Correlation of Modal Shaker Test Simulations with respect to Physical Measurements

Degree project in Mechatronics Engineering

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BACHELOR THESIS IN AUTOMOTIVE ENGINEERING

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An investigation of modeling aspects needed to capture the frequency and amplitude behaviour observed in the Modal shaker measurement results.

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ABSTRACT

Understanding, evaluating and testing vehicle dynamics is an area that is becoming more and more important as the car industry develops more advanced and complex systems every year. When a car manufacturer establishes their presence in what is known as the premium brand segment the secondary ride experience is of great importance. A smooth and comfortable secondary ride experience during different road conditions plays a big role in how the customer perceives the quality of the ride, and whether the car manufacturer meets the criterias to be classified as a premium brand. In order to ensure a good secondary ride the composition of the suspension components has to be chosen carefully with good validation. To find the right components such as bushings, tires and dampers, several different testing techniques are used at the Volvo Vehicle Dynamics Department, one of them being the modal shaker analysis. This analysis is done with a modal shaker rig test and a real vehicle which makes it both expensive and time consuming to try different component models. Due to the shaker test being highly repeatable and covers a large range in frequency and amplitude, it makes it a good candidate for correlation. If instead this modal shaker test could be conducted with a simulating software and still produce results with sufficient fidelity levels it could be used as a complement to the real rig test. In this thesis work the aim is to find modeling aspects for the simulated modal shaker event that affect the correlation between the simulation and real test. This is done by testing different types of bushing, tire and damper models in the simulation software Adams Car. The results are then processed in MATLAB and a comparison is made to establish the degree of correlation to the real rig test. During this work it could be established that the MATLAB functions used to estimate the modal parameters natural frequency and damping ratio were sufficient to achieve sufficient fidelity levels in correlation. The bushing model MXmount, tire model Ftire and damper model Advanced Damper model are in need of further investigation regarding their parameterization to capture their true beneficial effects.

PREFACE

The bachelor thesis work “Correlation of Modal Shaker Test Simulations with respect to physical measurements” was conducted during the time interval 18th of January 2021 to Xth of June 2021 at Chalmers University of Technology on behalf of Volvo Car Corporation. During this time period the recommended restrictions of the COVID19 pandemic made it inconvenient to conduct the everyday work at the Volvo CAE Vehicle Dynamics Department in Torslanda, Gothenburg. Therefore, most of the everyday work was conducted in remote form. The physical rig test was performed at Volvos Proving Ground in Hällered supervised by Marcus Lindner (Volvo). Thesis examiner was Peter Bövik (Chalmers) Senior Lecturer at Mechanics and Maritime Sciences, Division of Dynamics.

For their valuable inputs, helping guidelines and availability on and off working hours, we would like to thank our Supervisors Albin Johnsson and Marcus Lindner. We would also like to thank Kristofer Weiner, Manager of the Vehicle dynamics Concept Team for providing us with the tools and information necessary to conduct this thesis work

NOMENCLATURE

Abbreviations

SPA	Volvo Scalable Product Platform
SM	System Mules
VP	Verification Prototype
TT	Tools tryout
PP	Production Prototype
STD	Standard
LCA	Lower Control Arm
UCA	Upper Control Arm
RSUSP	Rear Suspension
RSF	Rear Subframe
KNU	Knuckle
FRF	Frequency Response Function
Vdyn	Vehicle dynamics
VCC	Volvo Cars Corporation
Diff	Difference
LSCE	Least-Squares Complex Exponential Method.
MDoF	Multi Degree of Freedom
Fv	Force-velocity
LNL	Linear/Non-Linear
F-tire	Flexible Structure Tire model
LSCF	Least-Square Complex Frequency
LSFD	Least-Square Frequency-Domain

Equations

x	Current position
x_s	Reversal point position
F_{last}	Last integration step force
F_{max}	Maximum friction force
$RDL\rho$	Logarithmic friction force internal parameters
S	Stress-strain in spring
D	Stress-strain in damper
E	Modulus of elasticity.
η	Viscosity
p	Number of frequency bins
m	Number of responses
n	Number of excitation signals
f_s	Sample rat

f_i	Sinusoid frequency
b_i	Damping coefficient
A_i	Amplitude
$\boldsymbol{\Phi}_\mathrm{i}$	Phase of sinusoid
a_i	Amplitudes
x_i	Poles

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1. Introduction

1.1 Background

Vehicles are exposed to different types of road conditions which creates exciting forces at different frequencies and amplitudes. These forces or vibrations need to be subdued to obtain a comfortable driving experience. Bushings are commonly used to reduce vibration and to minimize the forces that transfer between parts. To test different bushings on an actual vehicle is both time consuming and expensive, while a digital simulation offers a wider spectrum of options and parameters which can be adjusted. The car manufacturer Volvo evaluates their cars by driving on a test track with modified road conditions, through different laboratory tests and by digital simulation. In order for the digital simulation to produce accurate results that correspond to the laboratory test called modal shaker, it is essential to tune the virtual assemblie's models of components .The assignment to be carried out on behalf of Volvos request consists of laboratory testing with a modal shaker at different amplitudes and frequencies on a specific car model. The laboratory results will be compared to the simulated results produced by the Adams Car modal shaker event and deviations will be evaluated.

