the opportunity to find flexural modes of the subsystems as well as rigid body modes.

Since analysis of operational measurements and its analysis requires a reference signal the rear right engine mount was selected as reference. This because it is located directly on the source, i.e the engine.

Figure 6.1 shows all selected positions and their positions in the coordinate system. The positions are named according to the following schematic; *Component Name:PositionXX*. As an example the reference position is named Engi:ReLeMt, which stands for Engine:RearLeftMount or in text the Rear engine mount on the left side of the engine. In the appendix detailed pictures of the different components and all names for the 71 measured positions are provided together with an explanation of the used abbreviations.

The coordinate directions that yields for the measurements are set to:

X-direction	Lengthways of the truck, positive going from front to back
Y-direction	Sideways of the truck, positive going from driver side to passenger side, left hand side drive
Z-direction	Vertical, positive going upwards of the truck

7 Operational measurements

This measurements shall serve as data for both the ODS and the OMA. The measurement positions are specified in section 6.4 and in the Appendix. The excitation energy during the operational measurements came from the engine. Since the whole project focused on the vibration characteristics at the engine idle rpm the operational excitation was located there. However, it was not sufficient to have a static rpm since the EO was then harder to identify and separate from each other. Therefore rpm sweeps were carried out. They also provided a better excitation of the structure and its modes. The sweep started at 490 rpm and went to 710 rpm, the actual measured range was 500 to 700 rpm. It simplified the measurements if the sweep started below and finished above the actual range of interest since a trigger function in the acquisition software could be used. During the rpm-sweeps the truck was placed standstill on a concrete floor with the parking brake applied.

At first different sweep rates for the rpm sweep were tested to investigate if the sweep rate had an influence of the excitation of the system. Three different sweep rates were tested, 2,3 and 6 rpm per second. As a parameter study of EO1 was performed, in Figure 7.1 the level of EO1 at the position Cab:Seat1 is displayed and the same in Figure 7.2 for the rear right engine mount. The difference between the sweep rates was quite small and it is only in y-direction that there was a noticeable difference. In order to get as much details as possible a low sweep rate was desired, however going as low as 2 rpm per second lead to the chance of having the air compressor starting and affecting the measurement, i.e the measurement was not longer valid. Therefore the sweep rate of 3 rpm per second was selected.

Before the rpm-sweep a 3 hour long "temperature sweep" was conducted. A temperature sensor was mounted to the rear right engine mount to monitor the temperature close to the rubber parts of the engine mount. Starting with the truck in room temperature and finishing when the temperature reached steady state, which happened at 72°Celsius. This experiment also served as a mark for pre-heating the truck that was done before all rpm-sweeps and modal measurements.

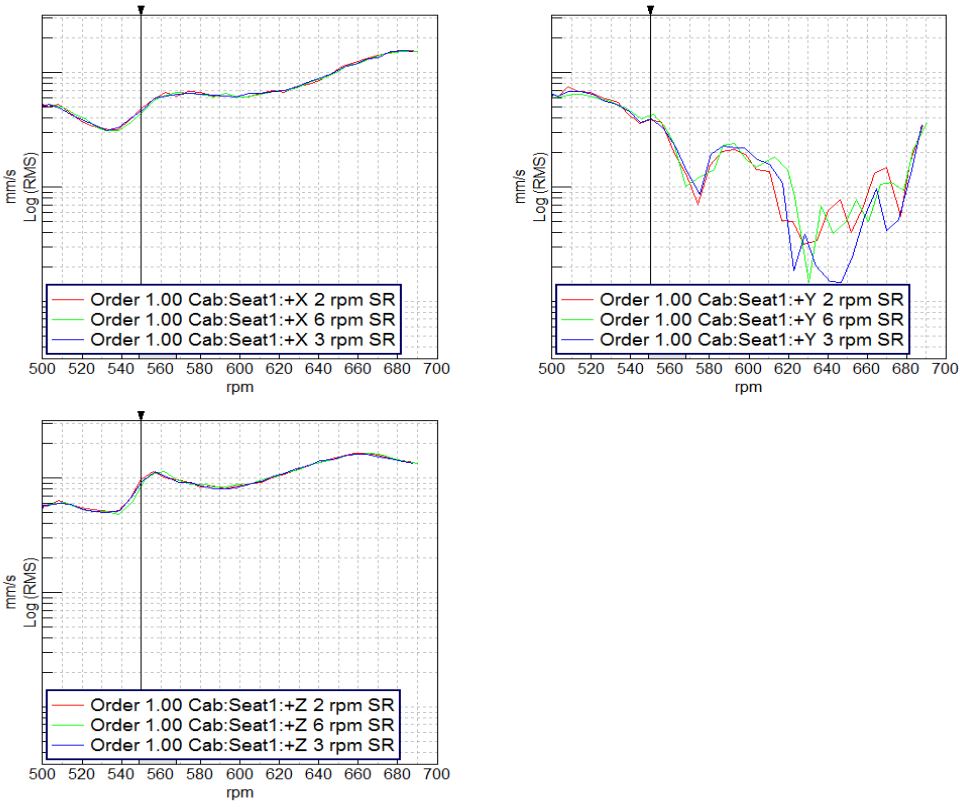


Figure 7.1: *EO1 Cab:Seat1 Sweep rate 2,3,6 rpm/s*

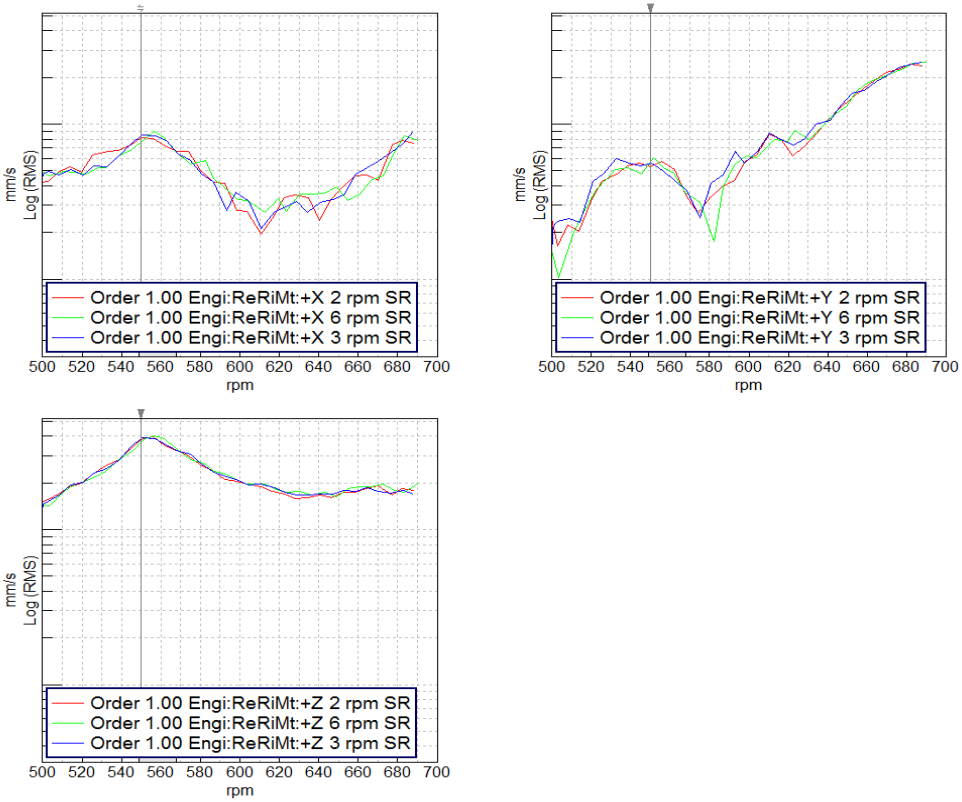


Figure 7.2: *EO1 Engi:ReRiMt Sweep rate 2,3,6 rpm/s*

7.1 Data acquisition and online processing

Time signal data from all measurements was saved. The time data was sampled at 1024 Hz frequency and processed online with 0.1 Hz frequency resolution. The rather high frequency resolution was needed in order to be able to separate closely spaced modes in the structure. The software featured the possibility to perform online processing of several parameters. It was used to validate every measurement run directly, before repositioning the accelerometers. As verification of the measurements the responses of the first and third EO were used since these are two prominent EO. If both responses were clear the measurement received a pass and the repositioning could be done.

Two different types of A-D sampling strategies were used, fixed samples per time unit with hanning windowing function, and fixed samples per revolution with rectangular window. The first to study events in the time domain and the latter in angle domain to track orders.

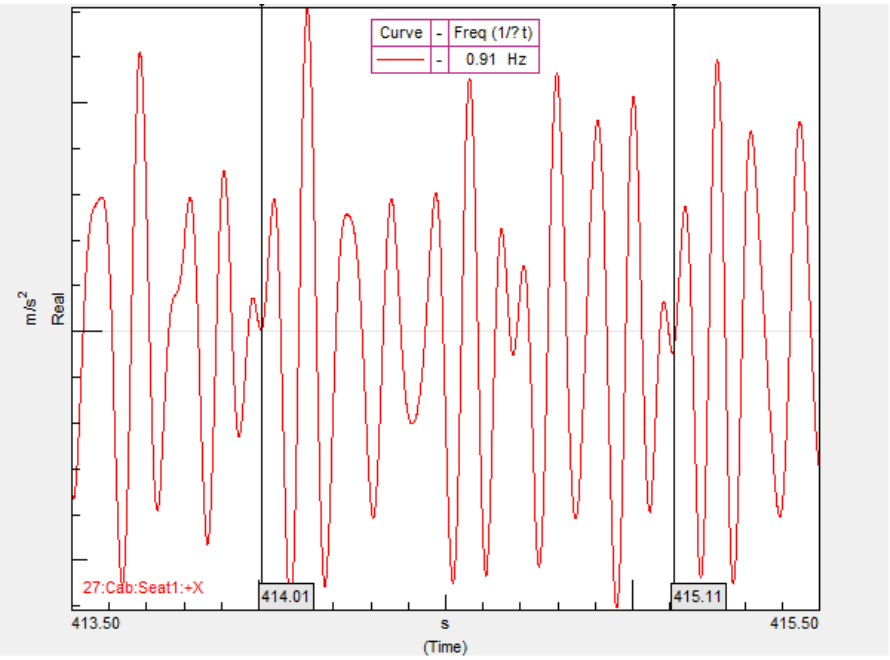
7.2 Operational measurements results

7.2.1 Cab seat vibrations

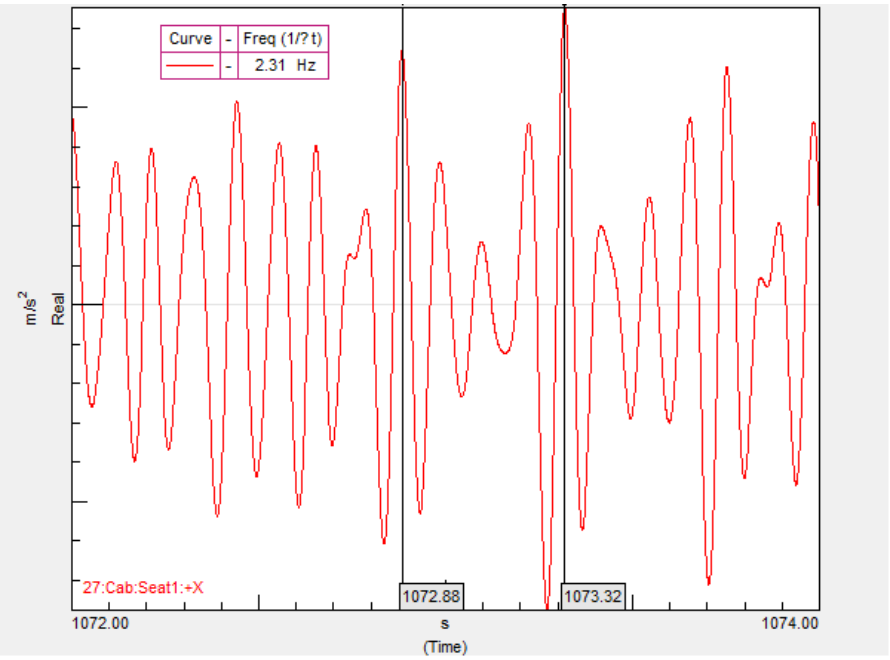
The aim of the work was to couple a subjectively observed beating in the seat vibrations to EO1. To do this it was vital that the beating in the cab seat could be objectively captured

since it gave information about the actual modulation frequency, which could help determine which resonance that was suspected to influence the beating. Before any measurements were carried out the truck was subjectively investigated and one could determine that there existed a beating effect in the driver seat. It was not as clear in the passenger seat as in the drivers seat. It was also observed that the behaviour of the vibrations in the seat changed as the engine became warmer. Along with the temperature increase the perceived vibration spectra in the seat became clearer, with more distinct harmonics and also a more clear beating effect, not necessary stronger but more distinct.

The acceleration signal recorded at Cab:Seat1+X, i.e the base of the driver's seat, was acquired with the settings described in Section 7.1. In order to analyse the low frequency seat vibrations the signal was filtered. A low pass butterworth IIR-filter with cut-of frequency of 13 Hz was used. The processed signal was then analysed and by that two different modulation frequencies were found, the first equal to 0.91 Hz and the second 2.31 Hz. These two modulation frequencies and the filtered signal are shown in Figure 7.3. The modulation is strongest in the x-direction and could be observed with both cold and warm engine.



(a) Beating with 0.9 Hz modulation frequency, x-direction



(b) Beating with 2.31 Hz modulation frequency, x-direction

Figure 7.3: *Beating in the X-direction*

7.2.2 Operational deflection shapes

The visualisation of the ODS is presented in Figure 7.5 and 7.7 and for each engine order the deflection is shown in two figures with phase difference 180 °between them. Figure 7.5 shows that EO1 excites not only the engine but also the chassis and the cab. Figure 7.7 shows that EO3 excites the engine and also the chassis but the cab was fairly stable, very little motion. The deformation of both cab and chassis due to EO1 proved that the outlined

theory in the beginning of the project was still valid, since the vibrations from the EO1 had to transfer from the engine to the rest of the different subsystems of the truck in order to create a modulation. At EO1 the engine had a pitching movement, the chassis bends around Y, i.e a vertical deflection in the rear of the chassis, and the cab had a small yaw-motion. Looking at EO3 there was a clear rolling motion of the engine and a bending deflection in the chassis but no, or very little deflection of the cab. Figure 7.4 displays the complete non-deformed geometry of the truck model.

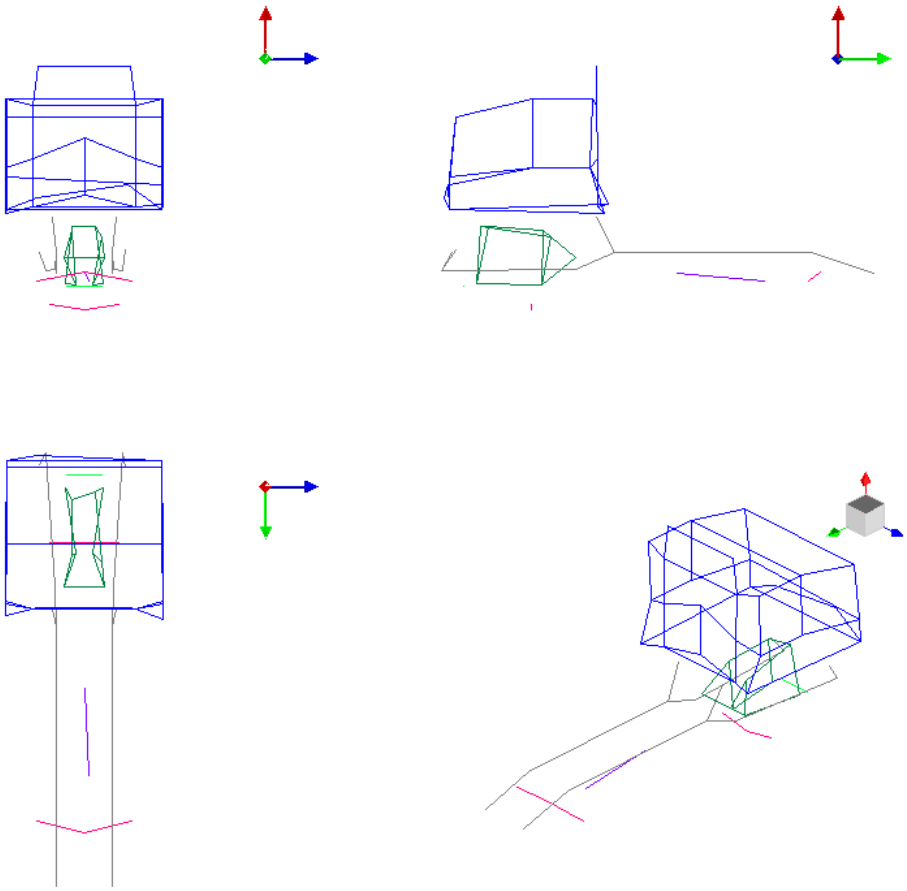
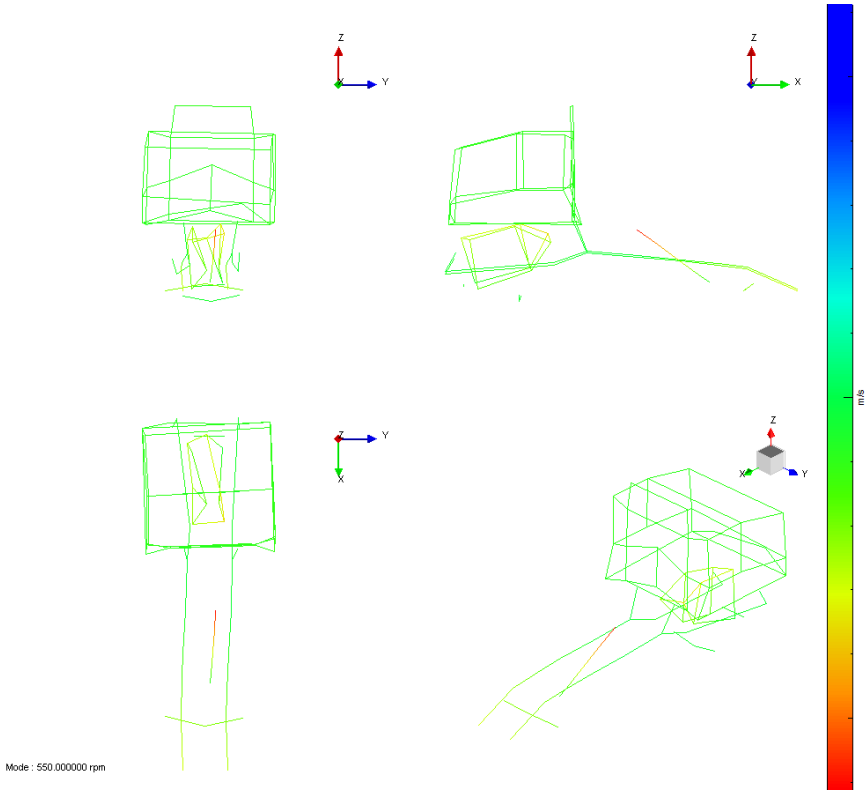
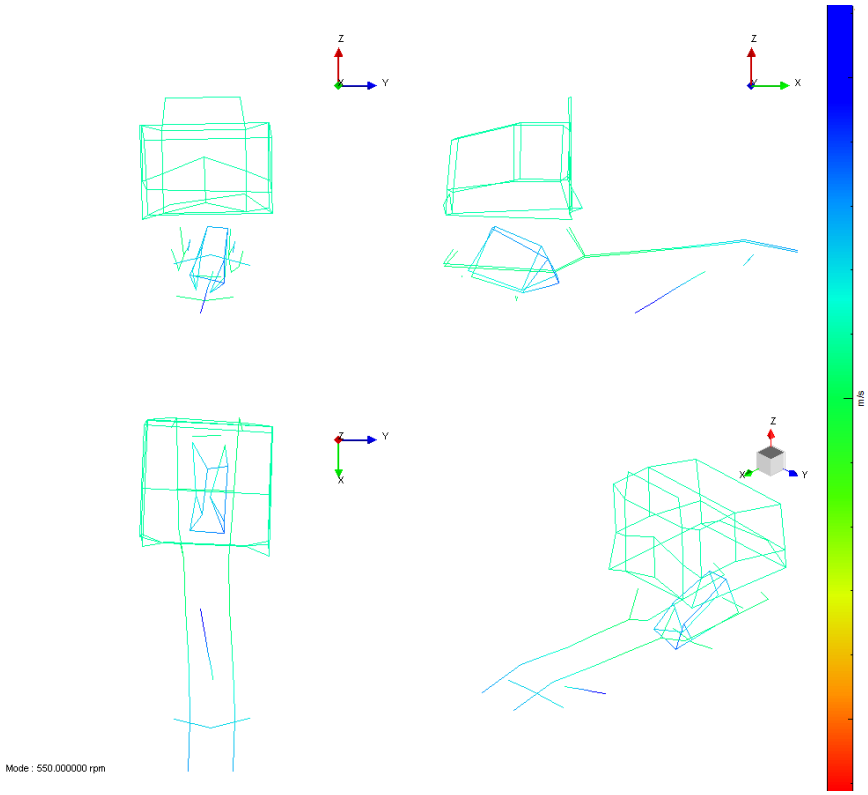


Figure 7.4: *Non-deformed geometry of the truck model*



(a) EO1, phase 0 °



(b) EO1, phase 180 °

Figure 7.5: *EO1 response at 550 rpm*

Figure 7.6 shows the sum of all measured transmissibilities and one can notice that there is a clear peak in the response at 557 rpm, very close to idle rpm.