1.2 Purpose

The purpose of this thesis is to correlate the vehicle model to the actual vehicle in a modal shaker test. If the model accurately predicts the shaker test it will improve confidence in all ride simulations done within the model and with models modeled in the same way. The purpose of studying the component model fidelities is to understand the influence of the model fidelity within the modal shaker test, and to some extent understand the influence tuning of the various components. The purpose of adding a compliment to the rig test is to make the testing more efficient instead of building a big series of test vehicles (SM, VP, TT, PP) and so forth. Achieving this will result in a lesser cost in both time and funding.

1.3 Research Questions

The research question of this bachelor thesis is to investigate how well the vehicle model used for simulation is correlated with the real testing subject. But also external factors such as accelerometer placement, force placement and characteristics of input force signal will be considered. Investigations will be done to identify which components in the model that contribute to worse or better correlation. The components considered are tires, bushings, and dampers.

1.4 Delimitations

- All examined parts will be analyzed as rigid bodies.
- The modal shaker tests analysis will not be conducted below the frequency 6Hz nor over 60Hz.
- The modal shaker amplitude will not be above 800N nor lower than 100N.
- The excitation signal is a swept sine with constant amplitude force.
- No other results than those provided by the rear knuckle accelerometers will be examined.
- The true longitudinal mode (mode 1) and the true vertical mode (mode 2) will be primarily evaluated.
- Only the rear left wheel has been studied.

1.5 Issues under investigation

Table 1. 1: Issues under investigation

Number	Statements	Verified		Rejected		Date
		Status	Comment	Status	Comment	
Nr 1.5.2	The bushing models STD and MXmount with standard parametrization are sufficient to achieve approvable fidelity levels.			x	STD model not sufficient. MXmount needs further investiga-tion.	2021-04-15
Nr 1.5.3	The LCA bushing model will be an important modeling aspect for good correlation on longitudinal modes.	x	(See 4.3 Comparis-on 3).			2021-04-19
Nr 1.5.4	The current modelling fidelity of Adams/car is sufficient to capture the main frequency and amplitude behaviours observed in the Modal shaker test			x	Partially true to some extent. (See 4.1 Comparison 1).	2021-02-16
Nr 1.5.5	The components chosen for further investigation accounts for most of the amplitude and frequency behaviour in the modal shaker test.	x				2021-05-03
Nr 1.5.6	Simulated modal event setup parameters such as placement of accelerometers and force have a big impact on mode correlation.			x	(See 4.2 Comparison 2)	2021-04-22

2. Theory

The chapter “Theory” contains information and explanations of relevant theory regarding the subjects modal analysis, implemented models, MATLAB functions and physical measurement.

2.1 Modal Analysis

The Modal Analysis section contains information and theory regarding modal analysis and its areas application.

2.1.1 Modal Analysis Theory

Modal analysis is a method used to determine and describe a system's dynamic characteristics such as mode shapes, damping and natural frequencies. These dynamic characteristics can then be translated to mathematical models used to describe the system’s behavior. Physical properties of the system such as damping, mass and stiffness defines the natural modes of vibration that the system possesses [1]. The mathematical model is often called the modal model of a system. In other words, the modes can be viewed as a property of a system or an object and are often identified through modal testing.

2.1.2 Modal Testing

Modal testing does not have a consistent regulatory framework but is more of an experimental technique where a system is exposed to an exciting force at different frequencies. The subject or system is often provided with different types of sensors such as accelerometers to collect the response data and force transducers to measure the exciting force signal. The collected response and excitation signals can then be used to calculate the frequency response signal.

2.1.3 Application of modal analysis

Modal analysis can be used to measure a variety of components for a car such as chassi, suspension systems and driver’s seat. The applications that modal analysis brings forth is a new way to study vibration as a secondary ride experience. With this application the behaviour of a car's chassi can be analyzed and the frequency and amplitude which creates the most disturbance to the secondary ride [1].

2.1.4 Mathematics for modal analysis

As mentioned earlier modal analysis is all about representing a system's dynamic characteristics with mathematical models. Due to modal analysis being done in both the time domain and frequency domain the involved mathematics are wide ranging. One of the more important sections in mathematics is matrix theory, thus modal analysis is largely based on multi-degree-of-freedom (MDoF) dynamic system analysis [1].

2.2 Modeling of suspension components

Modeling of suspension components contains modeling information regarding the suspension components starting with the two damper models used followed by the bushing models and tire models used during the work.

2.2.1 Damper models

During the thesis work two different damper models were implemented, the advanced damper model and the viscous damper model. The two different models offer some individual characteristics with the advanced damper model being a bit more complex whilst the viscous damper model is based on the force-velocity curve model. The advanced damper model is shown in Figure 2.2.

The viscous damper model uses one of most common damper models which is called the force-velocity curve. The force-velocity or FV curve lacks the ability to capture some dynamic phenomena such as dependency of damper displacement and hysteresis [2] and is shown in Figure 2.1.

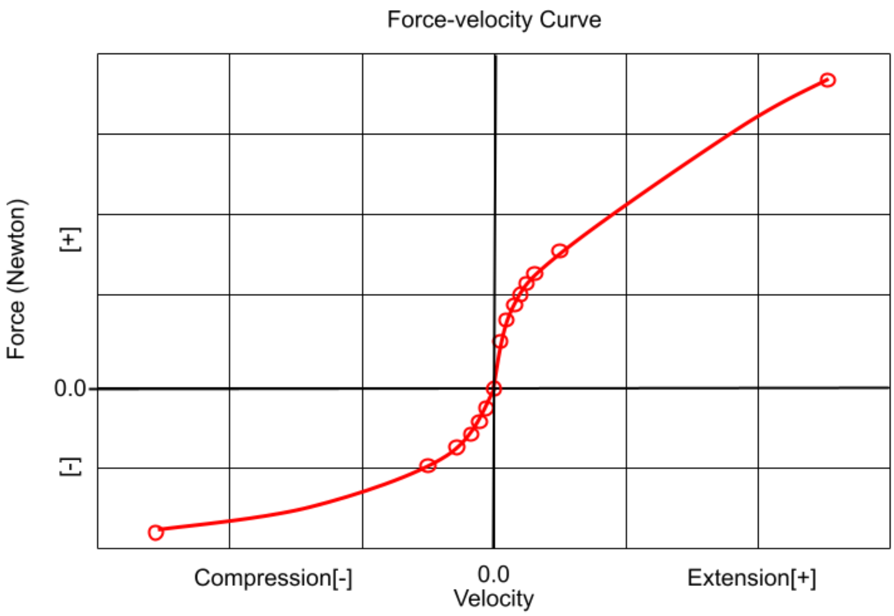

Figure 2. 1: Force-velocity curve

Figure 2.1 represents the non-linear force velocity curve where the negative force represents the rebound stroke, and the positive force represents the jounce. This model has two vectors for describing the force-velocity relationship and together they create the curve. The circles in the figure represent the vector pairs [3]. As seen in the figure there is no hysteresis in this ideal model which an actual damper would produce, this shows its inability to fully reflect the behaviour of a real damper.

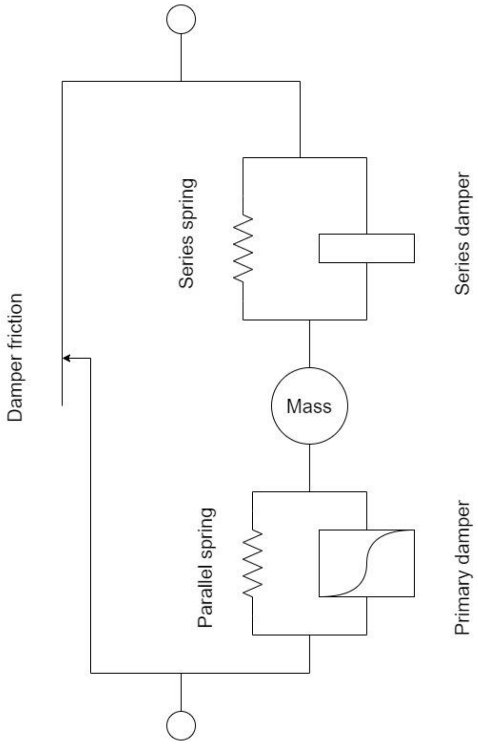

Figure 2. 2: Advanced damper model

As shown in Figure 2.2 the advanced damper model consists of several components, one of them being “Series spring” which represents the compressibility of damper gas and is referred to as “Series spring” because it is in series with the primary damper. In parallel with the series spring there is a “Series damper” whose main purpose is to dampen oscillations produced by the series spring. This component is also in series with the primary damper. The “Mass” is a representation of the mass of moving parts inside the damper, these parts are oil, air and piston rod. To represent the oil compressibility the “Parallel spring” was added, and it is parallel connected with the primary damper. The “Primary damper” is based on the force-velocity curve. The last component of this model is the “Friction model” where the entire damper friction is modelled [2].

2.2.2 Bushings

The STD bushing is modeled according to the Kelvin-Voigt model which is represented by a viscous damper parallel with an elastic spring [5]. Due to its parallel configuration the strain in each component will be the same and can be written as:

$$\varepsilon_{Total} = \varepsilon_S = \varepsilon_{D.} \qquad (2.1)$$

Where:
S - Stress-strain in spring
D - Stress-strain in damper.

Therefore the total stress is the sum of damper and spring stress:

$$\sigma_{Total} = \sigma_S + \sigma_D. \tag{2.2}$$

With respect to the time *t* the function for stress $\sigma(t)$ can be described as:

$$\sigma(t) = E\varepsilon(t) + \eta \frac{d\varepsilon(t)}{dt}. \tag{2.3}$$

Where:

E - Modulus of elasticity.

η - Viscosity.

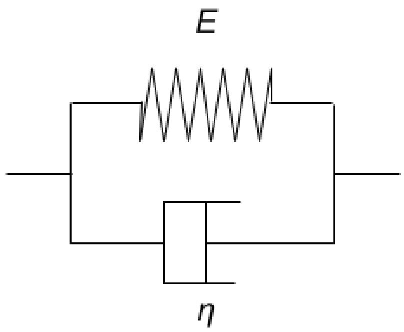

Figure 2. 3: Kelvin Voigt model

For a lot of applications the Kelvin Voigt model shown in Figure 2.3 is a sufficient model of a bushing by representing it as a spring and damper connected in a parallel setup. But for evaluating a systems amplitude and frequency dependent behaviour of elastomers it has some limitations making the MXmount model a more suitable model.