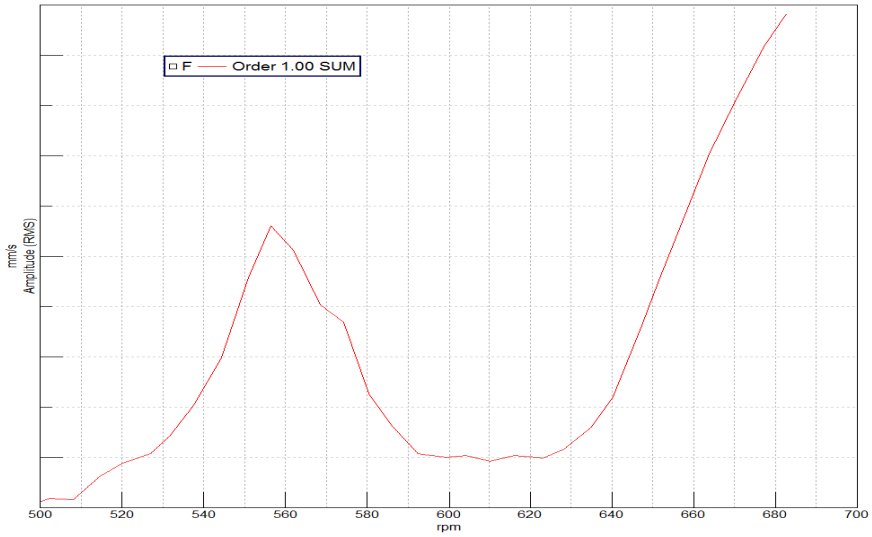
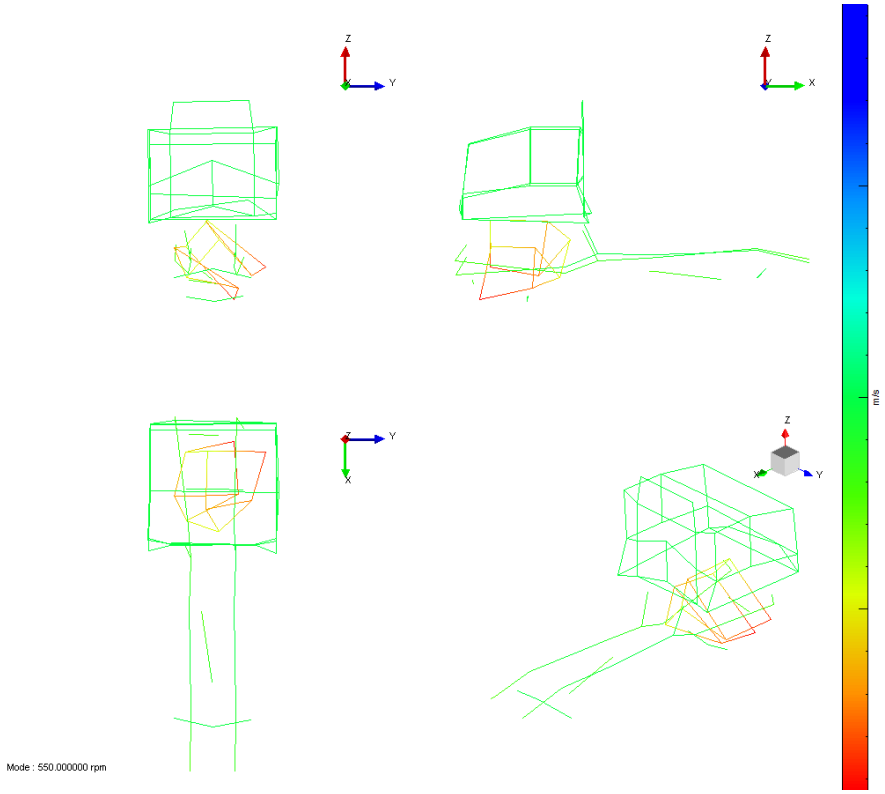
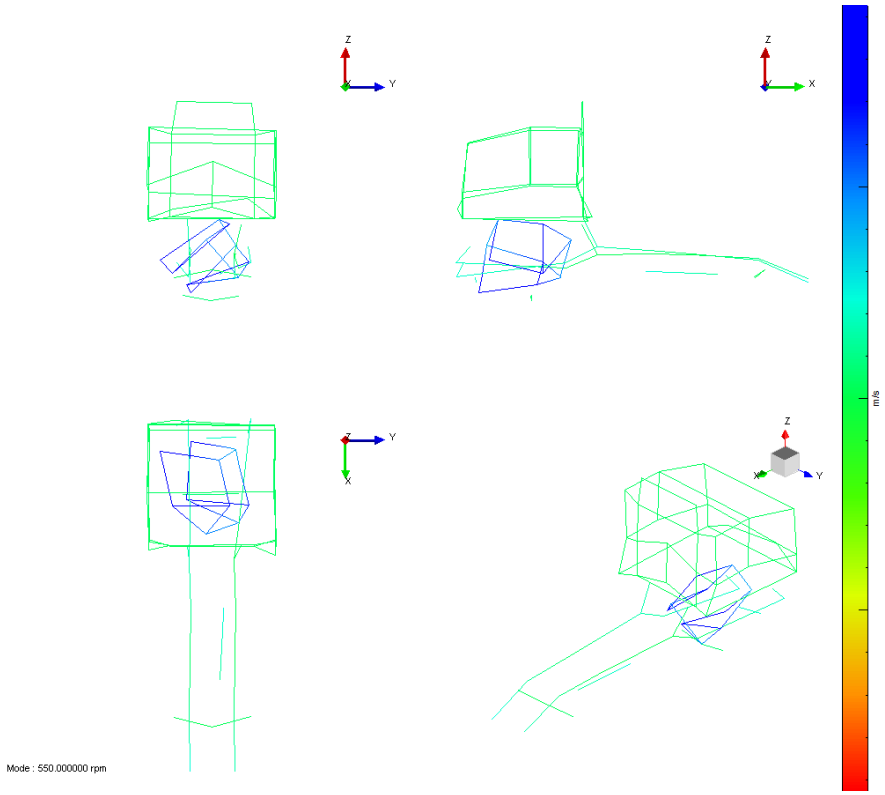


Figure 7.6: *Sum of transmissibilities for the first engine order*



(a) EO3, phase 0 °



(b) EO3, phase 180 °

Figure 7.7: *EO3 response at 550 rpm*

7.2.3 Vibration levels along transfer paths; Engine to Cab seat

The results from the ODS showed that EO1 leads to deflection on both chassis and cab, therefore a comparison of the first engine order vibration levels on the engine, chassis and cab was put together. It related the vibration isolation of EO1 through four different transfer paths, the closest transfer path between engine and driver seat. The four selected transfer paths and their specific measurement positions were displayed in Table 7.1. In Figure 7.8 to 7.11 the four paths are also displayed geometrically, the red markers and the red line show the specific measurement positions.

TP 1 - Front Left	TP 2 - Front Right	TP 3 - Rear Left	TP 4 - Rear Right
Cab:Seat1	Cab:Seat1	Cab:Seat1	Cab:Seat1
Cab:FEC1	Cab:FEC2	Cab:BEC1	Cab:BEC6
Fram:LeCabSusp1	Fram:RiCabSusp1	Fram:LeCabLeg1	Fram:RiCabLeg1
Fram:LeBe6	Fram:RiBe6	Fram:LeBe5	Fram:RiBe5
SubF:FrLeEngiMt	SubF:FrRiEngiMt	Fram:ReLeEngiMt	Fram:ReRiEngiMt
Engi:FrLeMt	Engi:FrRiMt	Engi:ReLeMt	Fram:ReRiMt

Table 7.1: The selected transfer paths

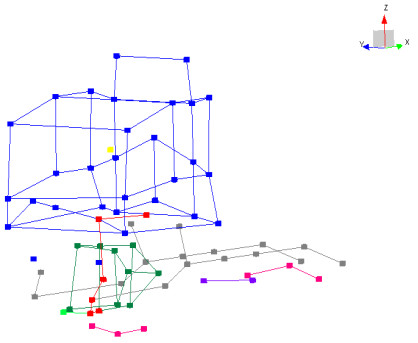


Figure 7.8: *Geometry TP 1, front left*

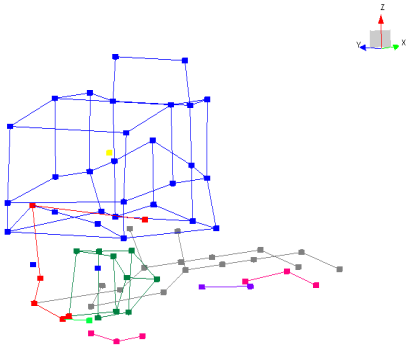


Figure 7.9: *Geometry TP 2, front right*

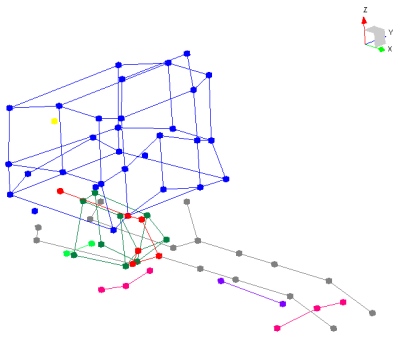


Figure 7.10: *Geometry TP 3, rear left*

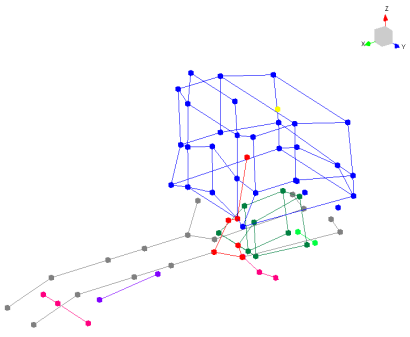


Figure 7.11: *Geometry TP 4, rear right*

Figure 7.12 to 7.15 display the vibration isolation of EO1 in the four transfer paths. In all four transfer paths the engine mounts isolates the EO1 vibrations but in the level increases at the left and right cab leg position and in the driver seat. The increase in level in this two positions

along with that the cab positions display almost equal level at the driver's seat implied that the cab might be resonant in this frequency range. Comparing the Y-direction showed that more or less non vibration isolation was achieved at the engine mounts or in the cab suspension which meant that the system has a strong coupling, i.e. the EO1 vibrations in Y-direction were not attenuated along the transfer paths. In Z-direction there were almost no isolation through the front engine mounts but at the rear engine mount the isolation was significant. The cab suspension did not provide any isolation in the Z-direction why the level in the driver's seat was dominated by the vibrations passing through the front engine mounts.

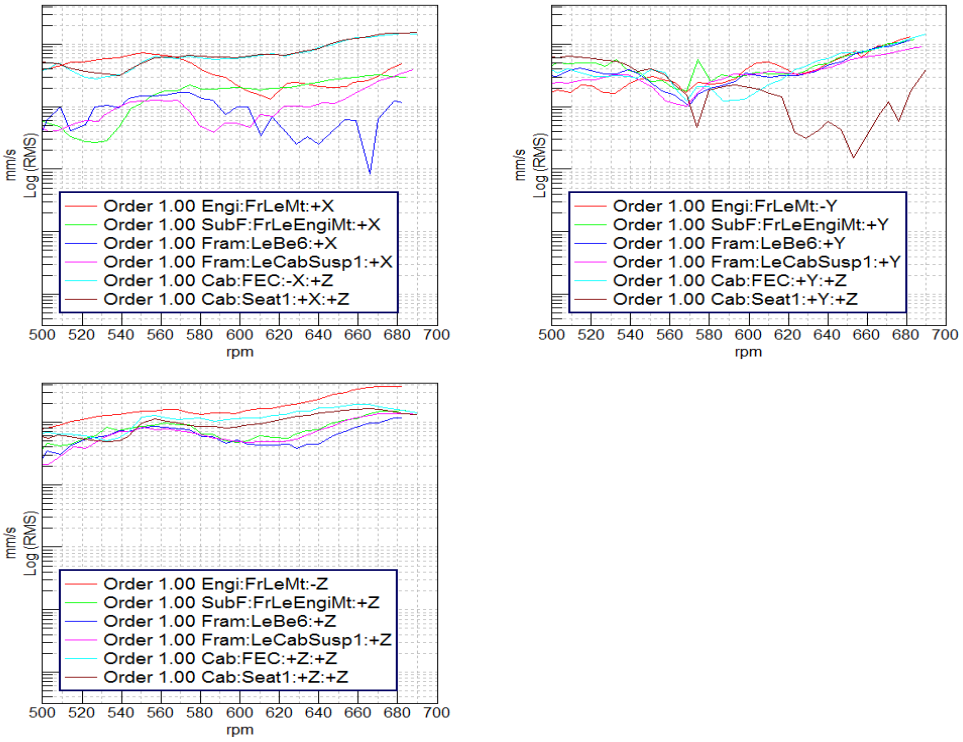


Figure 7.12: *TP 1, front left*

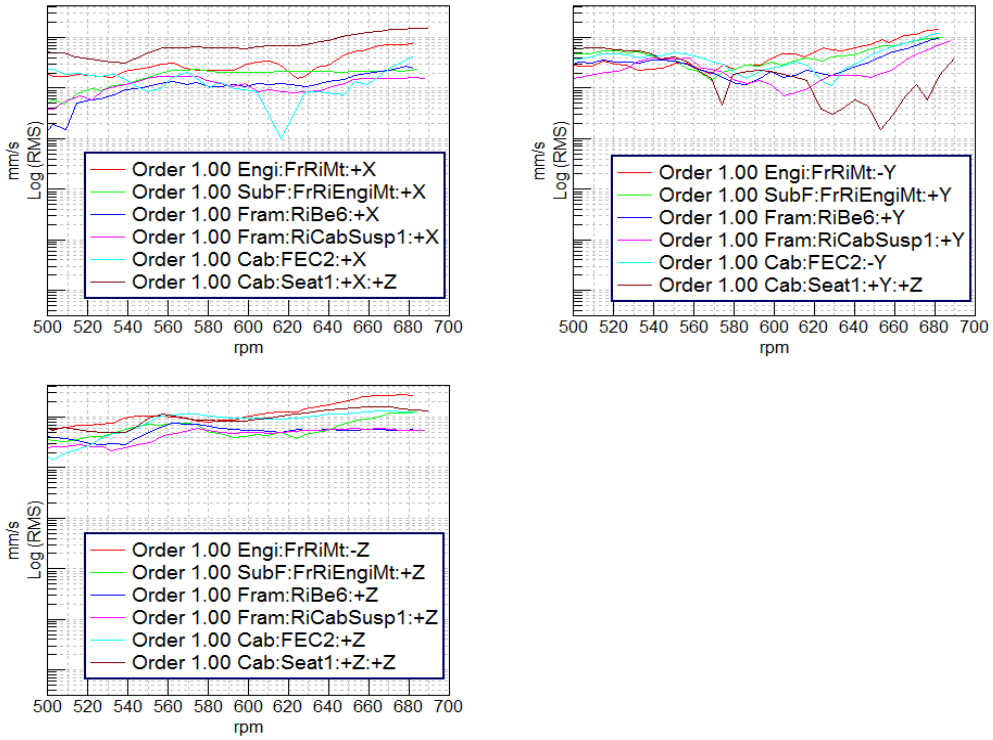


Figure 7.13: *TP 2, front right*

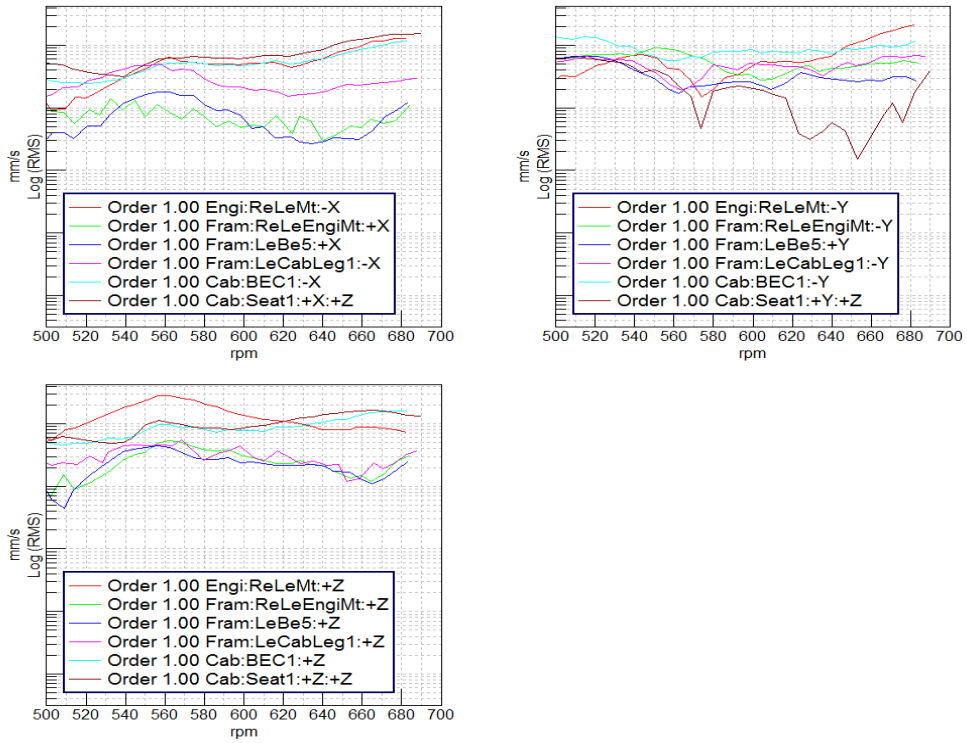


Figure 7.14: *TP 3, rear left*

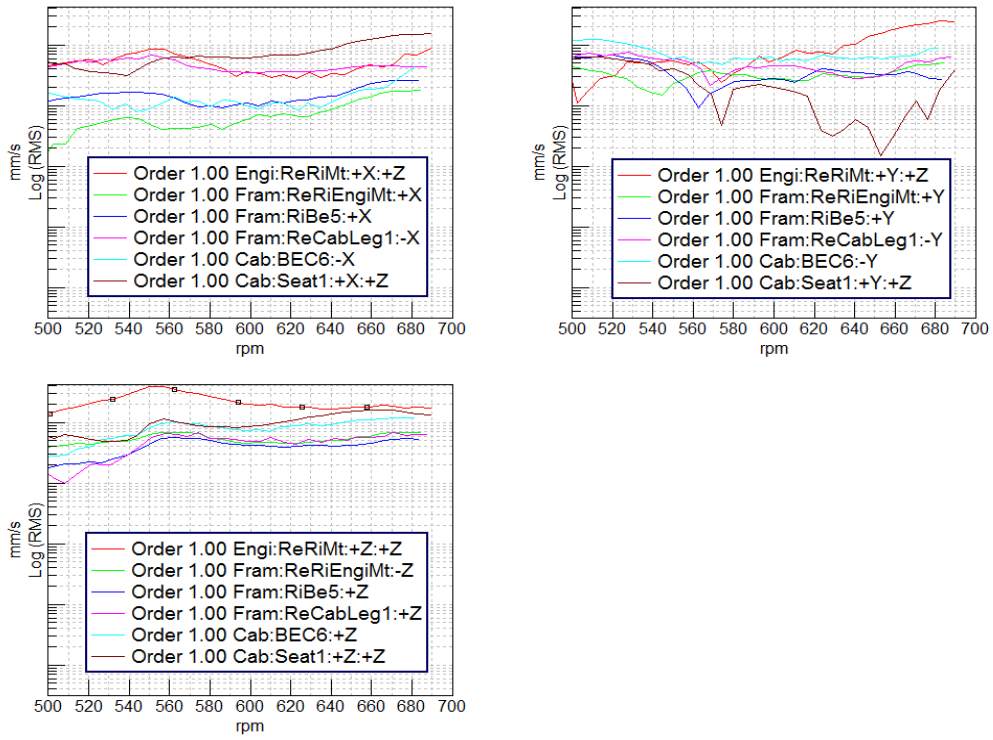


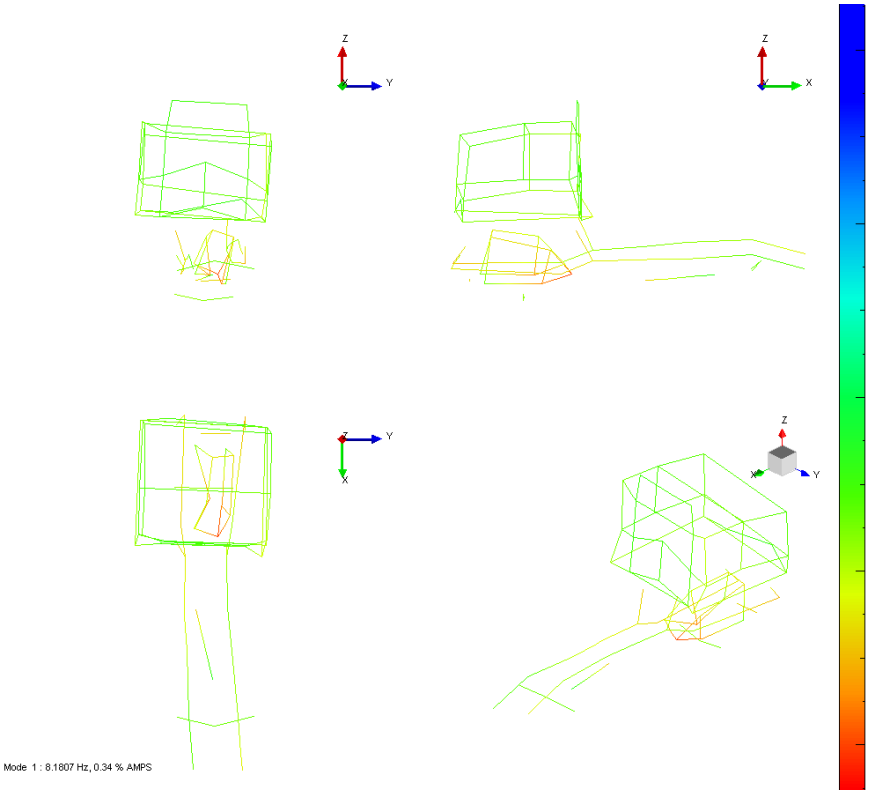
Figure 7.15: *TP 4, rear right*

7.2.4 Operational modal analysis

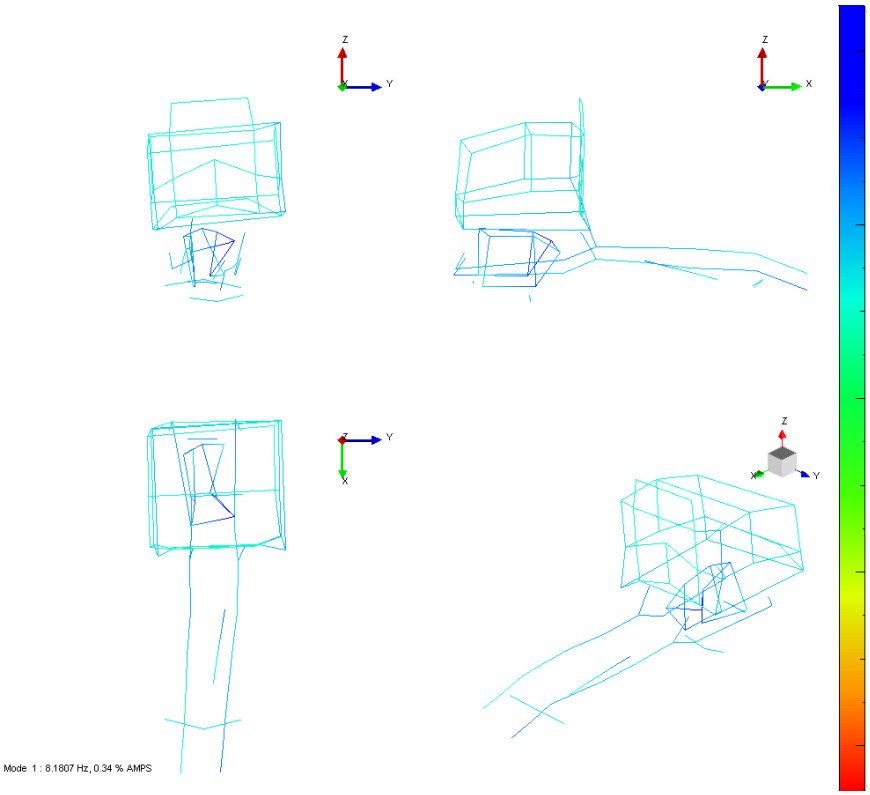
The same measured operational data used in the ODS was also used in the OMA. To distinguish the different modes for these particular boundary conditions LMS Test.Lab had a specific add-in for this analysis and it was combined with LMS Test.Lab's modal parameter estimator PolyMAX. The combination gave a clear stabilisation sheet constructing the synthesised model of the truck and its response to the rpm-sweep.