The MXmount bushing model has the ability to model the amplitude and frequency behaviour of elastomers due to its LNL (Linear/Non-Linear) elastomer model. This complete model is a composition of both the nonlinear and linear model in a parallel setup. As shown in Figure 2.4 the upper setup represents the linear module with a dual Kelvin-Voigt model representation. The mathematical representation for the model is [4]:

$$F = F_1 + F_2 \tag{2.4}$$

$$F_1 = k_1 u \tag{2.5}$$

$$F_2 = k_2 z + c_2 \dot{z} = c_1(\dot{u} - \dot{z}) \tag{2.6}$$

Where: $$\alpha = \frac{k_2}{k_1} \beta = \frac{c_2}{c_1} \gamma = \frac{c_1}{k_1} \tag{2.7}$$

Ending up with: $$F = k_1\left(u + \frac{c_1}{k_1}(\dot{u} - \dot{z})\right) = k_1(u + \gamma(\dot{u} - \dot{z})) \tag{2.8}$$

And: $$\dot{z} = \frac{1}{1+\beta}\left(\dot{u} - \left(\frac{\alpha}{\gamma}\right)z\right) \tag{2.9}$$

The k_1 represents the static stiffness of the model though it might not exactly be the same as the static stiffness of the elastomer. This linear model is used to describe the frequency dependency.

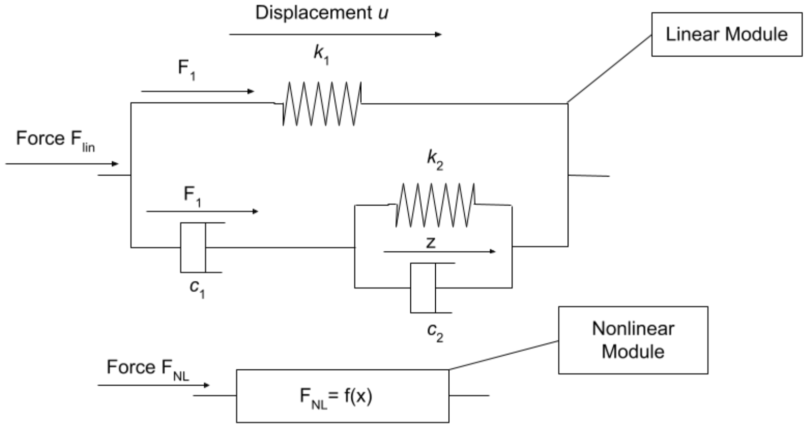

Figure 2. 4: MXmount combined model

The lower setup in Figure 2.4 represents the nonlinear module and is used to model the amplitude dependency. At time *t* the nonlinear force can be described as:

$$F_{NL} = \int_{t(x_s)}^{t} \left[\left(|x - x_s|\frac{RDL\rho}{\rho+1}\right)\left(\frac{F_{max} \pm F_{last}}{F_{max}}\right)\dot{x}\right]dt \tag{2.10}$$

Where:

x - Current position

x_s - Reversal point position

F_{last} - Last integration step force

F_{max} - Maximum friction force

$RDL\rho$ - Logarithmic friction force internal parameters

2.2.3 Tires

During the thesis work two different tire models were used, there are many different tire models but the ones implemented and used in the Adams Car modal shaker event were the Ftire or Flexible Structure Tire model and PAC2002 models. These two models offer some different characteristics and are based on different formulas. The PAC2002 is based on the Pacejka Magic Tire Formula. The mathematical Magic Formula that PAC2002 uses is developed to describe some basic tire characteristics such as pure cornering slip conditions, pure longitudinal slip conditions and combined slip conditions [6].

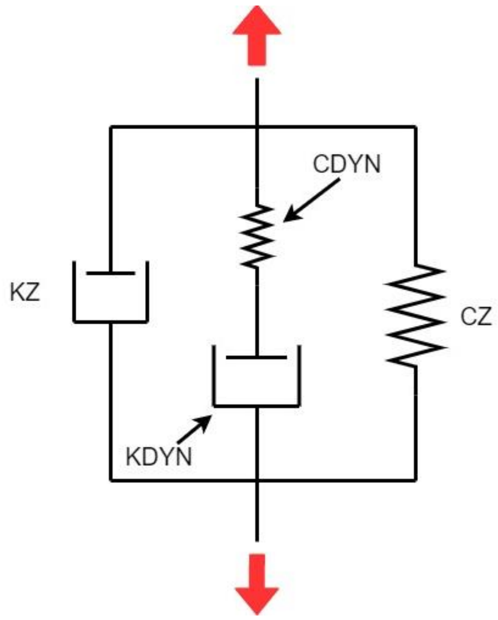

Figure 2. 5: Maxwell element

For this work the focus was mainly on the wheels vertical and non-rolling dynamics and therefore the PAC2002 model might not be as relevant as the Ftire model. Even though it offers a configuration where a Maxwell element is implemented to improve the models non-rolling tire properties [6]. The Maxwell element was not used during this correlation work, however the model is shown in figure 2.5. The main tire model used during the modal shaker event simulations was the Ftire model due to the Pacejka models inability to capture amplitude dependencies.

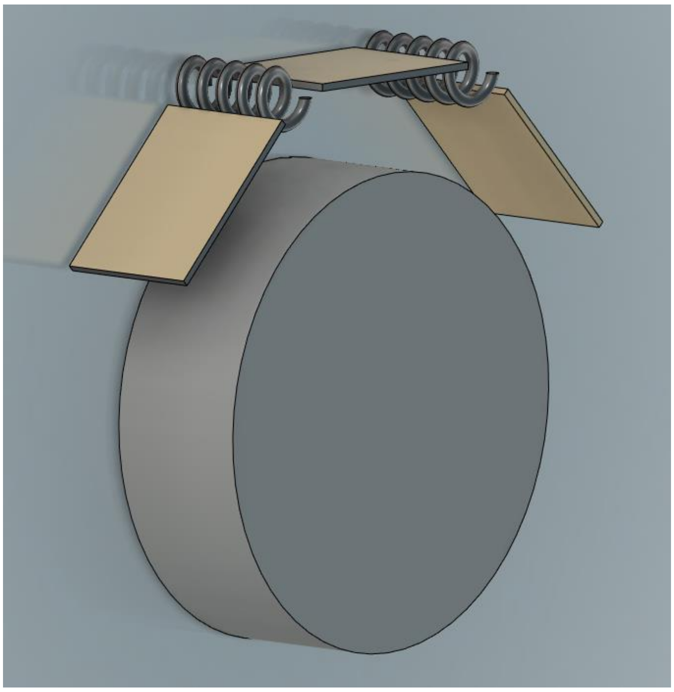
Figure 2. 6: Representation of Ftire belt elements

The other model used in Adams car was the Ftire, this model is able to capture the effects that can only partly be captured with the Maxwell element when using PAC2002. The reason that the non-rolling Ftire can pick up these effects is that the F-tire consists of belts and belt elements connected with springs as represented in Figure 2.6. With the structure that the F-tire is set up with it will be possible to determine the stiffness of the tire in a variety of directions. When using the F-tire a variety of different parameters can be set, such as, stiffness, damping, friction and wear. To make the model even more realistic it offers several features to set and customize. These are, to choose tread pattern, air pressure, mass etc [7].

2.3 MATLAB functions

This section explains the MATLAB functions and some theory behind them that is fundamental for understanding the functions and why they were used during the thesis work.

2.3.1 Modal FRF

The modal FRF function takes the input and output time signals and estimates a matrix of frequency response functions with the specified sample rate set by the user. The estimation is done using Welch's method. Welch's method, also known as the periodogram method for estimating power spectra, divides the time signal to consecutive blocks [8]. Each block with a periodogram.

Following modalfrf configuration was used during this work:
[frf, f] = modalfrf (x, y, fs, hann(L), noverlap, 'sensor, 'acc')

Output Arguments

frf - 3D array, matrix or vector containing the frequency response function of size $p \times m \times n$.

$p\ -\ number\ of\ frequency\ bins.$

$m\ - number\ of\ frequency\ bins.$

$n\ -\ number\ of\ excitation\ signals.$

f - Vector containing the frequencies.

Input Arguments

x - Matrix or vector with the excitation signals.

y - Matrix or vector with response signals.

fs - Sample rate in Hz.

hann(L) - function that returns a symmetric L-point window where L is a positive integer representing the window length.

noverlap - The number of overlapping samples as a positive integer.

Name-Value Arguments

sensor - Specify the sensor type.

acc - Voltage of response signal is proportional to acceleration.

Welch's Method

Denote the m th windowed, zero-padded frame from the signal x by:

$$x_m(n)\ \triangleq\ \omega(n)x(n+mR), n\ = 0,1.\dots. M-1, m = 0,1.\dots\dots K-1 \tag{2.11}$$

Let R represent the hop size and K the number of frames available. The *m* th block periodogram. *N* is the number of element in the complex vector x_m for the fast fourier transform *FFT* :

$$P_{xm,M}(\omega_k)\ =\ \frac{1}{M}\left|FFT_{N,k}(x_m)\right|^2 \triangleq \frac{1}{M}\left|\sum_{n=0}^{N-1}\ x_m(n)e^{-j2\pi nk/N}\right|^2 \tag{2.12}$$

Estimation of the power spectral density is calculated by the formula:

$$\hat{S}_x^W(\omega_k)\ \triangleq \frac{1}{K}\sum_{m=0}^{K-1}\ P_{xm,M}(\omega_k) \tag{2.13}$$

This is just the average of periodograms over time. Periodograms are created from non-overlapping blocks of data when ω*(n)* is the rectangular window [8].

2.3.2 Modal FIT

The modalfit function estimates the modal parameters natural frequency and damping based on the calculated frequency response functions.

Following modalfit configuration was used in this work.
[fn, dr] = modalfit(frf, f, fs, mnum, 'FitMethod', 'lsce', 'FreqRange', [6 60])

Output Arguments
fn - Normally a 3D array or matrix of natural frequencies but with 'lsce' specified 'fn' returns as a vector of 'mnum' elements independent of the size of 'frf'.
dr - Normally a 3D array or matrix of damping ratios but with 'lsce' specified 'dr' return as a vector of 'mnum' elements.

Input Arguments
frf - The frequency response functions as a 3D array, matrix or vector.
f - The frequency vector where the number of elements must match the number of rows of frf.
fs - The sample rate of measured data in hertz.
mnum - An integer of the number of modes searched for.

Name-Value Arguments
FitMethod - A fitting algorithm using the 'lsce' as default setting.
lsce - Least-Squares Complex Exponential Method.

Least-Squares Complex Exponential Method
This method calculates the impulse response for each FRF and then fits complex damped sinusoids with Prony's method.