In total ten modes could be distinguished in the OMA in the frequency range 8 to 17 Hz. All modes and their shapes are displayed in the appendix and only the most important are posted here together with the Auto-MAC-matrix. The Auto-MAC-matrix was used to validate the result from the synthesised model and related how unique the modes were compared to each other. The result with ten modes gave the best MAC-matrix and also each synthesised FRF had a relative small error and large correlation compared with the measured, aiming for less than 10 % error and above 90 % correlation.

The two modulation frequencies, 0.91 Hz and 2.31 Hz, gave a good estimation of which modes that could influence the beating. In the OMA two modes were found close to the first engine order at 550 rpm. The first one deviated 1 Hz and the second one 2.1 Hz, from EO1 at 550 rpm, they are displayed in Figure 7.16 and 7.17. In Figure 7.18 the OMA Auto-MAC-matrix is showed.

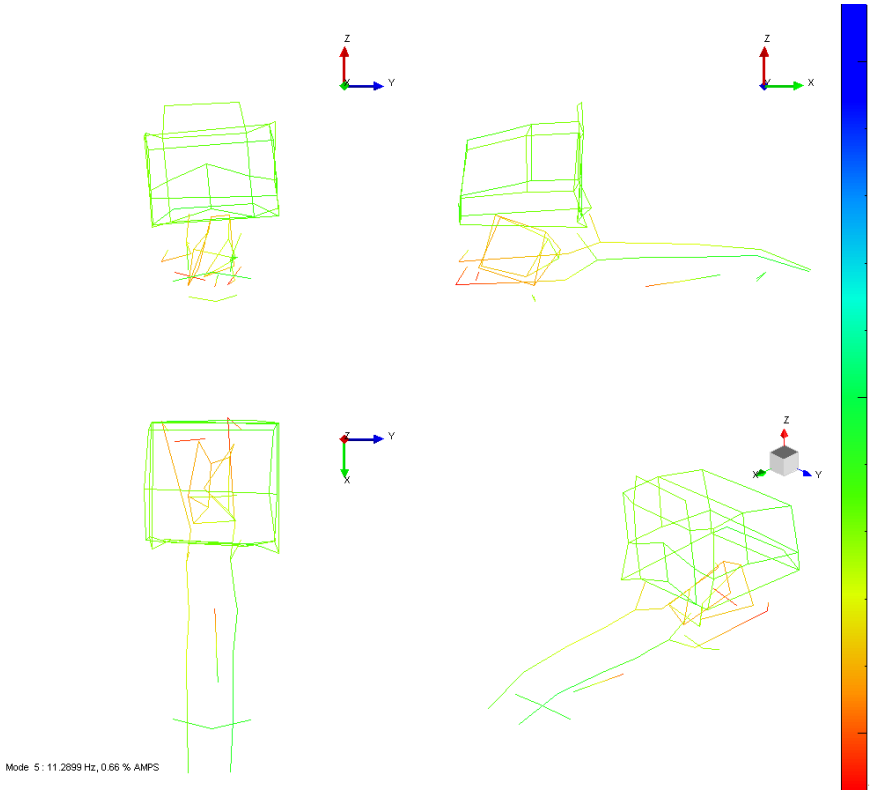


(a) Mode one OMA, 8.2 Hz phase 0 °

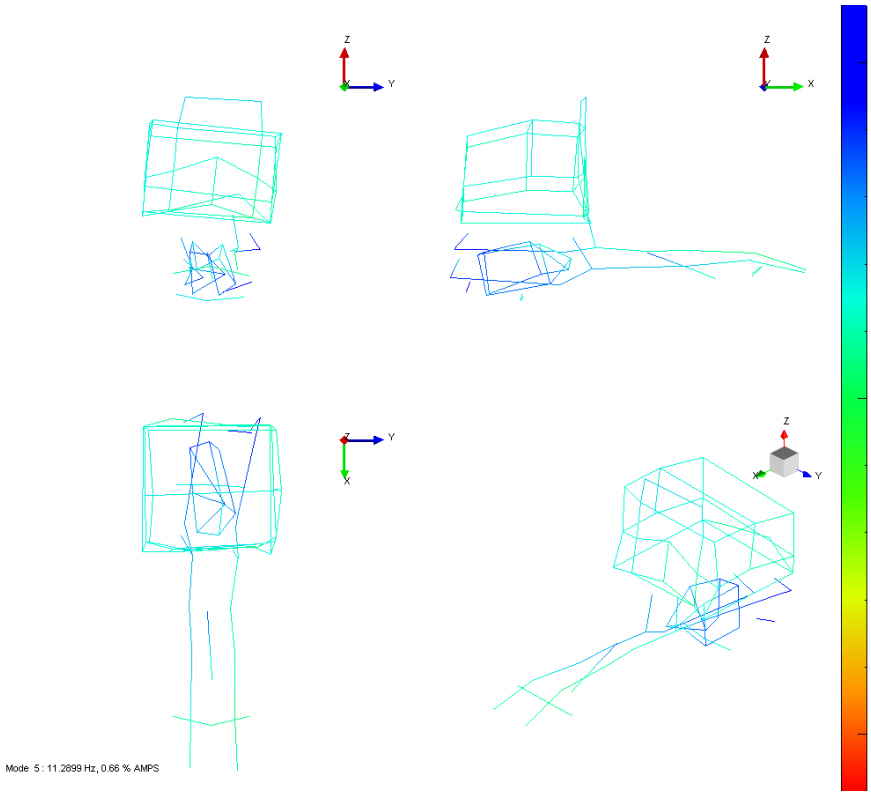


(b) Mode one OMA,8.2 Hz phase 180 °

Figure 7.16: *Mode one, 8.2 Hz, Engine roll and yaw (small). Cab roll (small)*



(a) Mode five OMA, 11.3 Hz phase 0 °



(b) Mode five OMA, 11.3 Hz phase 180 °

Figure 7.17: *Mode five, 11.3 Hz, Engine pitch*

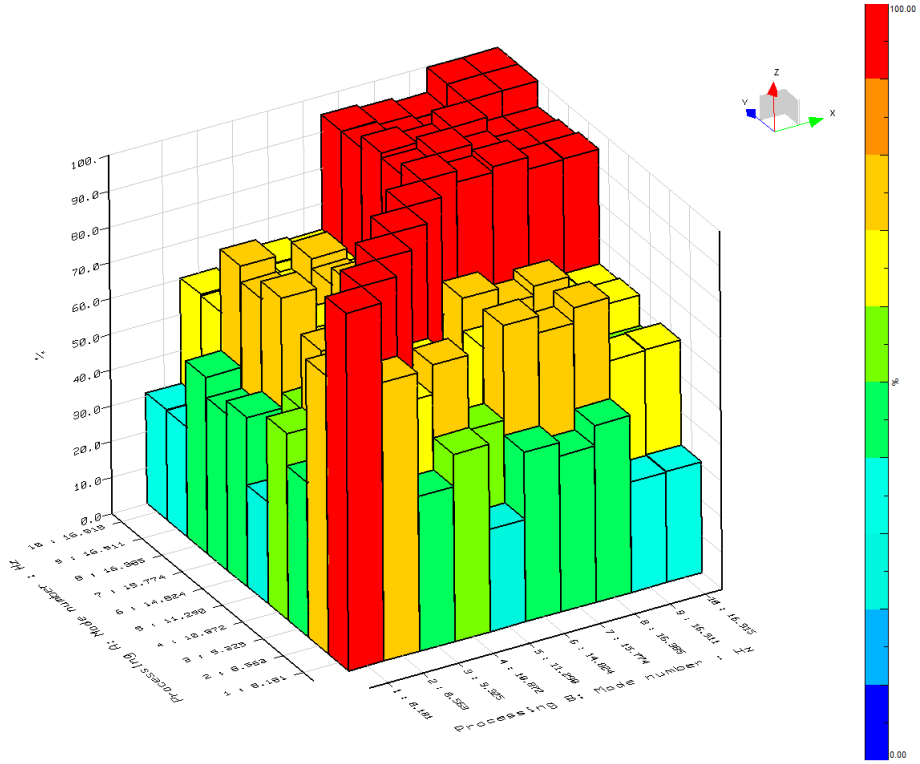


Figure 7.18: *MAC-matrix for the OMA mode set*

7.2.5 Discussion operational measurement

The results in Section 7.2.2 show that the deflection of the different subsystems of the truck was greater for EO1 than EO3. The reason for this is probably that the engine mounts could not isolate these low frequencies. At idle rpm EO1 was equal to 9.2 Hz and EO3 equals 27.6 Hz. The simulation of only the engine itself showed that the rigid body modes of the engine was spread out between 6 to 17 Hz, which also strengthens the theory of the rather low isolation of the first engine order through the engine mounts.

It can be seen in Figure 7.14 that the vibration level in x-direction increased between Fram:LeBe5 and Fram:LeCabLeg1 which implies that the cab anchorage was sensitive to first engine order vibrations in the x-direction.

Looking at the first engine order response of the truck, Figure 7.6 the peak at 557 rpm which implies that the behaviour might be resonant.

In total ten modes were found in the OMA and two of them corresponded quite well with the two different modulation frequencies found earlier. The MAC-matrix showed that the found modes are quite unique, not 100 % but they are clearly separated from each other.

One should bear in mind that the damping in a complete truck is very high. Therefore the frequency response for this type of excitation is quite flat compared to, if only a decoupled cab structure or chassis had been excited.

8 Modal measurements

Modal analysis measurements were carried out using the so called road simulator rig, which basically is an arrangement of several hydraulic pillars that the truck is placed onto. The truck stood with its four wheels on large concave steel plates, see Figure 8.1 and 8.2, that was attached to the top of the hydraulic pillars. The steel plates allows for some movement in the lateral direction. The pillars were then exciting the structure in a certain manner, in this case sinusoidal sweeps. To control the output signal one can either chose to control the output force or the displacement of the pillar. Displacement control was chosen for this setup, the decision was based on the information and knowledge of the rig operator that the rig performs best in this configuration.

To achieve a good excitation of the truck the parking brake was released, knowledge from the rig-operator. This let the wheel axles move freely in all directions compared to having the parking brake engaged. The truck was not secured to the rig in any way more than its own weight and the concave steel plates that make sure that it did not roll of. This ensured a good excitation and boundary conditions similar to the reality. However, this limited the acceleration amplitude that could be used for the excitation. Too high acceleration will cause the wheel to lose contact with the steel plate and at that moment it is no longer a sinusoidal excitation. This also limited the amplitude of the displacement and the sweep rate used. All these three parameters were tuned in before the measurements. Looking at the signal of the displacement and the force revealed if the wheel had lost contact with the plate. Another way would be to manually check, both by visual checks and listening to the excitation; if it loses contact it starts so clatter.

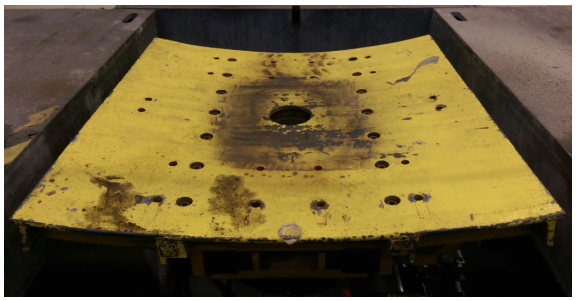


Figure 8.1: *Front wheel plate*

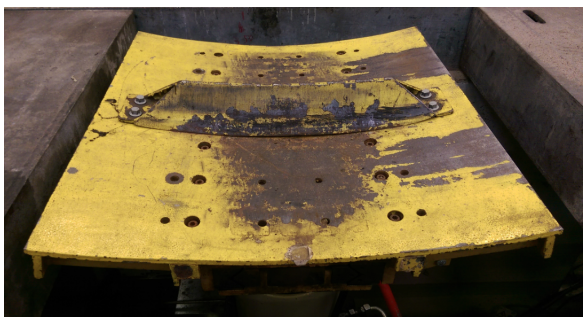


Figure 8.2: *Rear wheel plate*

8.1 Methodology - Modal analysis

8.1.1 Excitation - Sinusoidal sweep

The excitation of the truck was done by the hydraulic pillars as described above and the signal type was sinusoidal sweep between 5 and 20 Hz. White noise was also tested as excitation signal but was not able to input enough energy to the structure which lead to a bad frequency response function and low coherence. The frequency range was limited by two major factors;

The lowest possible frequency was 5 Hz due to that the control of the output signal in Test.Lab was 5 Hz. The upper limit was set by the rig itself and the hydraulic pumps, because there is a trade off in frequency range, displacement and input force.
In addition to the limit of maximum input force in the excitation, described earlier in Section 8 the force was also limited due to the chosen sweep rate, [Hz/s]. Too high sweep rate also cause the wheel to lose contact with the steel plate. A narrow frequency range of the sweep was easier to configure with respect to achieving a good sinusoidal sweep. Therefore the upper limit was set to 20 Hz, which also corresponds well with the specified upper limit from the prerequisites.

8.1.2 Data acquisition

The sinusoidal sweep was set from 5 to 20 Hz with a sweep rate of 0.25 Hz/s. It was equally important in this measurement that the resolution was good for the analysis why it was set to 0.1 Hz. The sampling frequency was 100 Hz and each block was then equal to 10 seconds. The total length of the sinusoidal sweep is 60 seconds and the acquisition settings were made to fit an even number of block into the total sweep time, in this case six blocks. In total five sweeps per setup were measured and used to achieve a stable averaged FRF.

A selection of measurement positions and their corresponding FRFs and coherences are shown in Figure 8.3 and 8.4, position Cab:Seat1, Fram:CabLeg1 and Engi:ReRiMt. They display the general shape of the FRF:s, i.e mostly damped and with a few anti-resonances, the anti-resonances correlates with the dips in the coherence curves, which is correct since low acceleration in the structure leads to bad coherence.

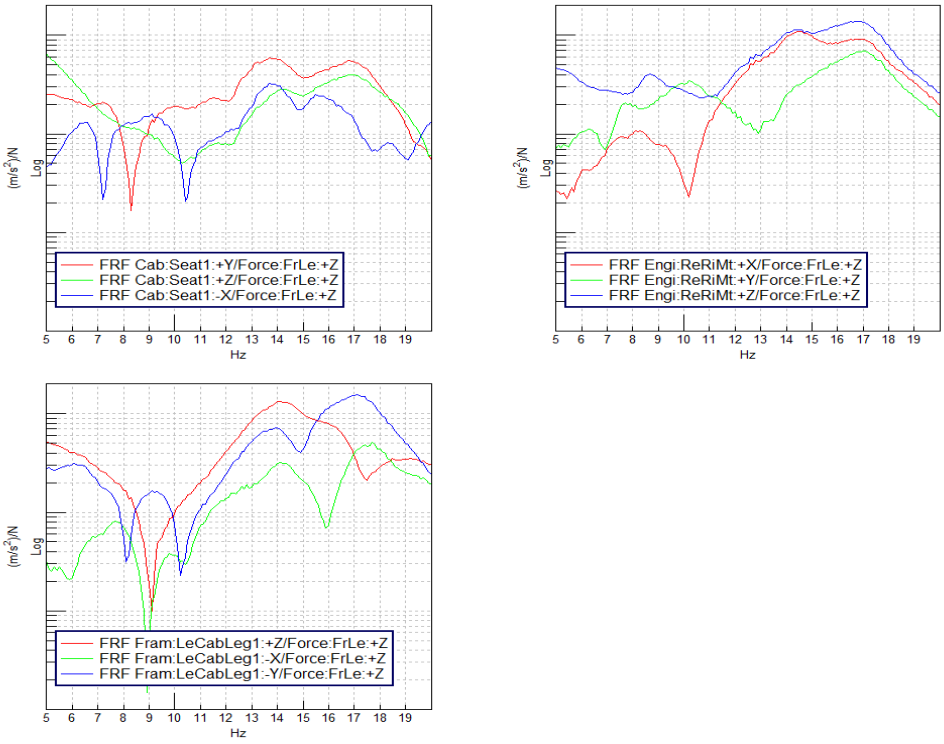


Figure 8.3: *FRF-curves for engine, chassis and cab positions*

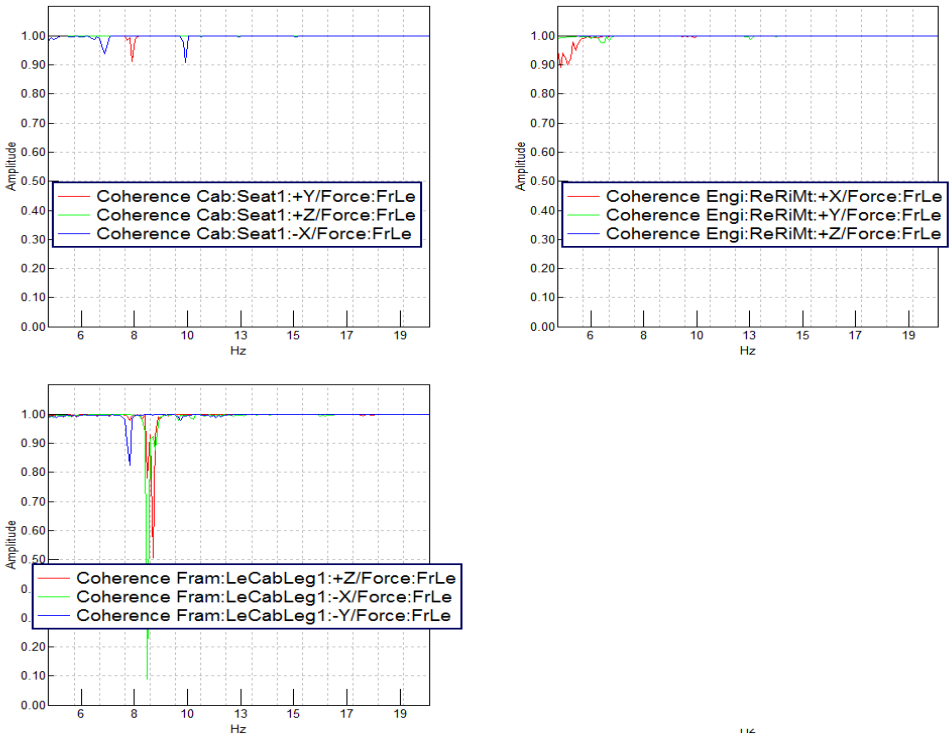


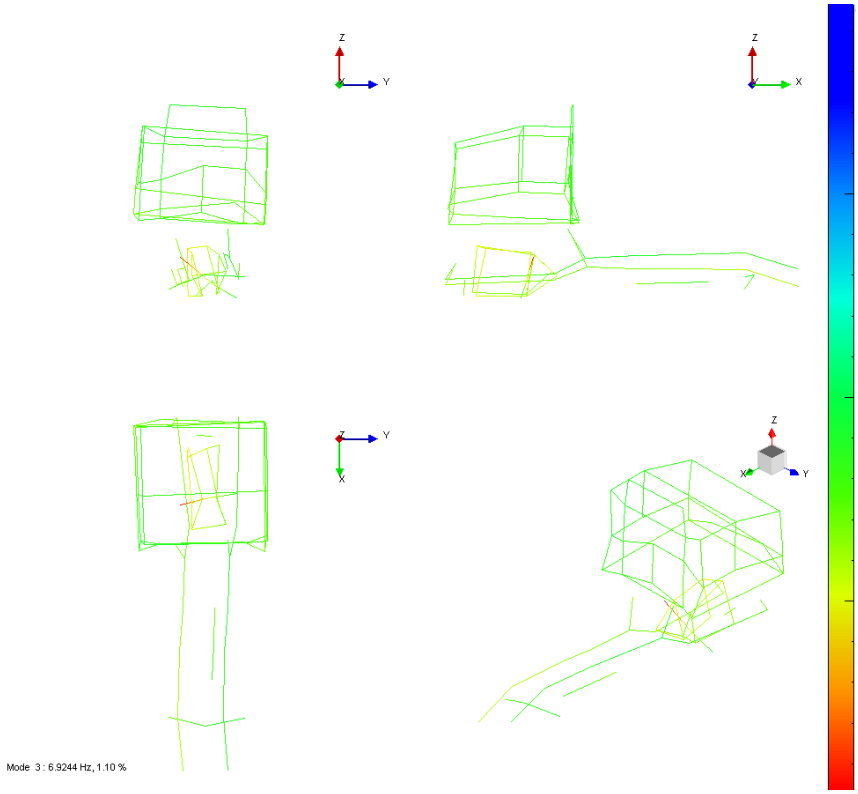
Figure 8.4: *Coherence-curves for engine, chassis and cab positions*

8.2 Results Modal measurements

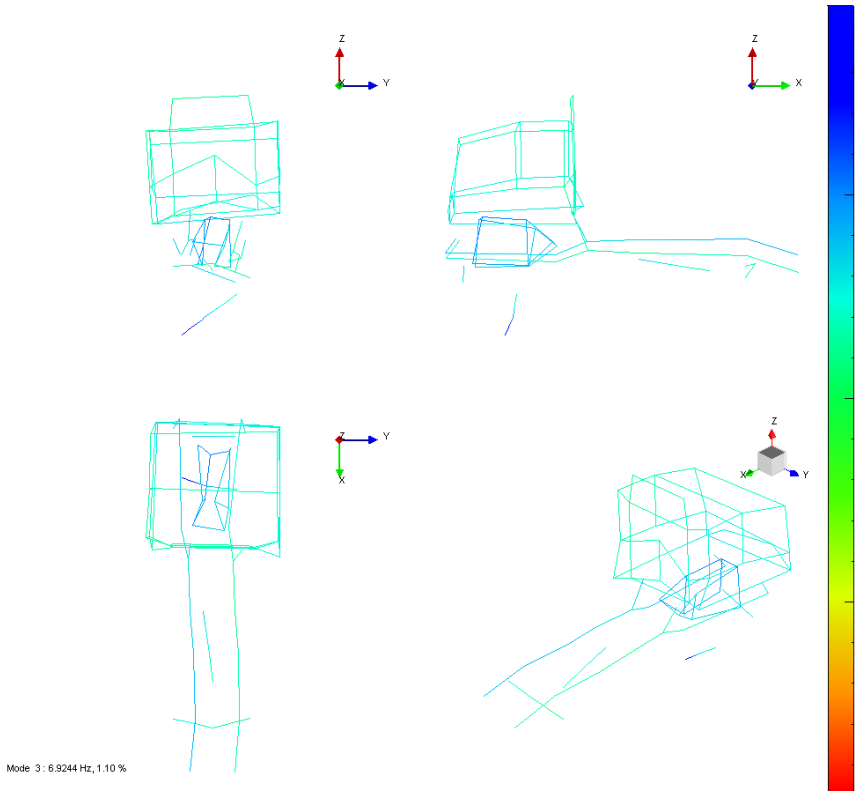
The result of the modal analysis was directly comparable to the simulated modal analysis, presented in Section 5. Both methods displayed the resonant behaviour of the truck without the engine as a driving source, which was the case in the operational analyses.