The damped sampled sinusoid can be created with the form,

$$\begin{aligned} s_i(n) &= A_i e^{-b_i n/f_s} cos(2\pi f_i n/f_s + \phi_i) \\ &= \frac{1}{2} A_i e^{\ j\phi_i} exp(-(b_i/f_s \ - \ j2\pi f_i/f_s)n) + \frac{1}{2} A_i e^{-\ j\phi_i} exp(-(b_i/f_s \ - \ j2\pi f_i/f_s)n) \\ &\equiv a_{i+}\, x_{i-}^n + a_{i-}\, x_{i-}^n \end{aligned} \tag{2.14}$$

$f_s - Sample\ rate$
$f_i - Sinusoid\ frequency$
$b_i - Damping\ coefficient$
$A_i - Amplitude$
$\boldsymbol{\phi}_i - Phase\ of\ sinusoid$
$a_i - amplitudes$
$x_i\ \ - Poles$

Prony's Method

The sample function $h(n)$is a superposition of $N/2$modes according to Prony's method where $N$is the number of amplitudes and poles [9].

$$h(0) = a_1\, x_1^0 + a_2\, x_2^0 + \ldots + a_N\, x_N^0$$
$$h(1) = a_1\, x_1^1 + a_2\, x_2^1 + \ldots + a_N\, x_N^1$$
$$\vdots$$
$$h(N-1) = a_1\, x_1^{N-1} + a_2\, x_2^{N-1} + \ldots + a_N\, x_N^{N-1} \qquad (2.15)$$

The poles x_i are roots of the polynomial $c_0, c_1, \ldots c_{N-1}$.

$$x_i^N + c_{N-1}\, x_i^{N-1} + \ldots + c_1\, x_i^1 + c_0\, x_i^0 = 0. \qquad (2.16)$$

An autoregressive model $L = 2N$ samples of $h$is then used to find the coefficients.

$$\begin{bmatrix} h(0) & h(1) & \cdots & h(N-1) \\ h(1) & h(2) & \cdots & h(N) \\ \vdots & \vdots & \ddots & \vdots \\ h(L-N-1) & h(L-N) & \cdots & h(L-2) \end{bmatrix} \begin{bmatrix} c_0 \\ c_1 \\ \vdots \\ c_{N-1} \end{bmatrix} = - \begin{bmatrix} h(N) \\ h(N+1) \\ \vdots \\ h(L-1) \end{bmatrix}$$

(2.17)

The poles are then found using the MATLAB 'roots' function. The poles are used to calculate the damping factors and frequency through identification of the real and imaginary parts of the pole algorithms. Following equation is used to construct the impulse response and solve for amplitudes.

$$\begin{bmatrix} h(0) \\ \vdots \\ h(N-1) \end{bmatrix} = \begin{bmatrix} x_1^0 & \cdots & x_N^0 \\ \vdots & \ddots & \vdots \\ x_1^{N-1} & \cdots & x_N^{N-1} \end{bmatrix} \begin{bmatrix} a_1 \\ \vdots \\ a_N \end{bmatrix}$$

(2.18)

2.4 Simcenter Testlab model analysis

When receiving the FRFs a least-square complex frequency-domain is estimated to be able to retrieve the poles and participation factors. After that is completed a second estimation is conducted but this time a least-square frequency-domain is done to estimate the residue and can only be done after the frequency and damping values have already been estimated one time. When the frequencies are estimated two times, the mode shapes consider the parameters and then determine if the modes are "real" or "complex" [10].

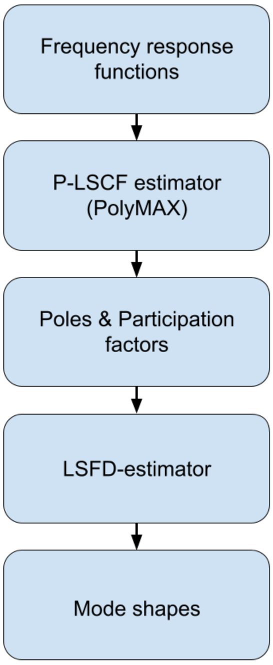

Figure 2. 7: Testlab workflow

In Figure 2.7 the workflow for using Simcenter Teslab with the PolyMAX plugin tool is shown. This workflow was used during the thesis work to estimate modes and mode shapes.

2.4.1 PolyMAX

PolyMAX is used to calculate the frequency and damping of the different modes. To be able to calculate the frequency and damping PolyMAX needs the FRFs. The PolyMAX function is an evolution of the LSCF. The estimation process is done in two steps, first a Least-square complex frequency-domain is done to the noisy FRF and once that is completed the second steps begin. In the second step the Least-square frequency-domain will be applied to the noise-reduced FRFs from the LSCF [10].

2.5 Modal Shaker Rig Test

The purpose of the rig test is to be able to simulate different road conditions and different scenarios where the points of interest are exposed to vibration. These parts or points are instrumented with acceleration sensors called accelerometers. The system is then subjected to an exciting force produced by an electromagnetic shaker called the modal shaker.