In total 12 modes were found in the modal analysis. They were a combination of rigid body modes (RBM) and flexural modes for the different subsystems. For the engine the three different rotational RBMs were all fund; roll, pitch and yaw, however the three translational; Surg, Sway and bounce, were not located. Also at the cab three RBMs were excited; roll, yaw and sway, a mixture of rotational and translational RBMs. Looking at the front wheel axle the roll and surging modes were found. In the chassis four flexural modes were located and no RBMs. The flexural modes were oriented both in the vertical and horisontal direction.

Comparing the resonance frequencies of the 12 modes and the deviation from EO1 gave in total four modes that correlated well with the two modulation frequencies. They were mode no. 3, 4, 6 and 7. No. 3 and 7 corresponded to modulation frequency 2.31 Hz and no. 4 and 6 to 0.91 Hz. The modes are displayed in Figure 8.5, 8.6, 8.7 and 8.8.

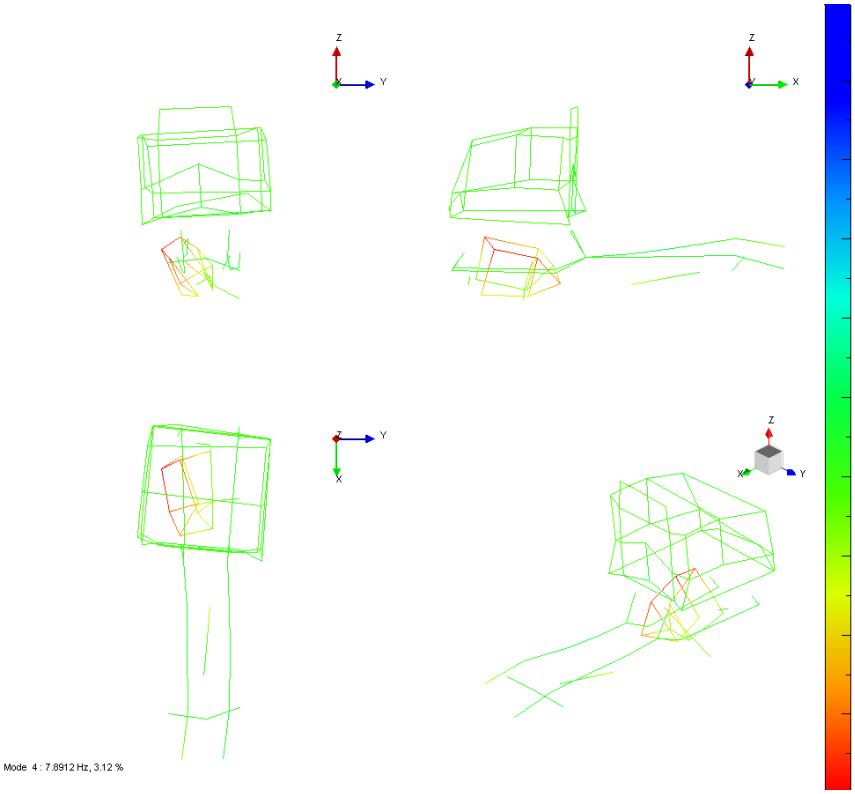


(a) Mode three, 6.9 Hz, phase 0 °

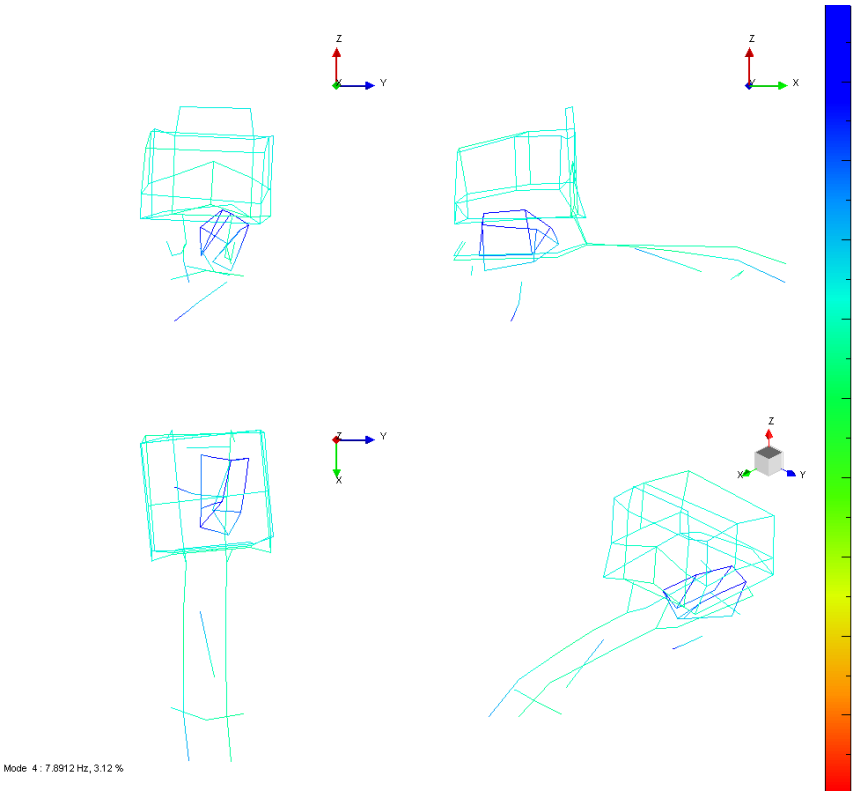


(b) Mode three, 6.9 Hz, phase 180 °

Figure 8.5: *Modal mode three, 6.9 Hz, cab yaw engine close to bounce*

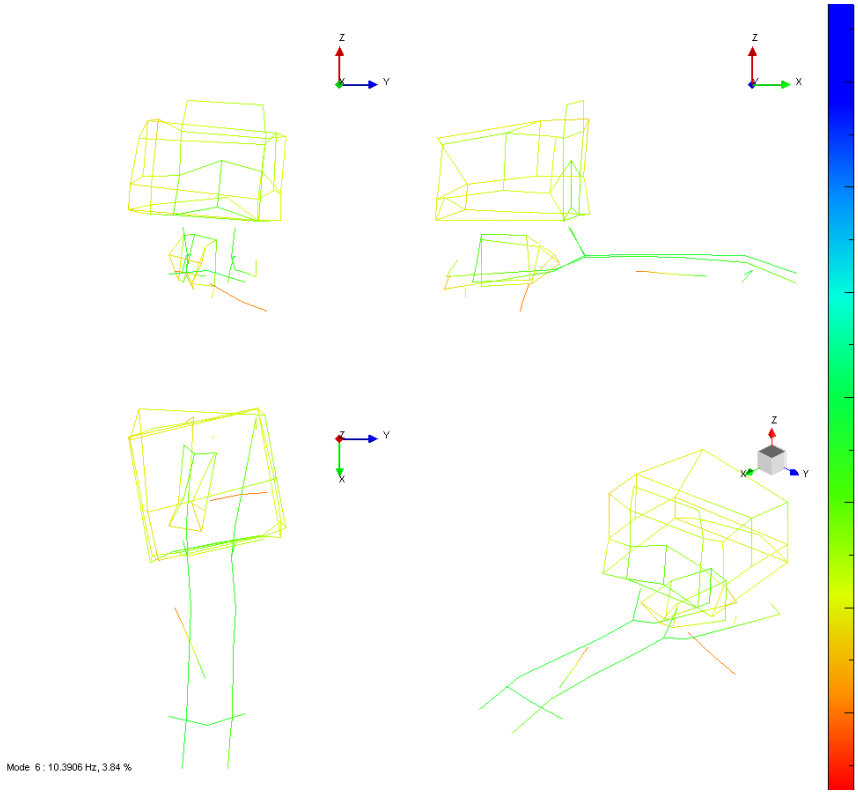


(a) Mode four, 7.9 Hz, phase 0 °

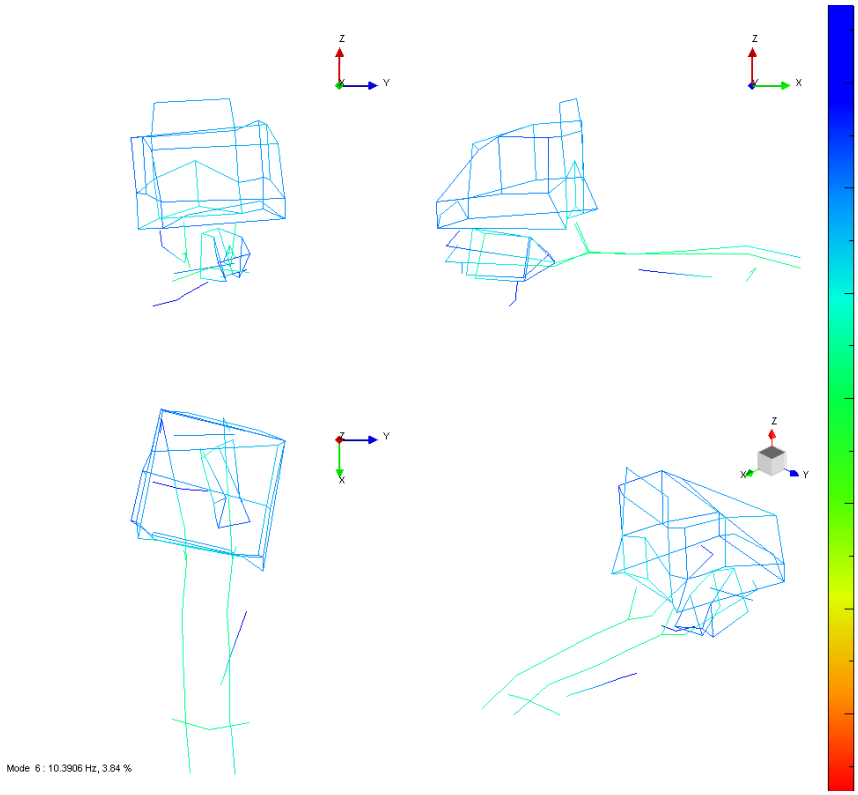


(b) Mode four, 7.9 Hz, phase 180 °

Figure 8.6: *Modal mode four, 7.9 Hz,Cab roll*

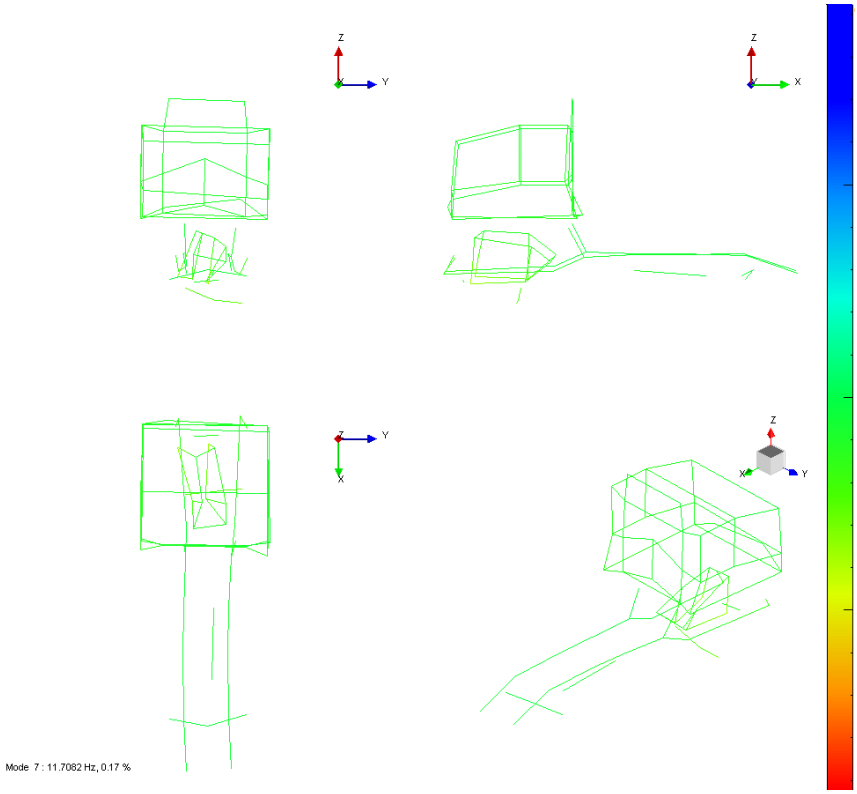


(a) Mode six, 10.4 Hz, phase 0 °

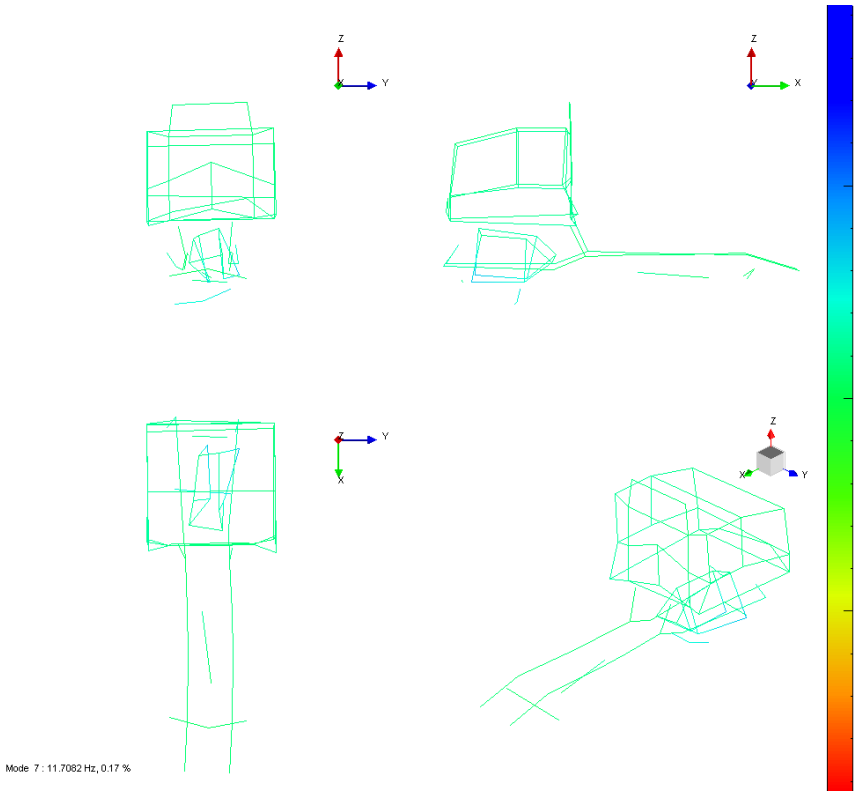


(b) Mode six, 10.4 Hz, phase 0 °

Figure 8.7: *Modal mode six, 10.4 Hz, Engine yaw, cab yaw (out of phase)*



(a) Mode seven, 11.7 Hz, phase 0 °



(b) Mode seven, 11.7 Hz, phase 0 °

Figure 8.8: *Modal mode seven, 11.7 Hz, Front axle roll, engine roll (small)*

The quality of the selected mode set is presented in the Auto-MAC-matrix in Figure 8.9. One can clearly see that the modes were less dependent of each other in the modal analysis compared to the OMA, except for the first three modes, which were similar to each other. Independent modes implied that the measurements together with the synthesised model was good and displayed a valid representation of the modal system.

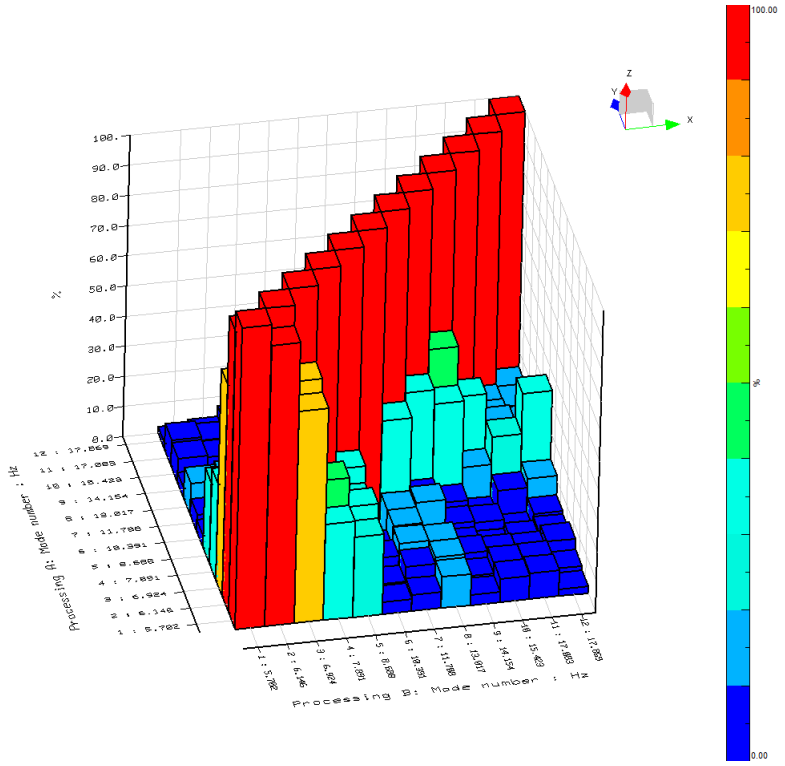


Figure 8.9: *Auto-MAC-matrix, Modal analysis*

A comparison between some synthesised and measured FRFs are plotted in Figure 8.10, 8.11 and 8.12 and show the quality of the synthesised model in the experimental modal analysis. Vertical lines in the figures show the location of the stable poles, i.e the modes in the system. Red curve is the measured FRF and green is the synthesised one. The overall aim was to achieve a correlation higher than 90% and error less than 10%, note that they should not add up to 100% together.