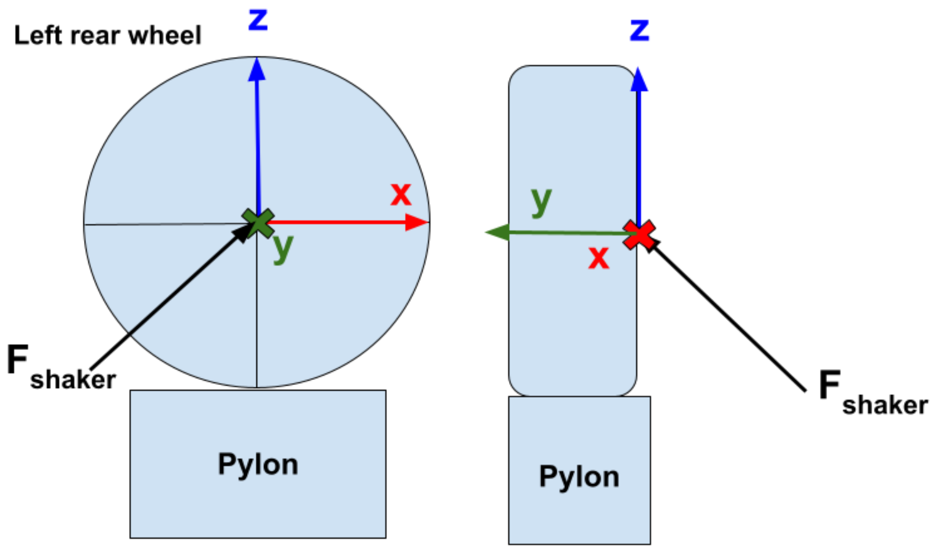

Figure 2. 8: Modal shaker rig test

As shown in Figure 2.8 the modal shaker is attached to one of the wheel bolts crossing the center of the wheel. The attachment point is instrumented with a force sensor to register an exciting force reference signal. The exciting force signal is a linear chirp sine signal which can be described as:

$$F(t) = A \cdot sin(\varphi(t)) \tag{2.19}$$

Where:

$$\varphi(t) = 2 \cdot \pi \cdot f(t) = 2 \cdot \pi \cdot (6 + r \cdot t) \tag{2.20}$$

$$\varphi(t) = 2\pi \int_0^t (6 + r \cdot t)dt = 2\pi(6t + \frac{rt^2}{2}) \tag{2.21}$$

And r is the sweep rate. A chip signal is a signal with increasing or decreasing frequency, in this rig test the evaluated results frequency interval is 6-60Hz though in reality the modal shaker can not start at the frequency 6Hz or end exactly at 60Hz which leads to some noise in the beginning and end of the measured data. The linear chirp signal can be described as:

$$f(t) = ct + f_0 \tag{2.22}$$

Where $f_0$is the frequency at time $t = 0$while c is the constant chirp rate calculated as:

$$c = \frac{f_1 - f_0}{T} \tag{2.23}$$

and f_1 is the final frequency while T is the sweep time between $f_0$and f_1 [11].

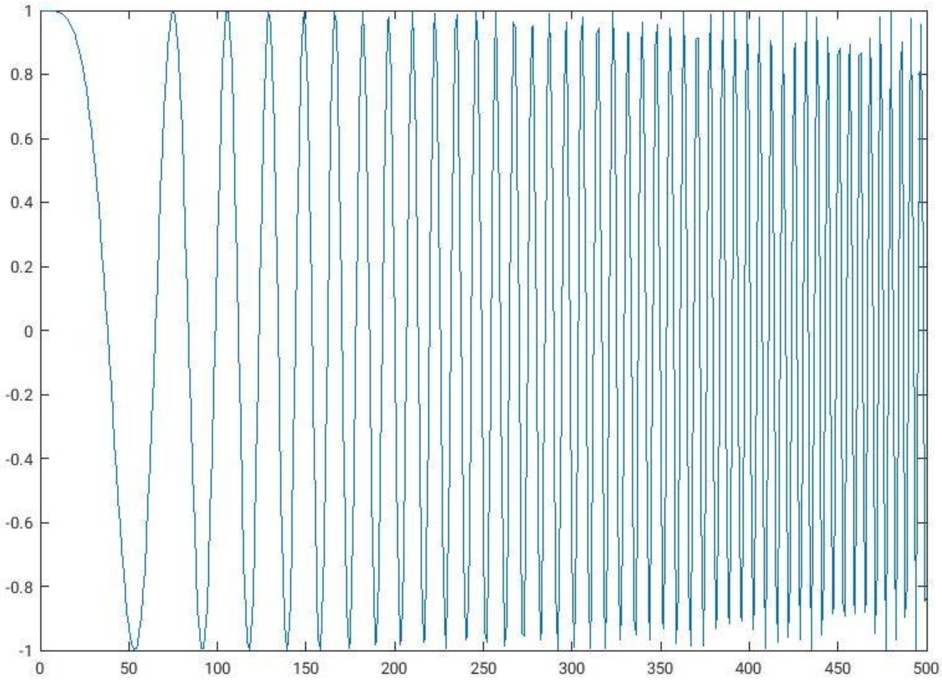

Figure 2. 9: Example of chirp signal

In Figure 2.9 it is shown what an example of the chirp signal used in the rig test with increasing frequency might look like. The example has an initial frequency of 0Hz and a target frequency of 60Hz its target time is 32 seconds with a sample rate of 128 and 500 number of samples per frame.

3. Methodology

The chapter Methodology contains explanations and information regarding how the different steps of the thesis work was conducted, starting with the original rig test followed by the simulation event and MATLAB usage. The workflow of the thesis work is displayed in Figure 3.1 as a flowchart.