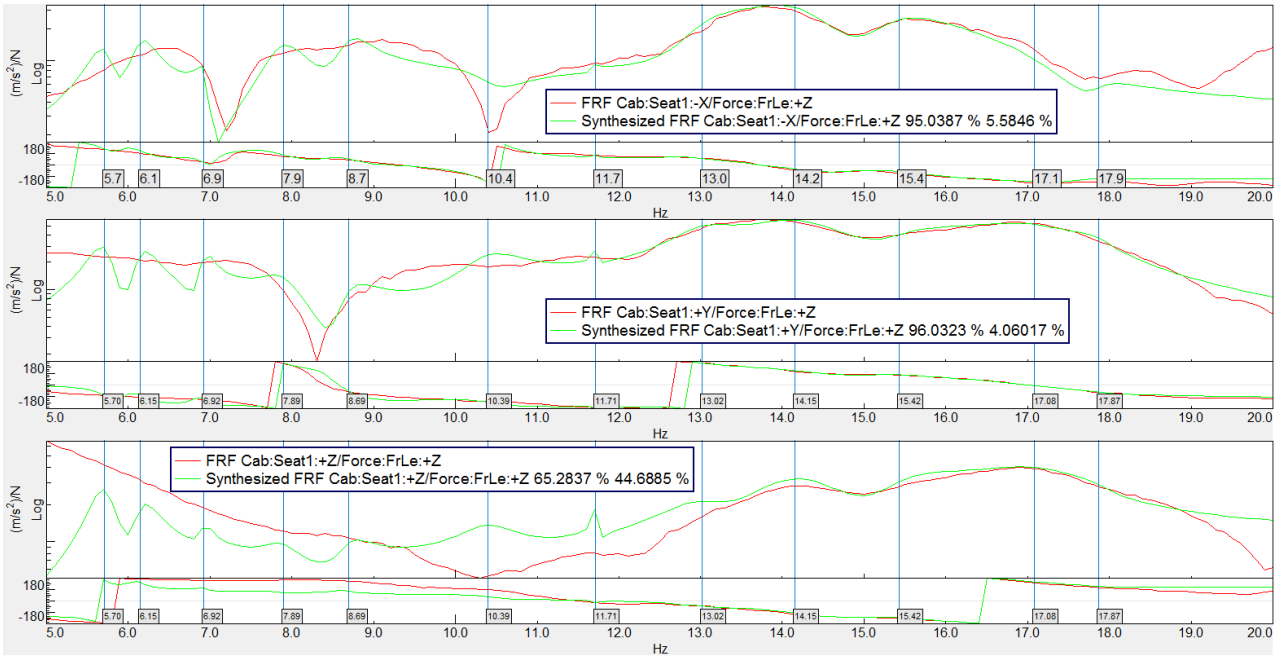


Figure 8.10: *Synthesis in Cab:Seat1*

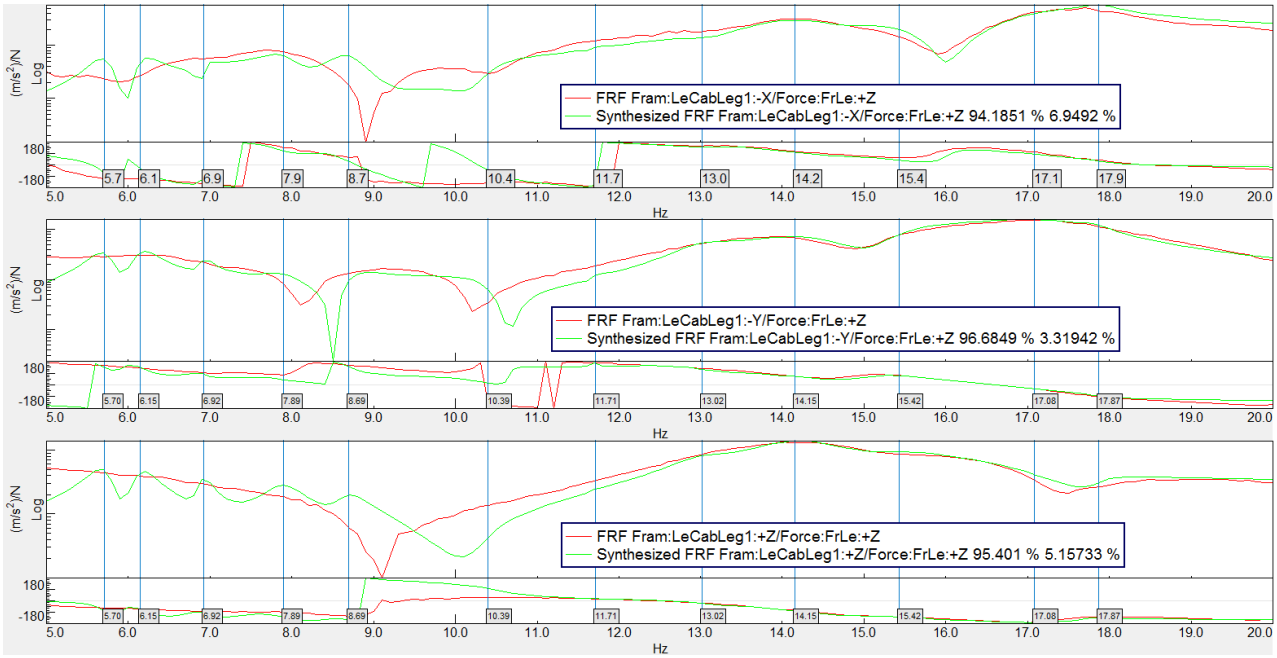


Figure 8.11: *Synthesis in Fram:CabLeg1*

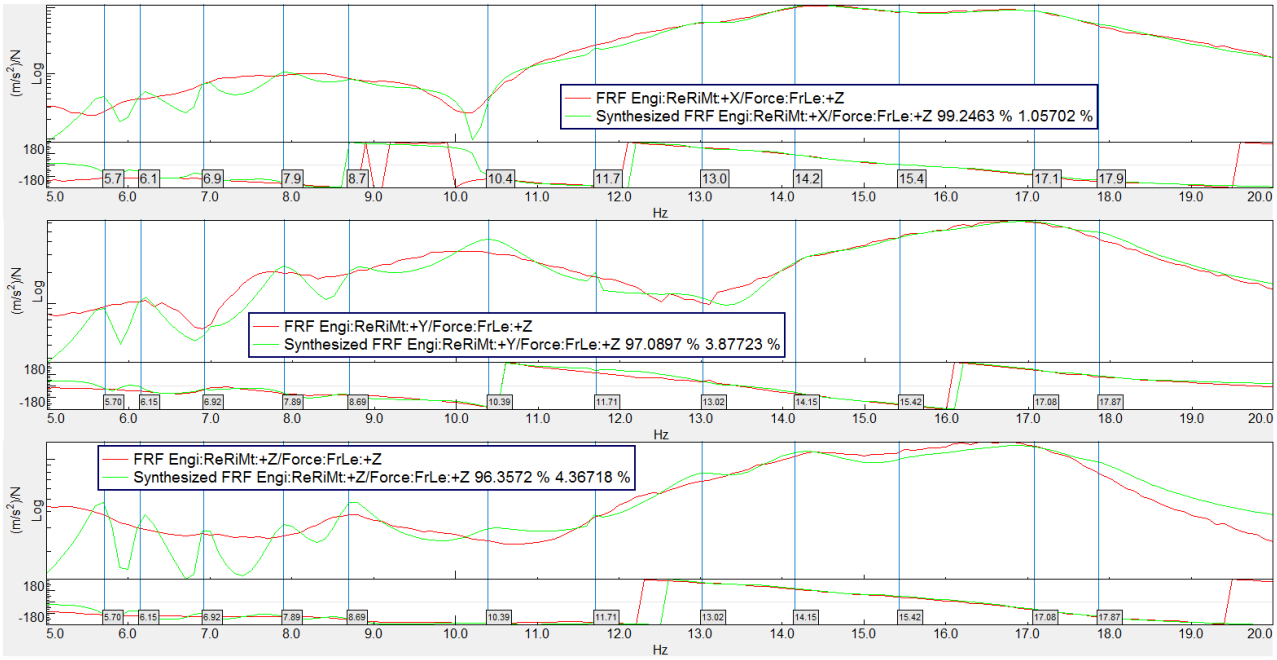


Figure 8.12: *Synthesis in Engi:ReRiMt*

8.3 Discussion modal analysis

The experimental modal analysis revealed that the simulated modes, presented in Section 5, were very similar to the modal analysis. The amount of modes found in both methods in the frequency range 0 to 12 Hz were more or less identical to each other, most of them also very close in frequency, 0.2 Hz, which was a satisfying deviation in an analysis of a complete truck. Above 12 Hz the experimental modal analysis missed some modes, mainly flexural modes of the chassis, compared with the simulated results. This could be due to the type of excitation of the truck in the modal analysis. Neither of the two methods could be seen as the absolute correct method. Both has their advantages and disadvantages. The experimental modal analysis is based on an real structure compared to the finite element model in the simulation; boundary conditions are more complex in the reality but are also the real ones. One obvious difference is the response of the truck's tires and their response was seen as linear in the model but in reality they are not linear. This might lead to overtones of the excitation frequency transferred to the structure. In total the simulated modal analysis resulted in 20 modes found, compared with the 12 modes found in the experimental modal analysis. However, within the most crucial range, 6 to 12 Hz, the experimental modal analysis and the simulated results were comparable, which was a satisfying result.

9 Mode-map

Table 9.1 displays the experimental analysis methods used in the project. In the left column the geometrical mode shape is listed and in the following columns the specific frequency of each mode is stated, if the mode type is found in the analysis. The advantage of the mode-map is that the same geometrical mode shape and frequency can be compared between the different analysis methods used, which show the difference between methods but also their correlation. OMA, operational modal analysis and EMA, experimental modal analysis. The results of the simulated analysis cannot be published with regard to secrecy.

Mode shape	**OMA [Hz]**	**EMA [Hz]**
Engine bounce	-	6.9
Engine sway	8.6	-
Engine surg	16.4	-
Engine yaw	-	10.4
Engine roll	8.2; 14.8; 15.8; 16.4; 16.91; 16.92	5.7
Engine pitch	9.3; 10.9; 11.3	8.7
Cab bounce	-	-
Cab sway	-	6.1; 13.0
Cab surg	-	-
Cab yaw	-	6.9; 10.4
Cab roll	14.8	7.9
Cab pitch	-	17.9
FrWhAx roll	-	11.7
FrWhAx surg	-	17.1
FrWhAx sway	-	-
FrWhAx bounce	-	-
Chassis torsional	9.3	8.7
Chassis bending vertical	-	13.0; 14.2; 17.1
Chassis bending horisontal	-	8.7; 14.2; 15.4; 17.9
ReWhAx yaw	-	-
ReWhAx bounce	-	-

Table 9.1: Mode map, all used methods put together

10 Discussion

There is always an uncertainty whether subjectively detected vibrations and phenomena like beating can be objectively quantified and coupled to each other with accuracy. However, a beating phenomenon was visible in the time signal of the measurement position on the seat base, Cab:Seat1. It was also most prominent in the x-direction which also was the case when one is subjectively feeling the beating. A small pushing effect in the back rest of the

seat, in x-direction. Not only one but two modulation frequencies were found in the analysis, which could explain the perceived difference in modulation frequency over time. These two frequencies repeated themselves over time regardless of temperature. It was the same when subjectively evaluating the room temperature situation. With a warm engine the spectrum was clearer and also more stable. Since the different modulation frequencies were found it is probably not only one but two resonances that that need to be located in the effort to find the reasons for the beating phenomenon in the cab seat vibrations. The modulation frequencies 0.91 Hz and 2.31 Hz were not clear harmonics of each other. Therefore it is most likely that it is two or more resonances that shall be found rather than one as the initial guess implied.

Hence, it was of interest to find resonances in the frequency region 6 to 12 Hz which corresponded to EO1 plus minus 2.31 Hz with some extra margin at the limits, which also covers the 0.91 Hz modulation frequency. In Table 10.1 each mode with its shape and frequency that corresponds to the above specified frequency range are listed. The table is a smaller version of the complete mode map. The results show that ten modes fit the smaller frequency range. The distinct frequencies that corresponded to the deviation to EO1 at idle, 9.2 Hz, are; 6.9, 8.3, 10.1 and 11.5 Hz. Those modes that match the modulation frequencies are written in boldface in Table 10.1.
The correlation between the experimental and simulated modal analysis was good, within the 6 to 12 Hz range the maximum deviation was 1.6 Hz. Comparing the operational modal analysis, the experimental modal analysis and the finite element modal analysis the last two, EMA and FE-Compl. truck, corresponded better to each other than the OMA did. This is due to that the engine when running is an energy source, in the OMA case, but in the EMA and FE-Compl. truck case, the engine is just a mass with springs and damping. They display two different behaviours, the resonant and the driven one.

Mode shape	**OMA [Hz]**	**EMA [Hz]**
Engine bounce	-	**6.9**
Engine sway	8.6	-
Engine yaw	-	10.4
Engine roll	**8.2**	11.7
Engine pitch	11.3	8.7
Cab yaw	-	**6.9**; 10.4
Cab roll	**8.2**	7.9
FrWhAx roll	-	11.7
Chassis bending	-	8.7
Chassis bending vertical	-	-
Chassis bending horisontal	-	8.7

Table 10.1: Smaller mode-map, representing the smaller frequency range 6-12 Hz

Both methods, operational and modal, were relevant since they displayed different behaviours of the structures, the driven and the resonant. They also displayed two different analysis methods. In the operational case the extracted modes and deformations were a relation between the reference position and other positions, transmissibilities. Compared with the modal case where the response was normalised to a known input force. It was therefore not surprising that the two methods displayed different mode shapes at different frequencies.

To minimise one of the non-linearities of the rubber in the engine mounts the temperature was monitored during the test, at the rear right engine mount, and the truck was pre-heated before all measurements. This should minimise the difference between the operational and modal measurements regarding the damping and stiffness properties of the engine mounts. Since this was believed to have an effect on the comparison of the different methods.

The modal validation of both EMA and OMA regarding synthesised FRFs compared with measured were satisfying for almost all nodes, except for a few positions on the cab and mostly in the OMA where the adaptation was poor. This was due to that the real FRFs did not contain clear modal peaks, but was rather flat, since the complete truck contained a lot of damping.

As shown in Section 7 the vibration level of EO1 was higher on the cab side of the cab anchorage than on the chassis side close to the cab anchorage. It was also clear that the cab anchorage had a higher deflection in the x-direction which was necessary if a yaw motion was to be present, i.e as several of the resonant modes close to EO1 had. The higher vibration levels could therefore be due to the modal behaviour of the cab. It also implied that the modal behaviour of the cab was due to the weakness in the cab anchorage. The 6.9 Hz resonance in the experimental modal analysis displayed the yaw motion of the cab and this resonance was also one that corresponded to the 2.3 Hz modulation frequency. By changing the stiffness of the cab anchorage the resonance frequency of this mode could be altered, either up or down in frequency. Lowering the resonance frequency would move the resonance further away from EO1 which might decrease the beating phenomenon.

The front wheel axle had a clear resonant behaviour at 11.7 Hz in the modal analysis but not found in the OMA. The reason could be that during the operational measurements the parking brake was applied, but in the experimental modal measurements the parking brake was released. This since the truck was placed on the concave steel, Figure 8.1 and 8.2, which prevented the truck from moving. In the operational measurements the engagement of the parking brake was needed to prevent the truck from moving. This changed the rotational properties of the wheels and by that the modal behaviour was altered.

The engine was the subsystem that was the easiest to excite in the modal analysis and where almost all RBMs were found. However, there was a clear difference between the modal and operational case where the engine had a very driven behaviour. The motion at idle rpm was dominated by the pitch and roll modes. The roll motion could be derived to EO3 because of the force transmission between the ignition and and the crankshaft. This was displayed in Figure 7.7 where the motion of EO3 is filtered from the rest. The pitch motion was not as clear as the roll but it depended quite much on EO1 as seen in Figure 7.5. One should also remember that this was transmissibilities, which is a relation between the acceleration measured at the truck. The resonant behaviour can still be vital even though its modes was not visible in the OMA. Therefore the two different methods complement each other well and provide useful information.

11 Conclusion

It was found that the subjectively determined beating in the cab seat could be measured and that the behaviour, regarding coordinate direction, was the same. The beating was strongest in x-direction. Not only one but two modulation frequencies were found, 0.91 Hz and 2.31 Hz, see Section 7.2.1.

As reported in Section 9 the different methods used in the project have together shown that there existed modes in the truck's different subsystems that correlated with the two modulation frequencies found. In total ten different mode shapes fit the 6 to 12 Hz interval and five of those; engine bounce, engine roll, cab yaw, cab roll, chassis bending horisontal, match the modulation frequencies. It was therefore at this stage not possible to specify one specific mode that was most responsible for the beating phenomena without further investigations. There was also the possibility that they were all needed to create the beating.

From the comparison between the simulated and experimental modal analysis it could be said that the correlation between FEM and EMA was good. In the 0 to 20 Hz range the maximum deviation for one mode was 6.6 Hz but in the 6 to 12 Hz ,which was the most interesting, the deviation was only 2.3 Hz. Several of the modes deviated less then 1 Hz in both ranges and an exact match was also found in a couple of cases. By those results the performed experimental modal analysis was concluded to output reliable mode shapes and frequencies.

The results are limited to the specific test truck used in this project and cannot be applied on a general truck. In general, beating phenomenon can appear when one or several subsystems, containing a number of resonances, are coupled and subjected to a vibrational force. In the project only one loop of measurements has been carried out and therefore no attempts to change the modal behaviour of, for example the cab anchorage, has been performed.

12 Future work

The project resulted in a number of conclusions but also raised new questions that could not be answered by the measurements and analysis conducted. In the future it would be interesting to evaluate a design change of the cab anchorage to confirm the conclusion of that the cab yaw mode at 6.9 Hz is responsible for the beating with 2.3 Hz modulation frequency. This could be done by either stiffen the cab anchorage or make it weaker in order to change the resonance frequency. It would also be interesting to further analyse the vibration pattern in the driver seat more in detail and try to determine why the vibration level of EO1 is increased in the driver seat position in the operational measurements.

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A Measurement positions & Geometry

Engi	Engine
Cab	Cab
Fram	The two major chassis beam
WhAx	Wheel Axle
StWh	Steering Wheel
Shaft	Cardan shaft
SubF	The crossbeam under the front of the engine, where the two front engine mounts are connected
Le	Left
Ri	Right
Fr	Front
Re	Rear
Be	Beam
Mt	Mount
LEC	Left End Cab
REC	Right End Cab
FEC	Front End Cab
BEC	Back End Cab

Table A.1: List of abbreviations used in the geometry to name the measurement positions

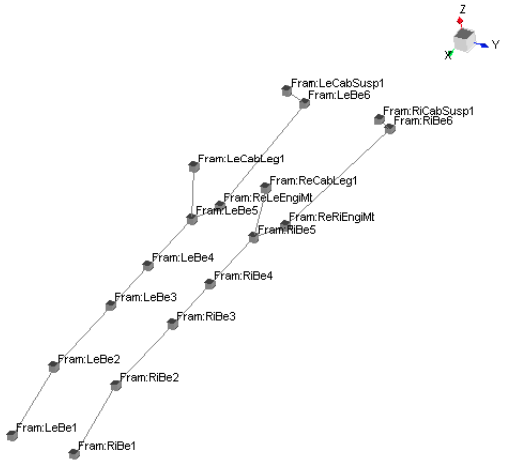


Figure A.1: *Measurement positions on the chassis*

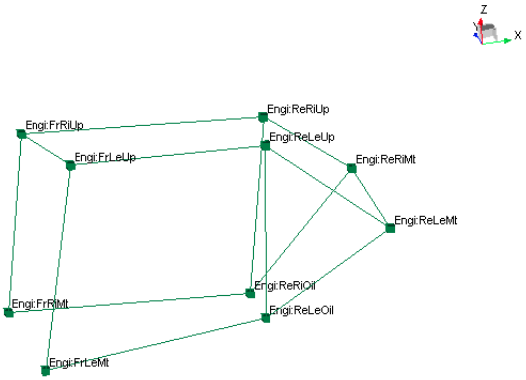


Figure A.2: *Measurement positions on the engine*

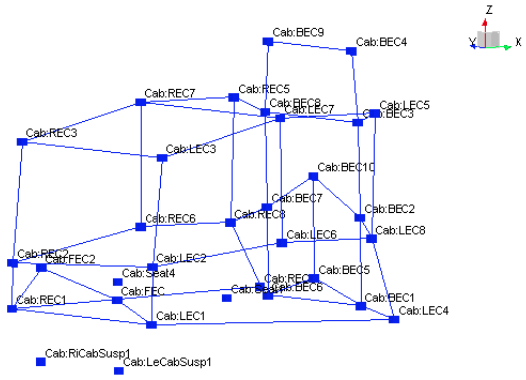


Figure A.3: *Measurement positions on the cab*

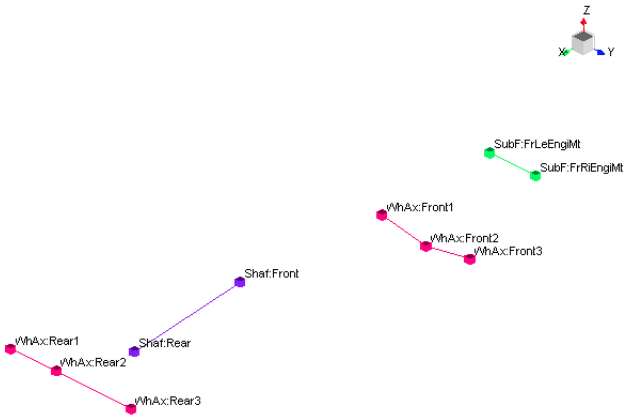
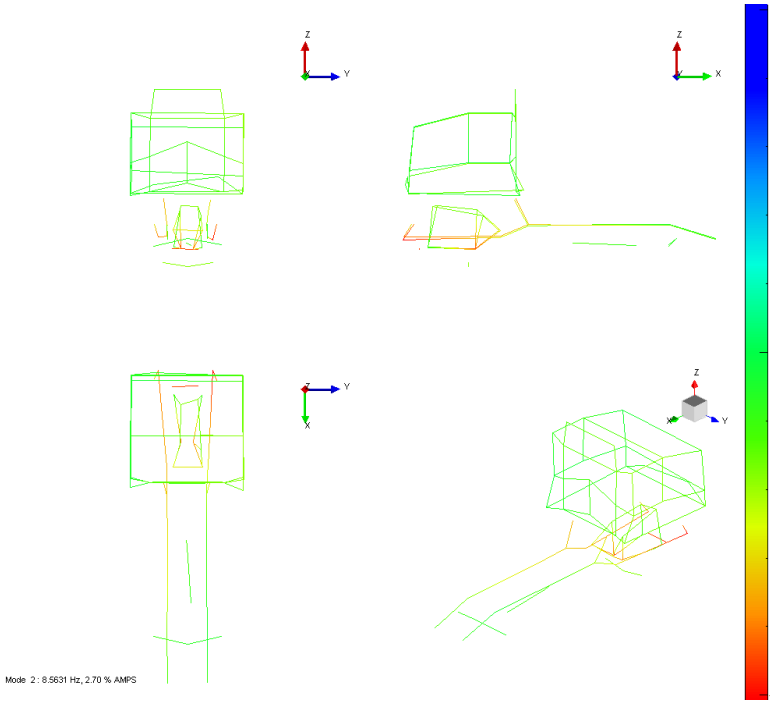


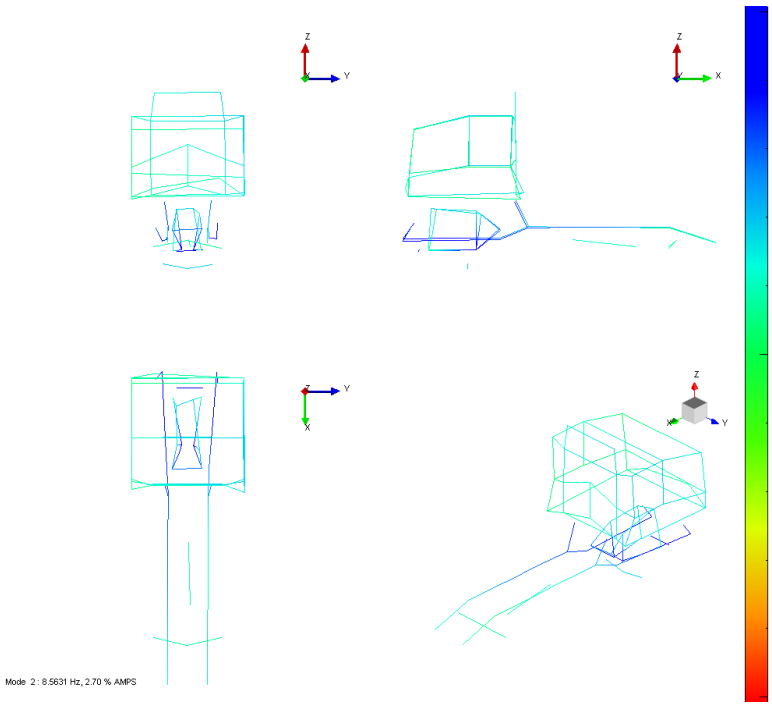
Figure A.4: *Measurement positions on wheel axles, cardan shaft and the cross beam under the front of the engine*

B Operational modal analysis modes

Here are all the modes that were found in the OMA but not displayed in the result section since they were not that vital for the beating phenomena.

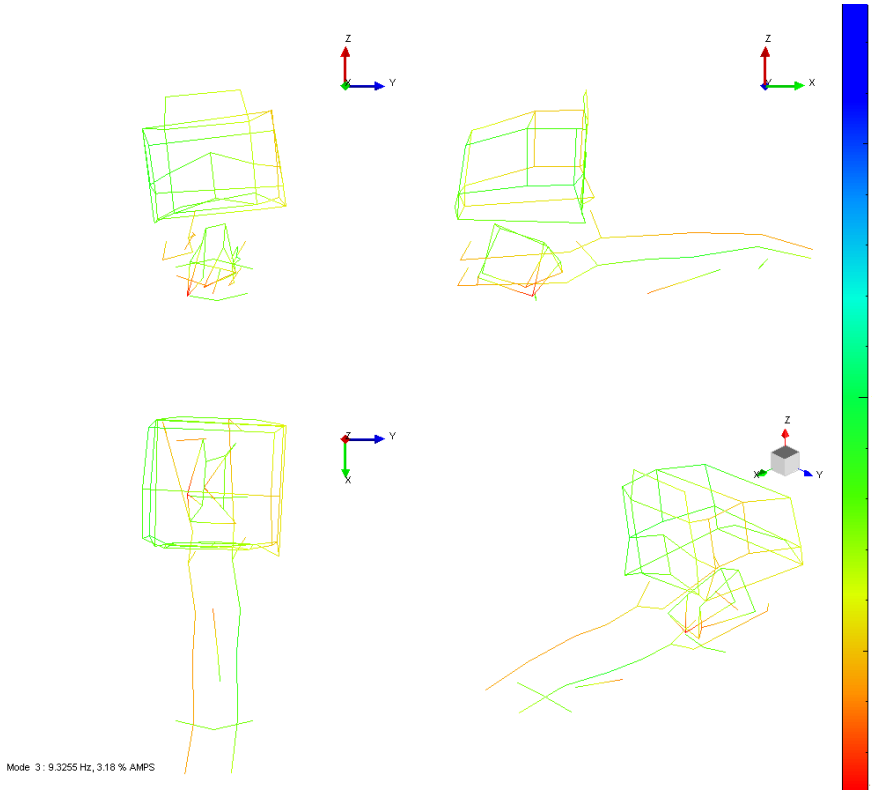


(a) Mode two, 8.6 Hz, phase 0 °

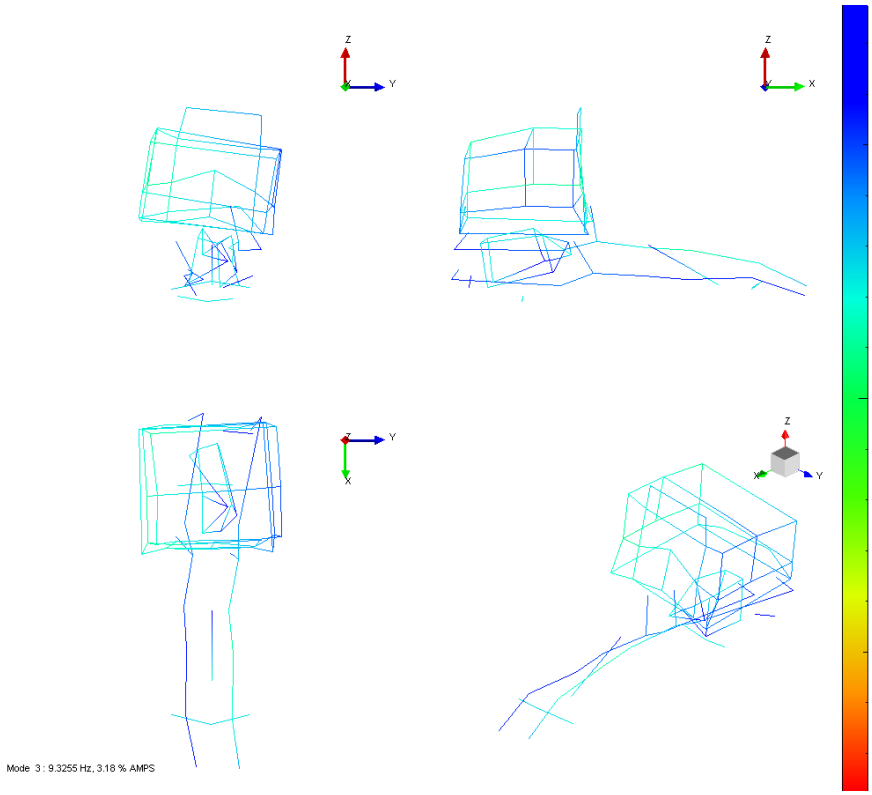


(b) Mode two, 8.6 Hz, phase 0 °

Figure B.1: *OMA mode two, 8.6 Hz*

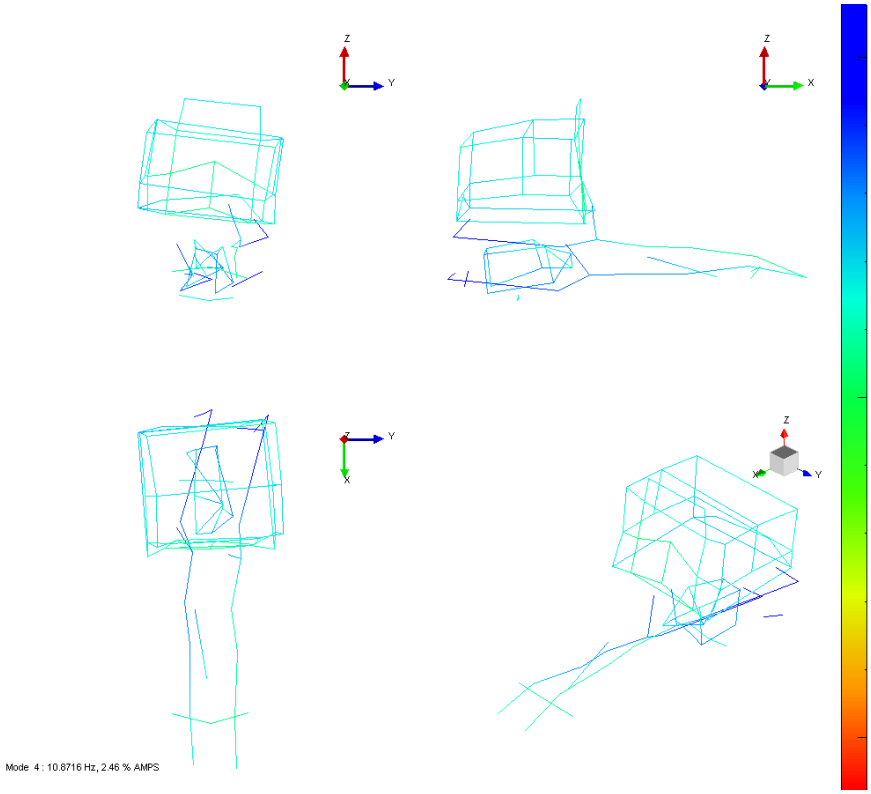


(a) Mode three, 9.3 Hz, phase 0 °

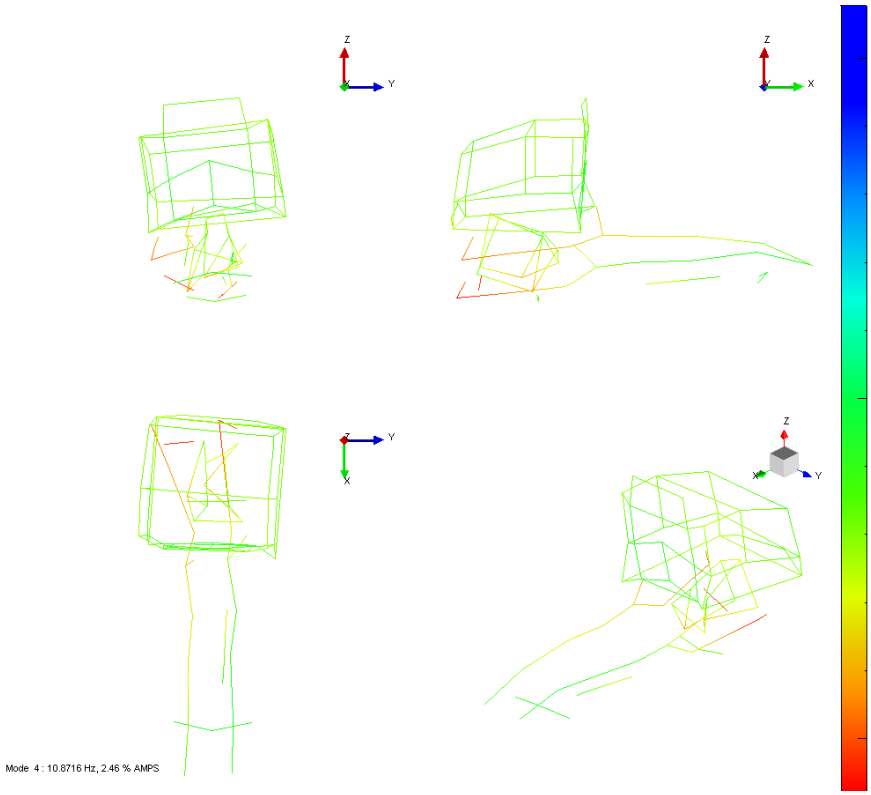


(b) Mode three, 9.3 Hz, phase 180 °

Figure B.2: *OMA mode three, 9.3 Hz*

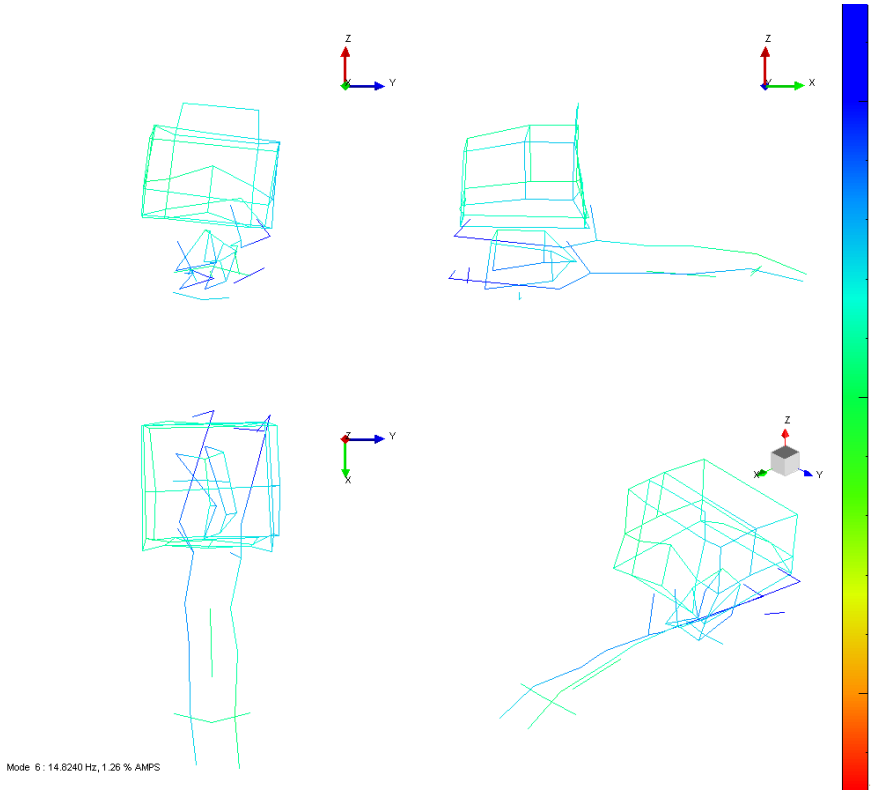


(a) Mode four, 10.9 Hz, phase 0 °

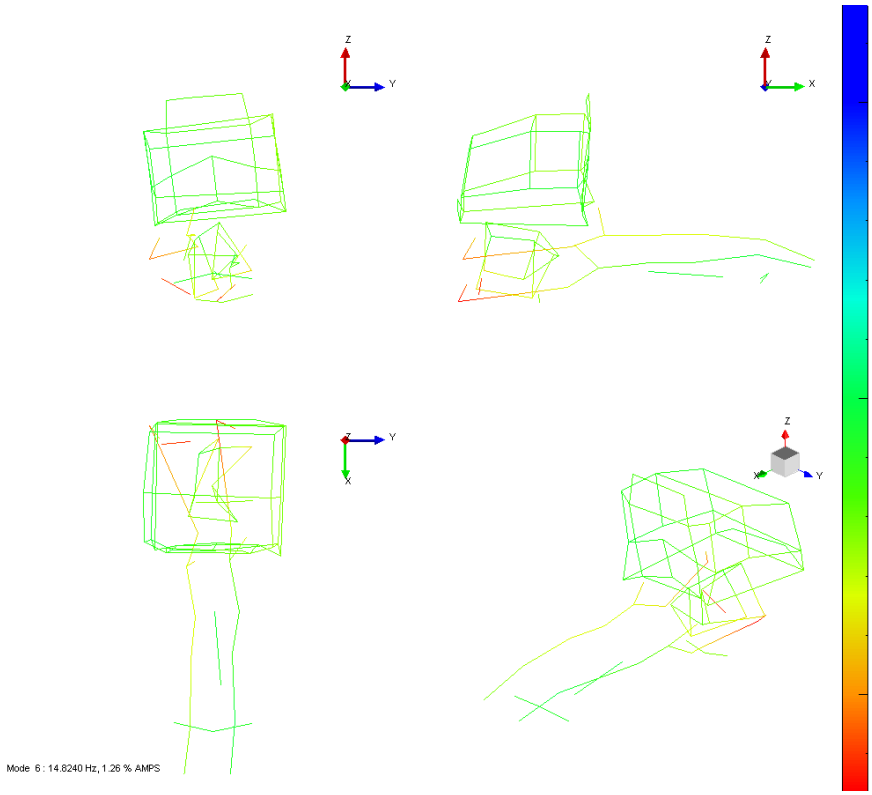


(b) Mode four, 10.9 Hz, phase 180 °

Figure B.3: *OMA mode four, 10.9 Hz*

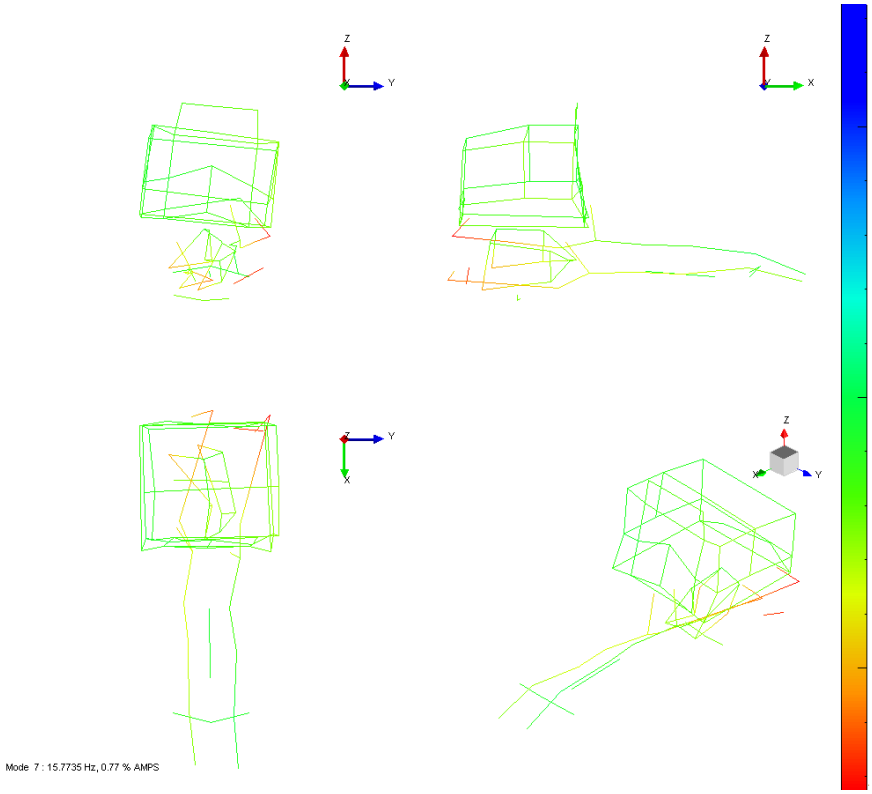


(a) Mode six, 14.8 Hz, phase 0 °

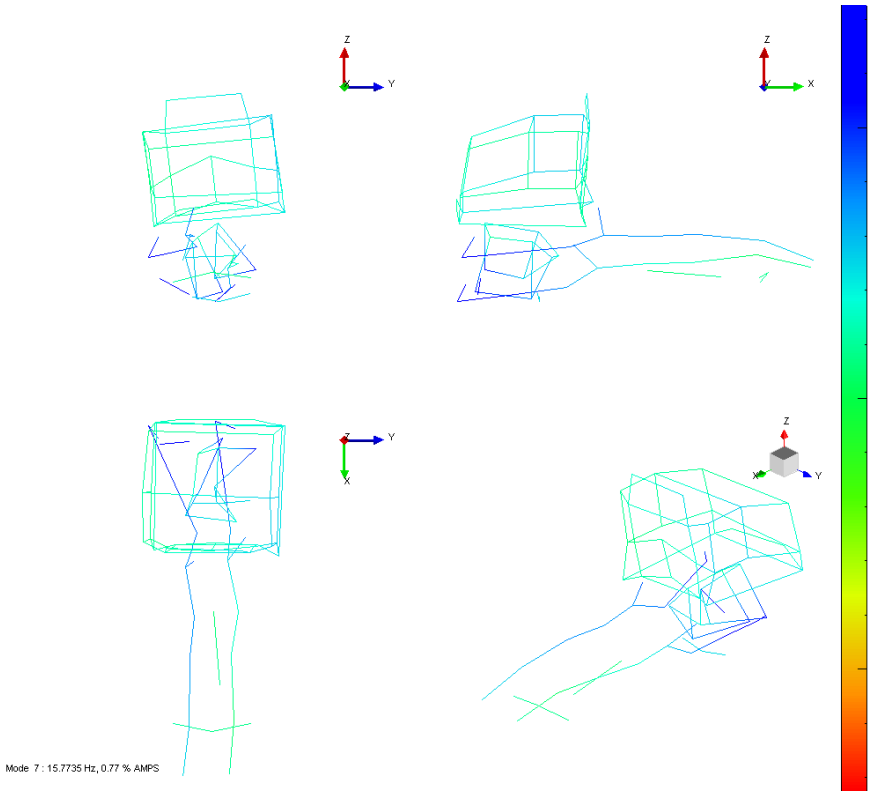


(b) Mode six, 14.8 Hz, phase 180 °

Figure B.4: *OMA mode six, 14.8 Hz*

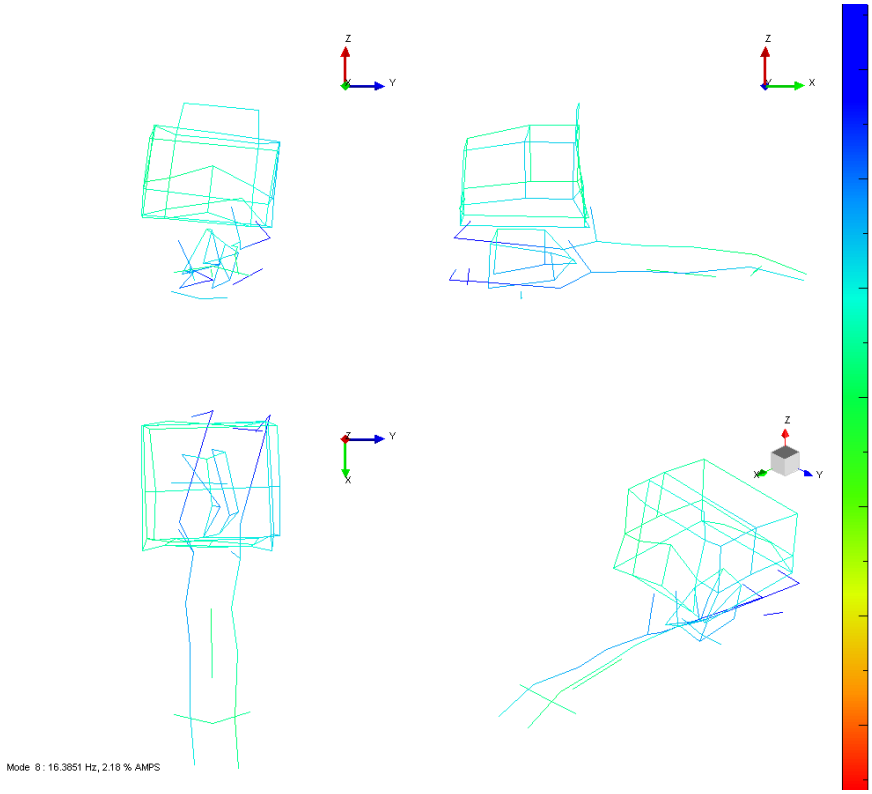


(a) Mode seven, 15.8 Hz, phase 0 °

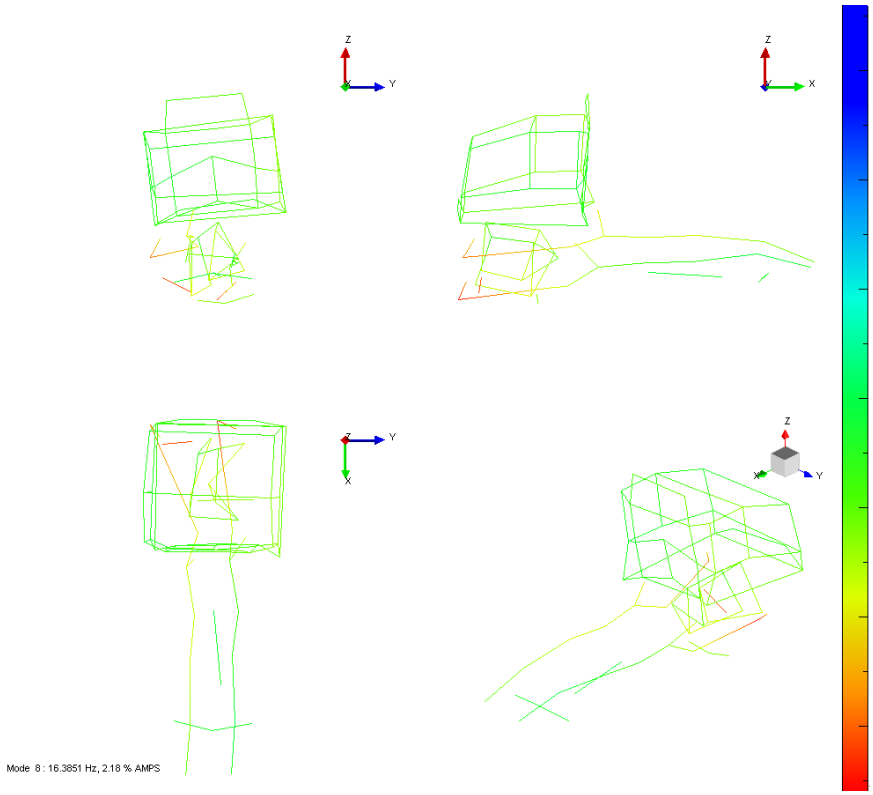


(b) Mode seven, 15.8 Hz, phase 180 °

Figure B.5: *OMA mode seven 15.8 Hz*

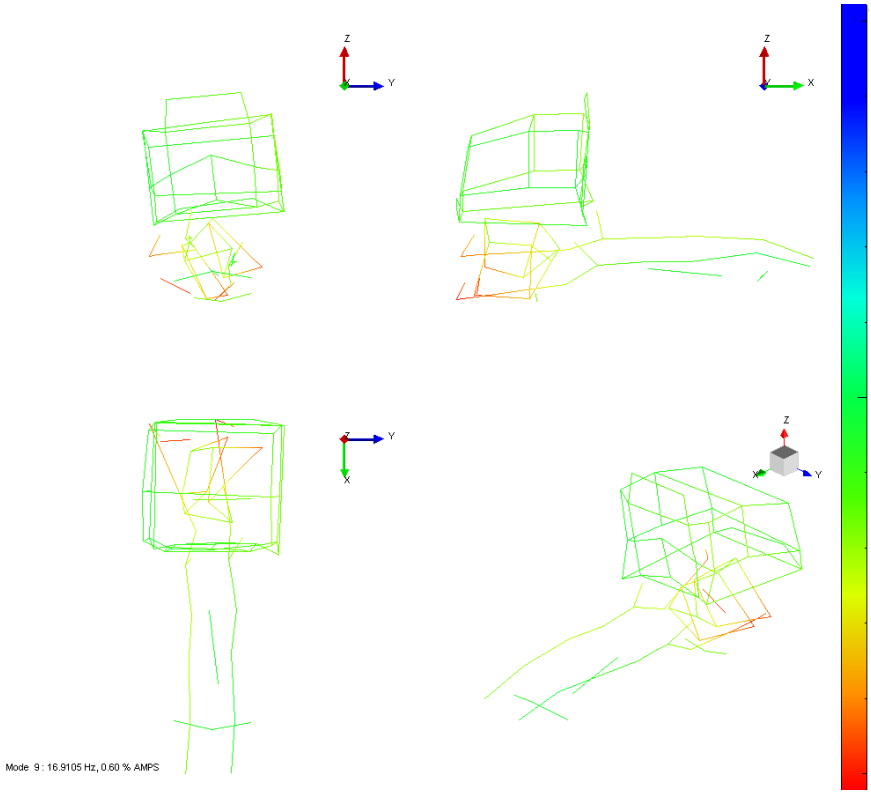


(a) Mode eight, 16.4 Hz, phase 0 °

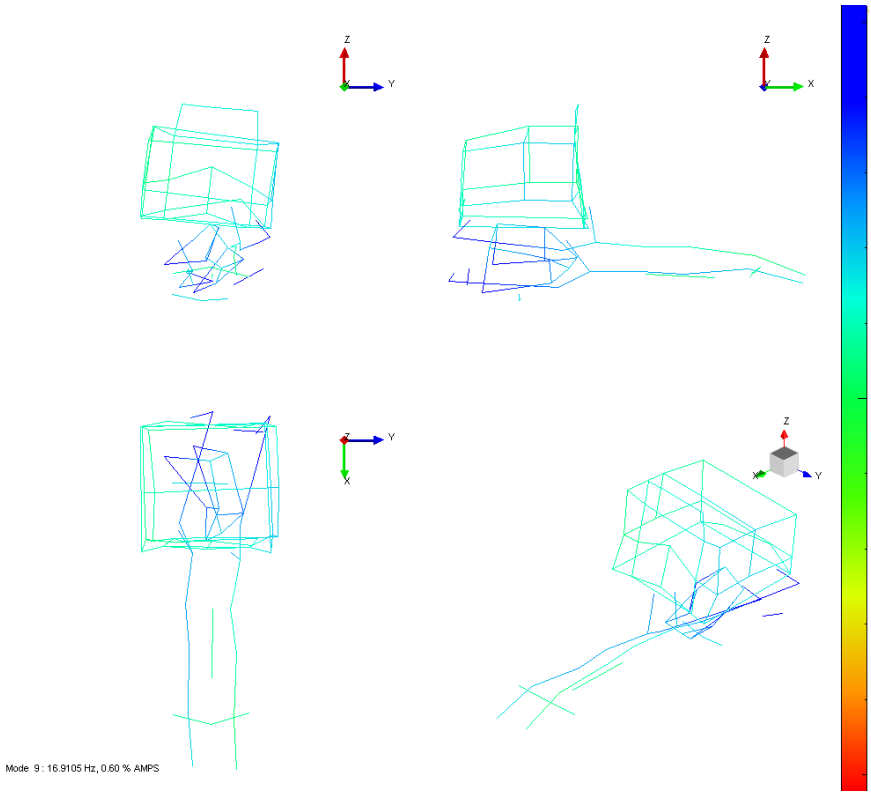


(b) Mode eight, 16.4 Hz, phase 0 °

Figure B.6: *OMA mode eight, 16.4 Hz*

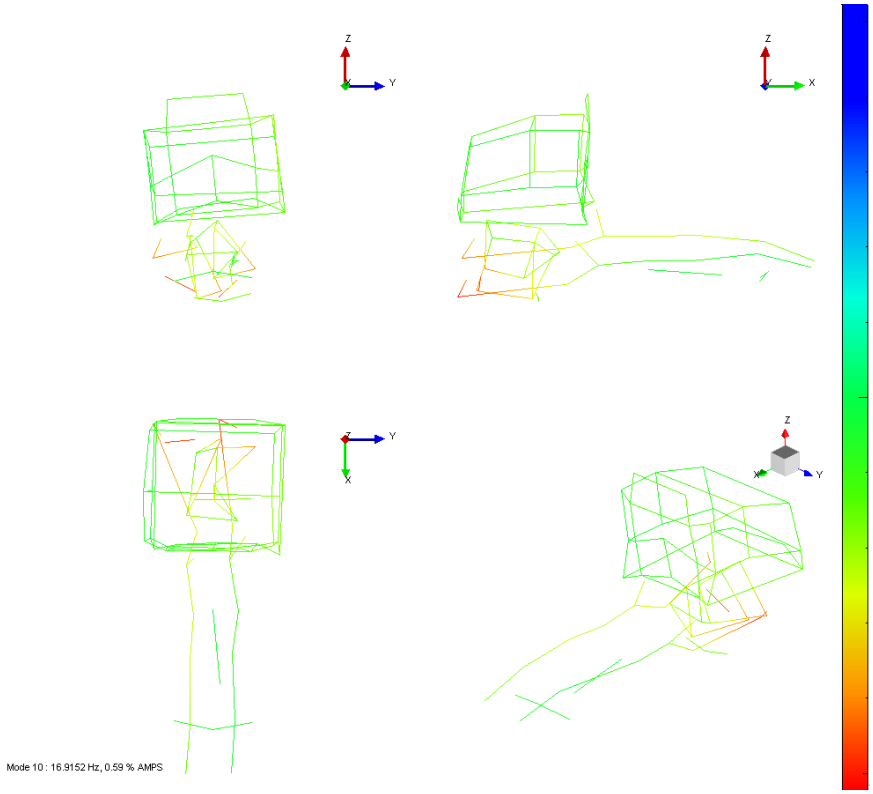


(a) Mode nine, 16.91 Hz, phase 0 °

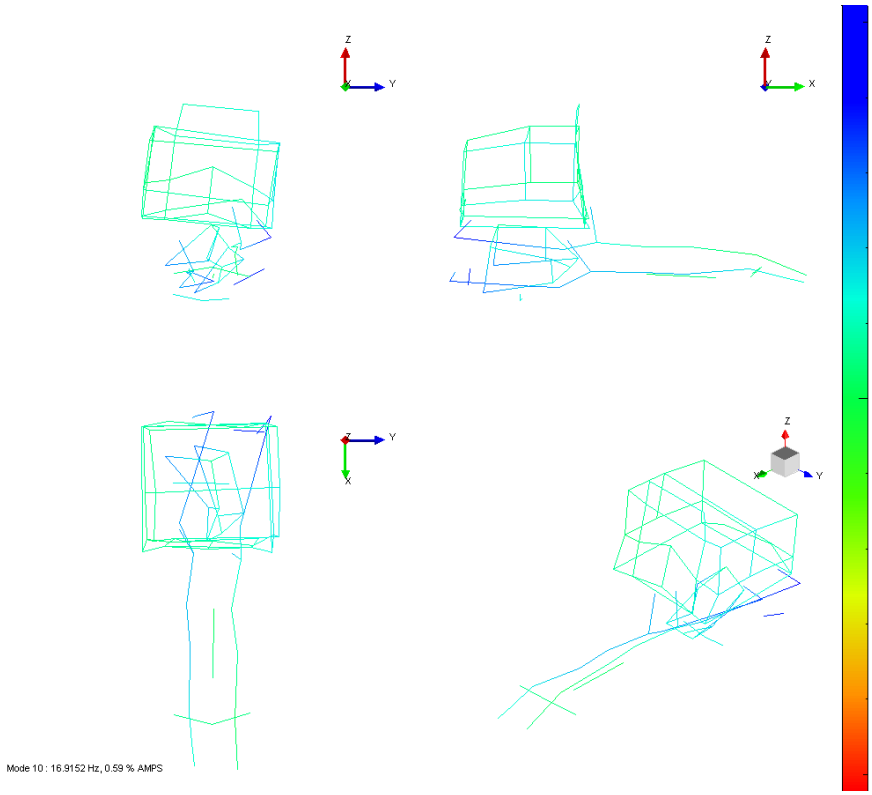


(b) Mode nine 16.91 Hz, phase 180 °

Figure B.7: *OMA mode nine, 16.91 Hz*



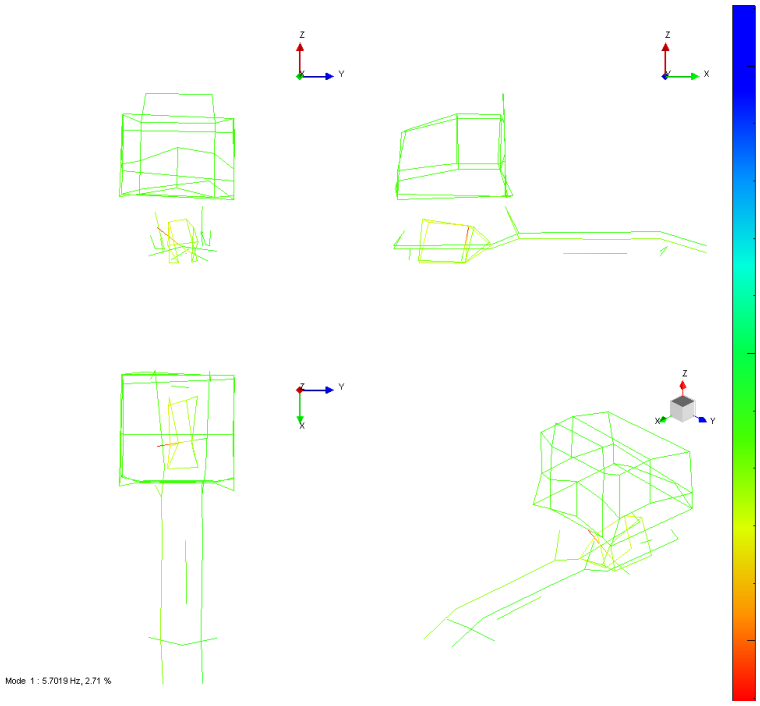
(a) Mode ten, 16.92 Hz, phase 0 °



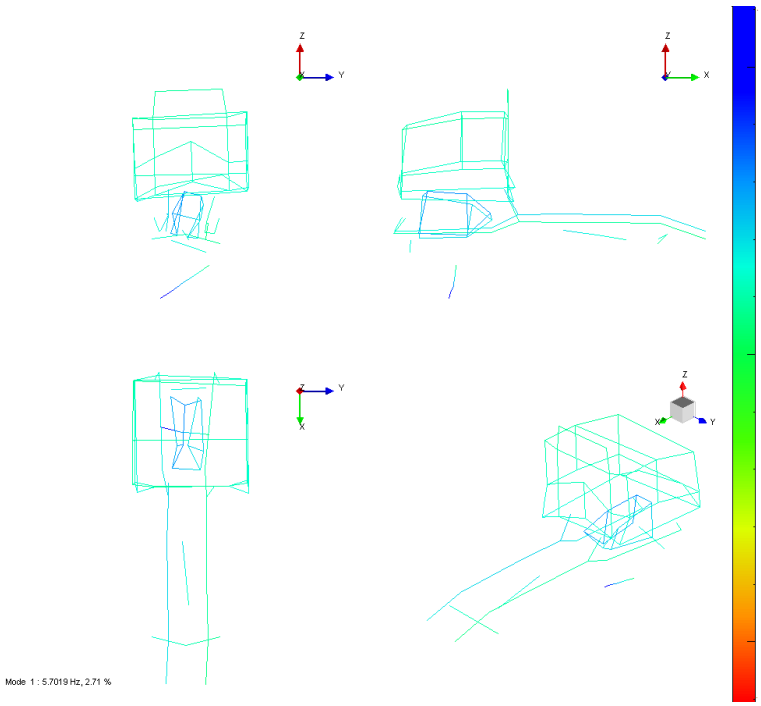
(b) Mode ten, 16.92 Hz, phase 180 °

Figure B.8: *OMA mode ten, 16.92 Hz*

C Experimental modal analysis modes

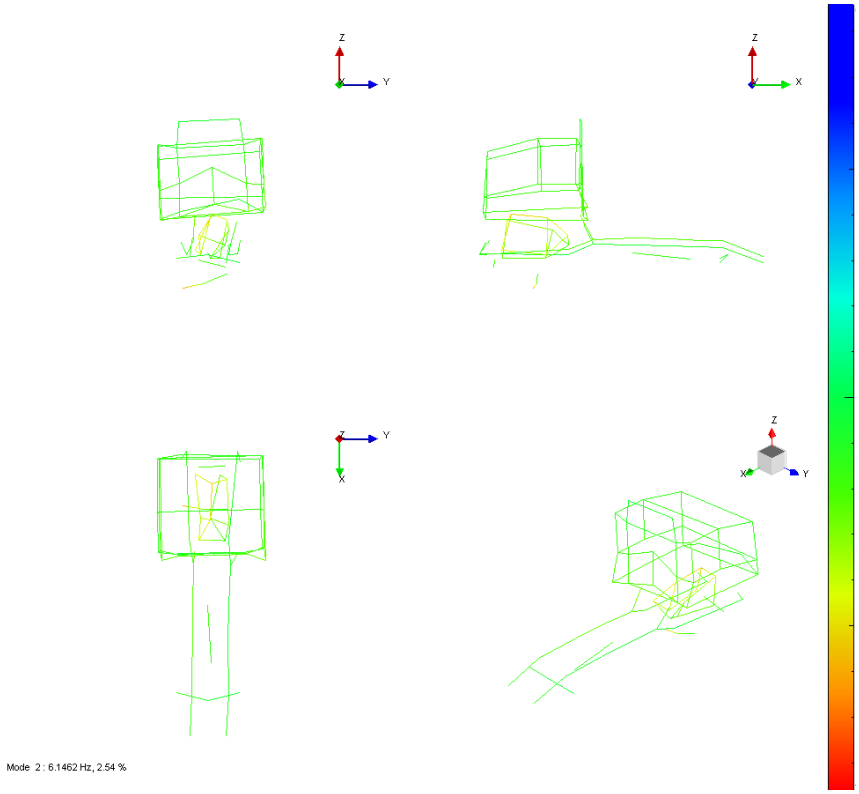


(a) Mode one, 5.7 Hz, phase 0 °

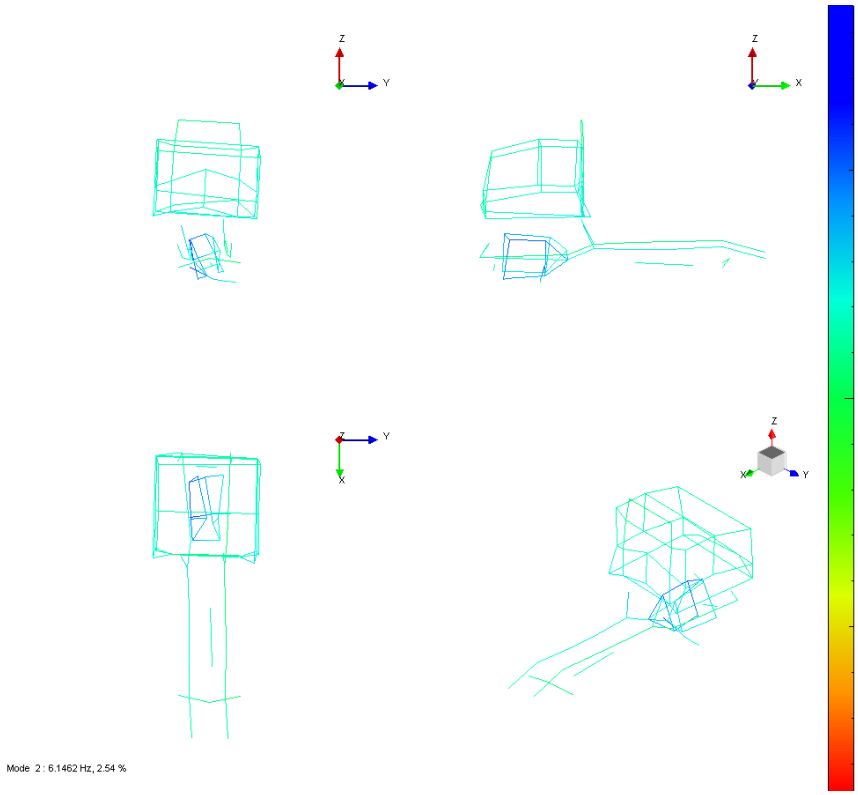


(b) Mode one, 5.7 Hz, phase 180 °

Figure C.1: *Modal mode one, 5.7 Hz*

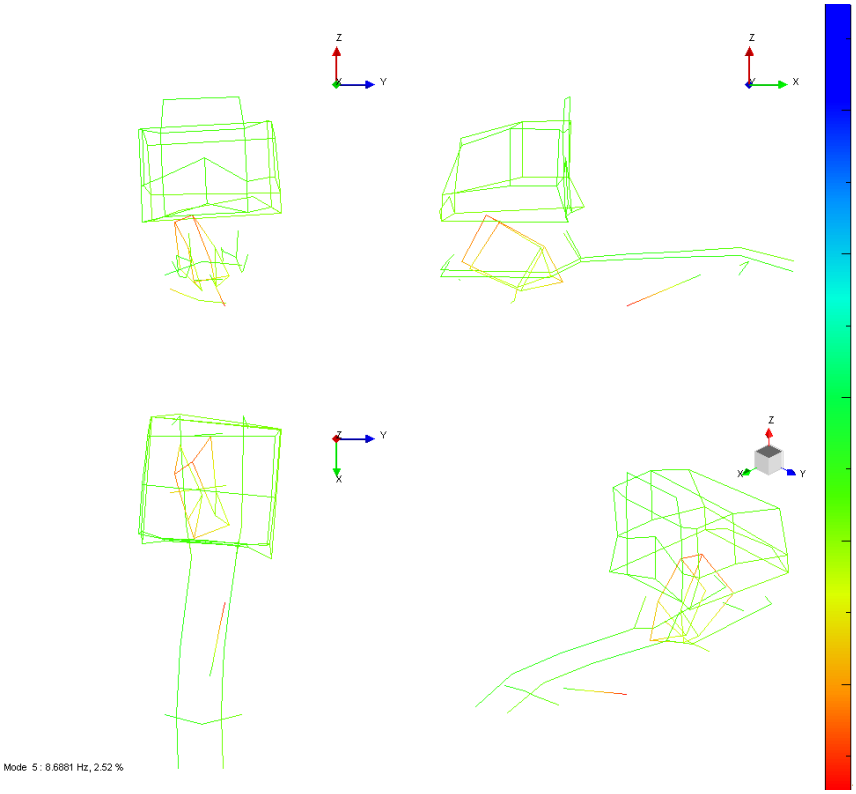


(a) Mode two, 6.1 Hz, phase 0 °

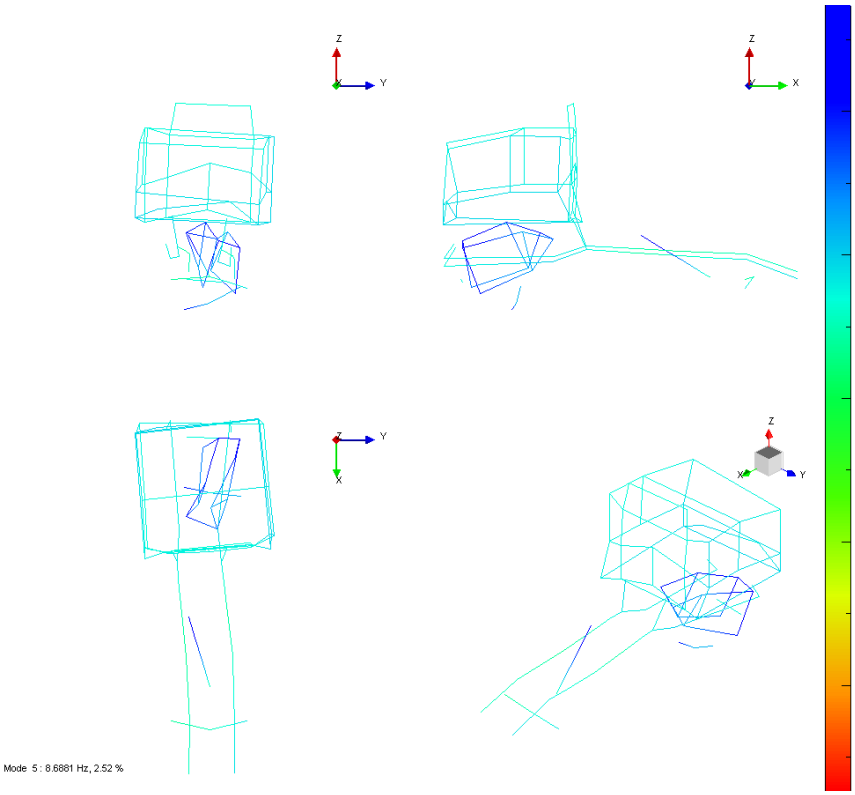


(b) Mode two, 6.1 Hz, phase 180 °

Figure C.2: *Modal mode two, 6.1 Hz*

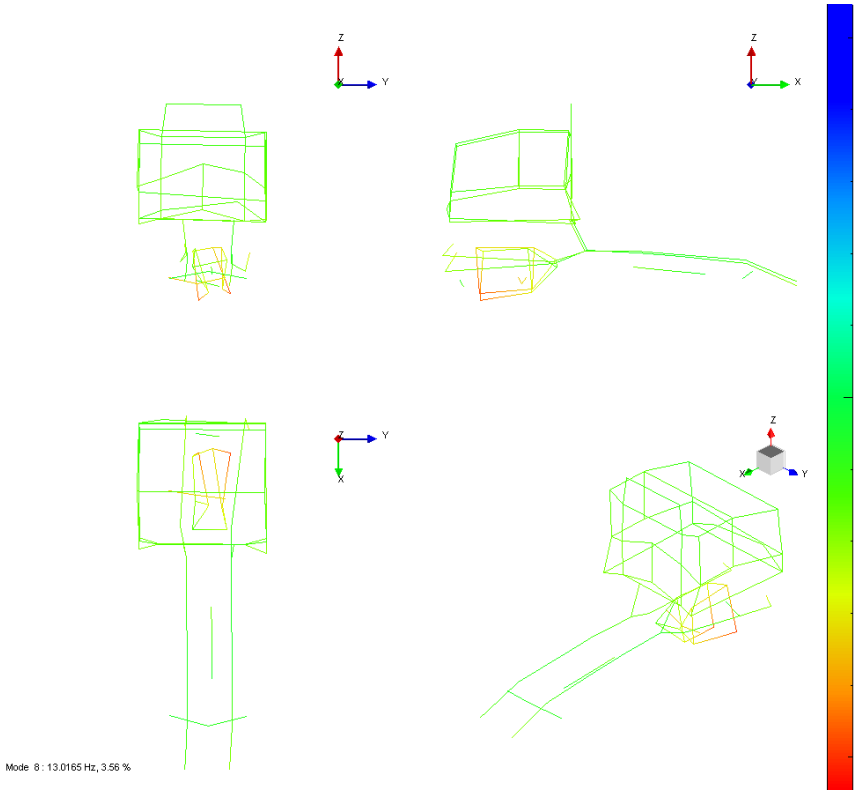


(a) Mode five, 8.5 Hz, phase 0 °

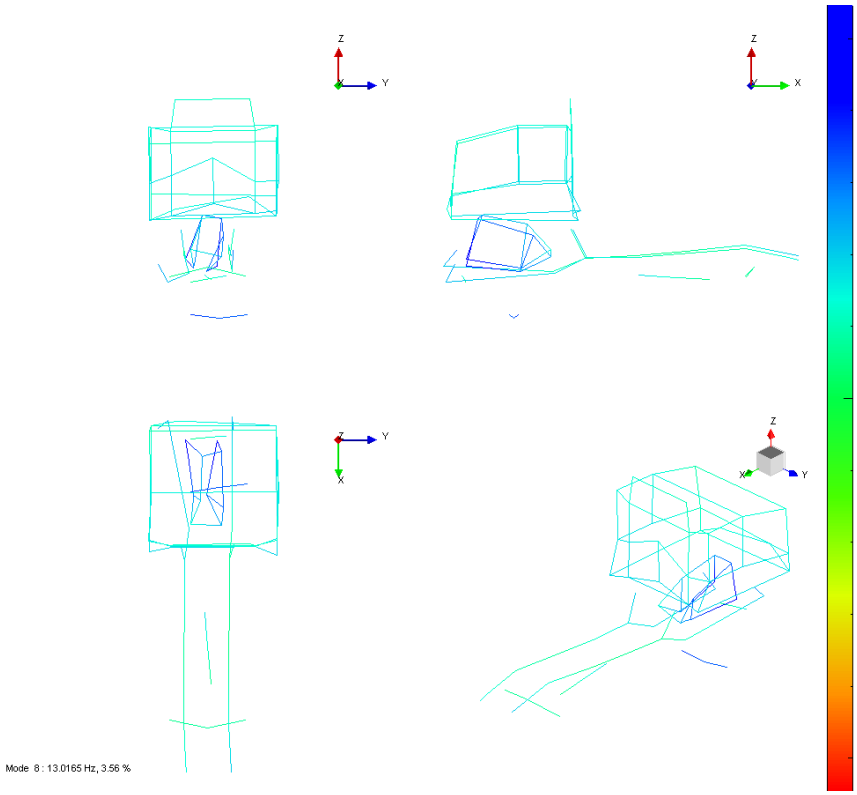


(b) Mode five, 8.5 Hz, phase 180 °

Figure C.3: *Modal mode five, 8.5 Hz*

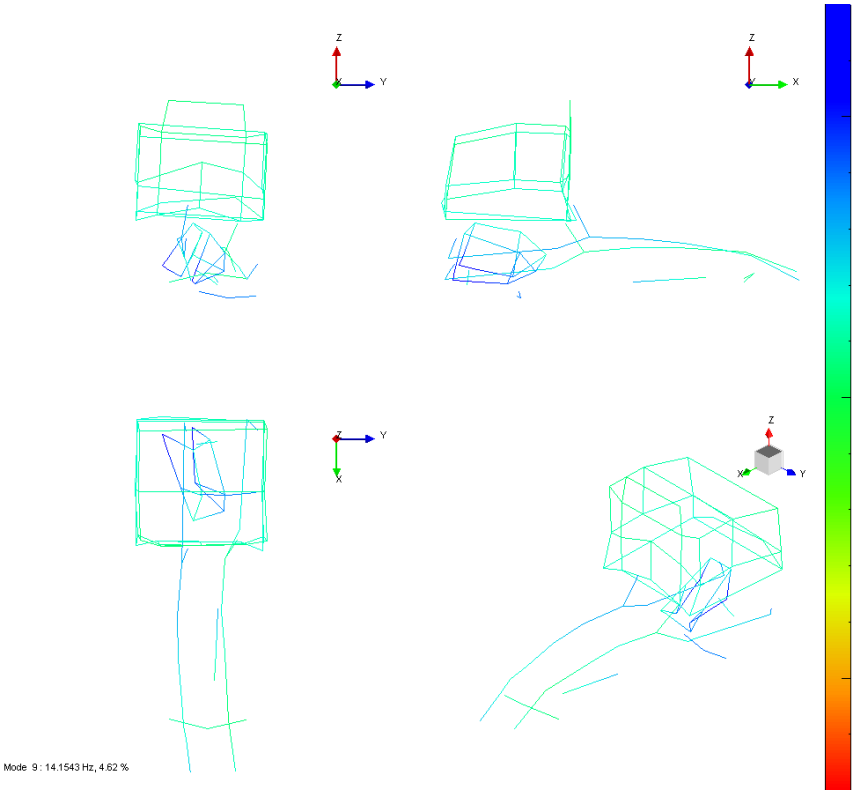


(a) Mode eight, 13.0 Hz, phase 0 °

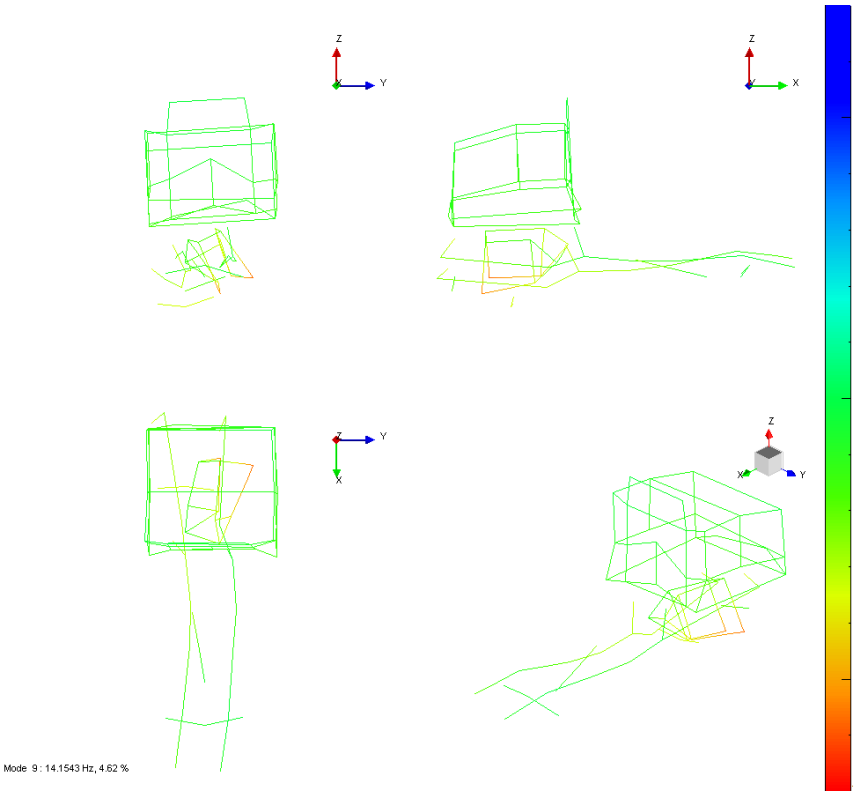


(b) Mode eight, 13.0 Hz, phase 180 °

Figure C.4: *Modal mode eight, 13.0 Hz*

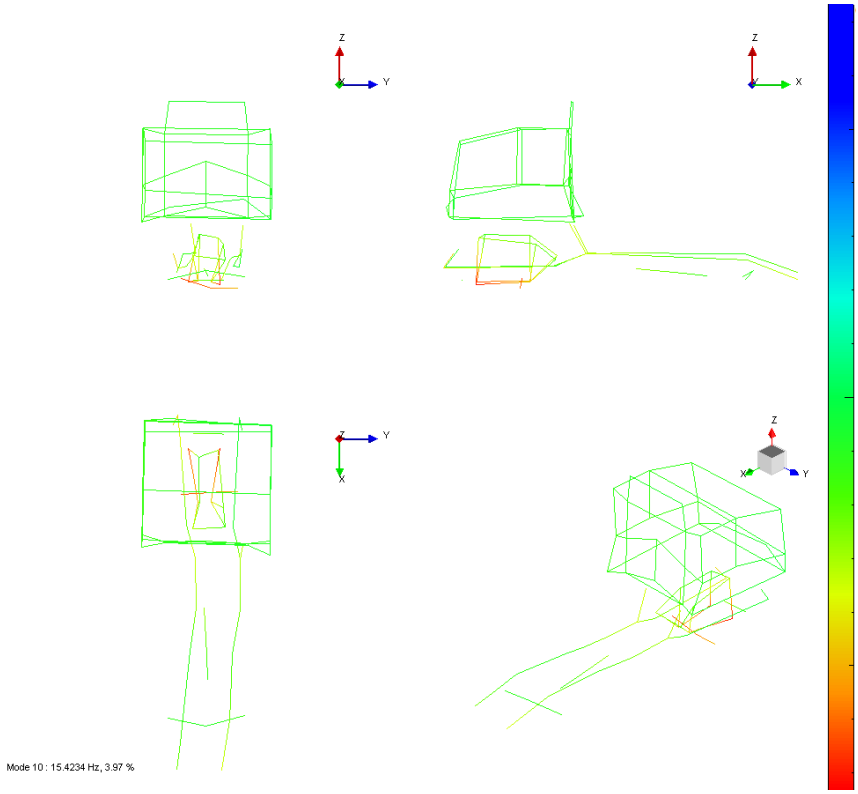


(a) Mode nine, 14.2 Hz, phase 0 °

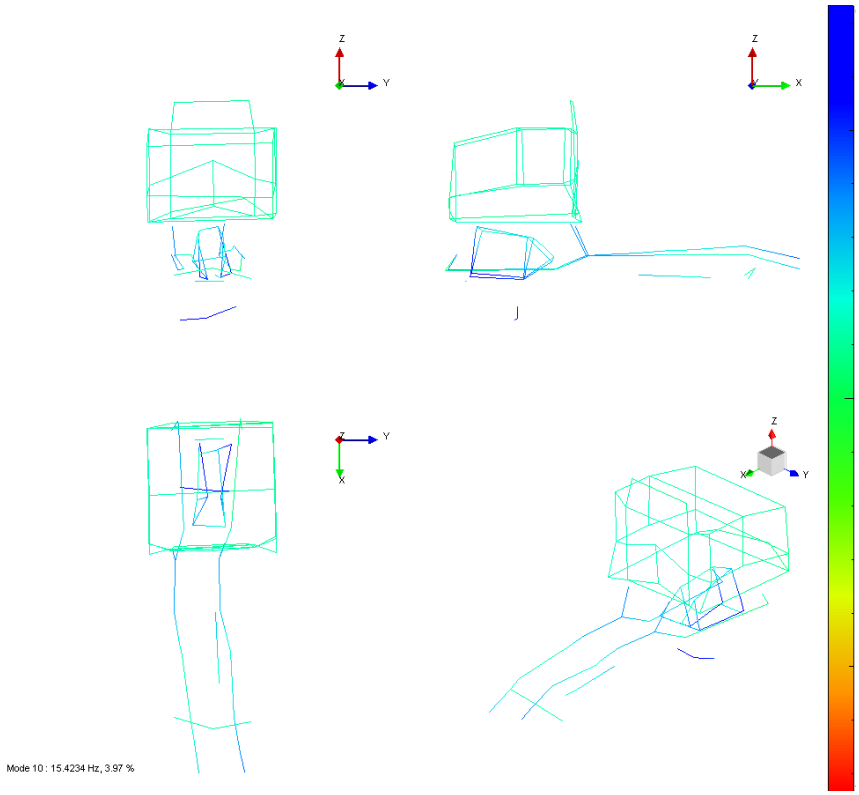


(b) Mode nine, 14.2 Hz, phase 0 °

Figure C.5: *Modal mode nine, 14.2 Hz*

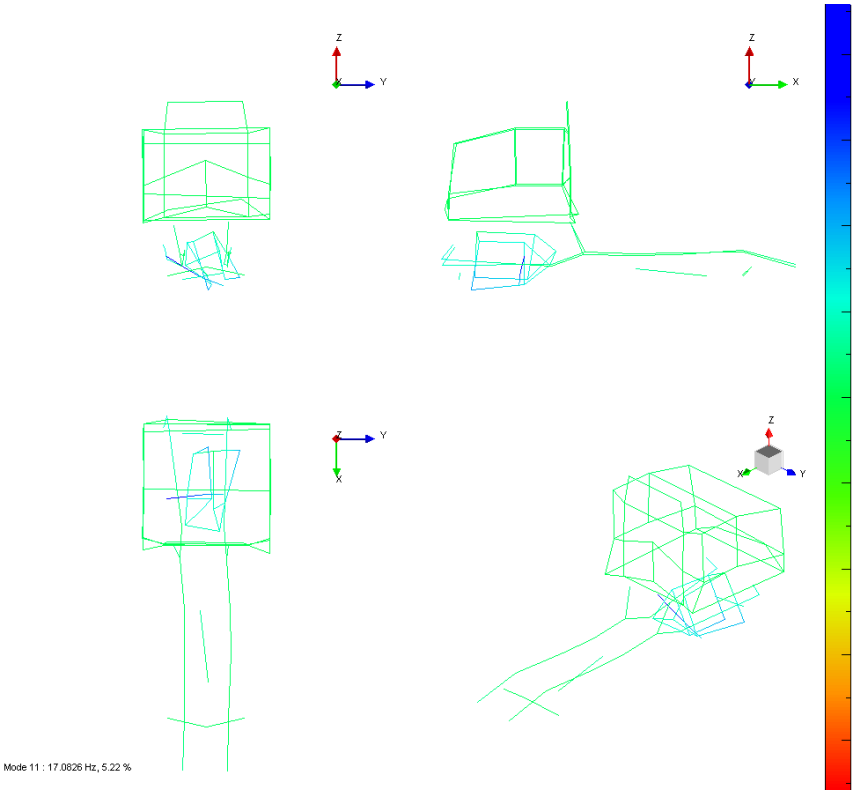


(a) Mode ten, 15.4 Hz, phase 0 °

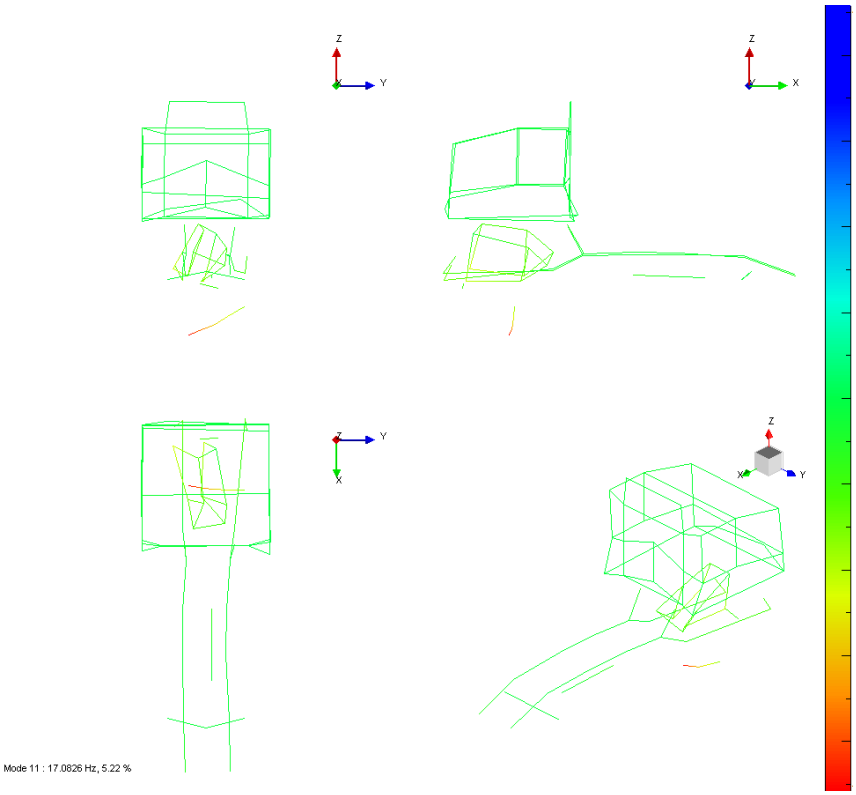


(b) Mode ten, 15.4 Hz, phase 180 °

Figure C.6: *Modal mode ten, 15.4 Hz*

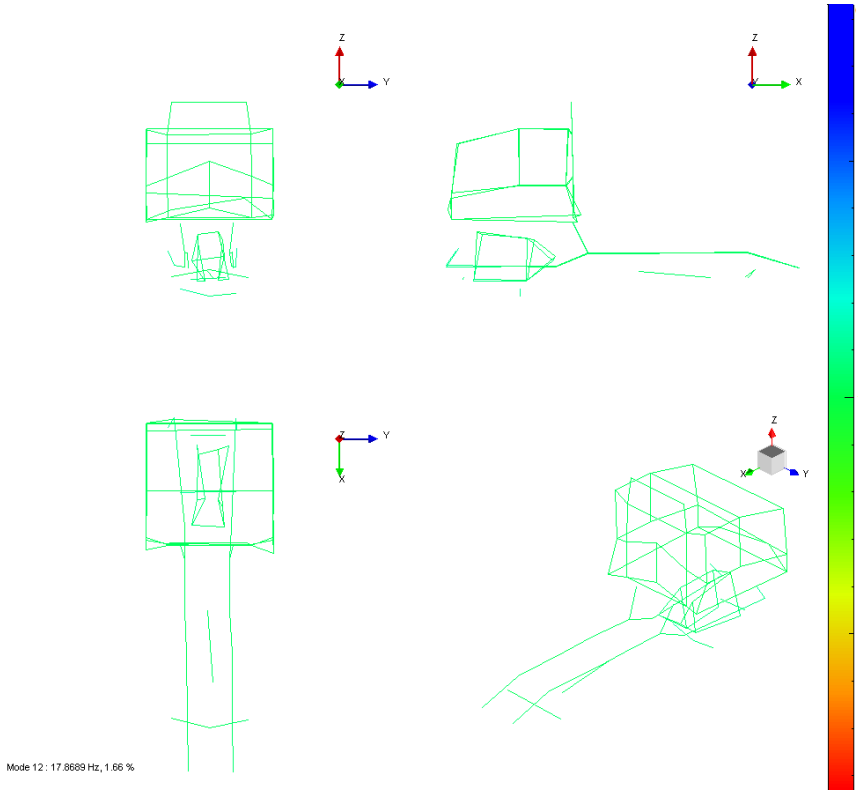


(a) Mode eleven, 17.1 Hz, phase 0 °

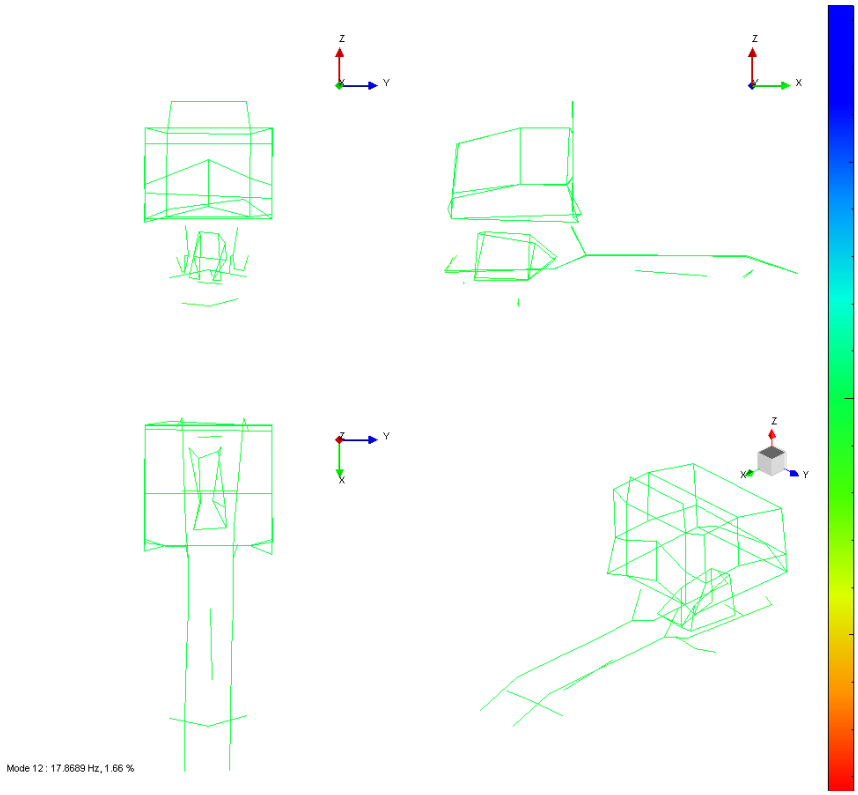


(b) Mode eleven, 17.1 Hz, phase 180 °

Figure C.7: *Modal mode eleven, 17.1 Hz*



(a) Mode twelve, 17.9 Hz, phase 0 °



(b) Mode twelve, 17.9 Hz, phase 180 °

Figure C.8: *Modal mode twelve, 17.9 Hz*