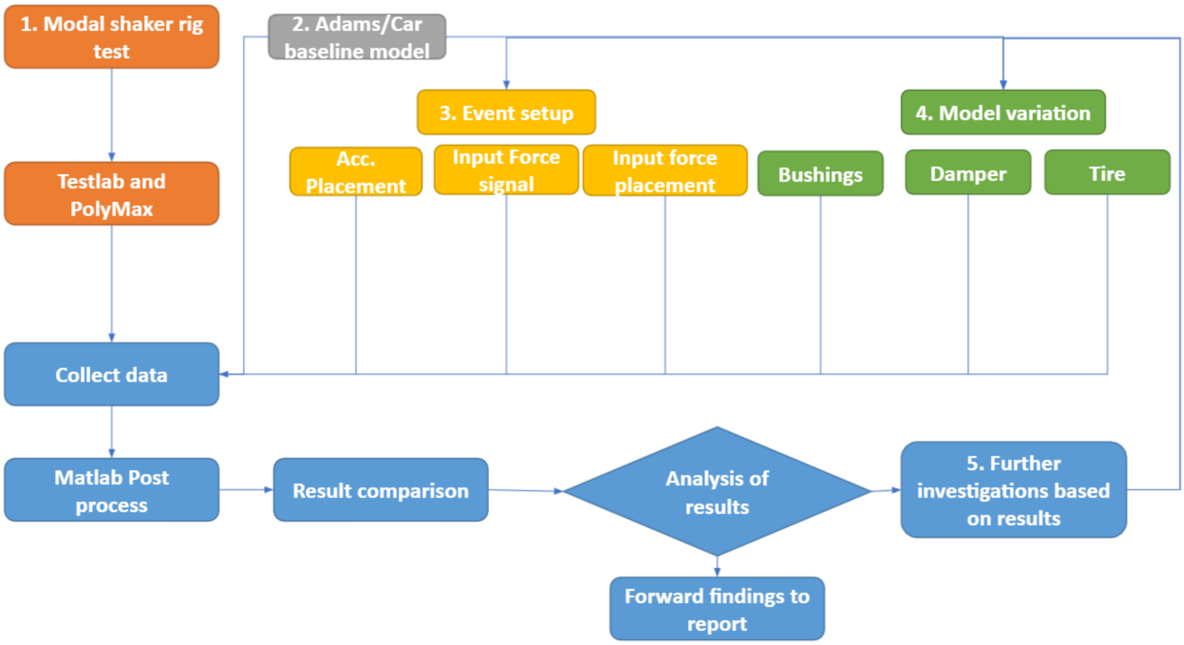

Figure 3. 1: Workflow chart

The work starts with a modal shaker rig test in order to gather credible reference data which relate to the real behavior of the test object's suspension when subjected to an exciting force at different frequencies. After the rig test a simulated modal shaker event is done using the Adams Car software. The FRFs, natural frequencies and damping are then calculated using the MATLAB function modalfrf and modalfit, then the results are compared in a data comparison to establish modeling or setup aspects for the simulated modal shaker event that are in need of further investigation. The changes are then made and further simulations are performed. All modeling or setup aspects that are found are forwarded into this final report.

3.1 Rig test

The rig test was conducted at Hällered Proving Grounds and is done with a modal shaker. The test is conducted in seven cycles and an approximation is executed from those seven cycles. When the test was finished the raw data was collected and then transferred to Testlab for analysis.

3.1.1 Setup

First the car was raised with a lift to be able to place the accelerometers on the different parts and points that are of interest to measure. When the accelerometers were placed the car was lowered onto four pylons to be able to attach the modal shaker. After the equipment is set up, the testing begins. The modal shaker starts to excit force into the wheel in the range from 100N to 800N with steps of 100N, 200N, 400N, 600N, 800N and with frequency from 6Hz to 60Hz for each step.

3.1.2 Rig test equipment

In Table 3.1.2 the equipment that was used during the rig test are specified.

Table 3. 1: Equipment used

Product Name	Supplier	Part/Type	Note
LMS frequency analyser	[redacted]	[redacted]	[redacted]
Modal Vibration Test System	[redacted]	[redacted]	[redacted]
Blower	[redacted]	[redacted]	[redacted]
Accelerometers	[redacted]	[redacted]	[redacted]
Force sensor piezo-electric	[redacted]	[redacted]	[redacted]
Cable for Accelerometers	[redacted]	[redacted]	[redacted]

3.1.3 Accelerometer placement

Placement of accelerometers can be found in appendix A - Pointset for Accelerometers.

3.1.4 Simcenter Testlab & PolyMAX

The software simcenter Testlab with the plugin tool PolyMAX was used in order to estimate and animate modes. These are useful tools in order to establish a mode's characteristics such as vertical or longitudinal tendencies for a better understanding of the estimated modes.

3.2 Adams car simulation

3.2.1 Modal analysis event

The modal analysis event in Adams/car was used to run the simulations. This event needed some configurations in order to provide the ability to change the placement of input force and to use the input force data measured in the rig test. These modifications were made in order to study the effect that force placement and input force characteristics had on the mode correlation.

3.2.1.1 Create accelerometer request

Adding a new accelerometer request to Adams/car was done with a script that specified the new accelerometer's name and position (see appendix A). In the script, all the accelerometer requests were added in predefined markers in the model i.e. close to, but not exactly where the accelerometers were placed in the physical rig test. All the accelerometers used during the rig test were added.

3.2.1.2 Accelerometer placement

To change the placement of accelerometers created to be exactly where the physical accelerometers were placed a second request script was made (see appendix A). This script specified the coordinates for the actual position for each accelerometer, created a new marker and placed the accelerometer in that position.

3.2.1.3 Force placement

The point at which the force was applied during the first simulations were at first in the center of the wheel but to receive a more correlated result the force was moved to more represent the placement from the rig test. The point of interest was moved from the center of the wheel to the wheel bolt located just above. This was not exactly where the force was applied during the rig test but was closer to the actual point.

3.2.1.4 Force input

The force used in the first four simulations were set to a fixed value, 800N for the high amplitude and 100N for the low amplitude. For the next simulations the force used where changed to the force that was measured during the physical rig test.

3.2.2 Model replacements

The most common assembly model used during the work was the [redacted] or ".ride" assembly. This assembly in its original form with no replacements done has the following setup:

PART	MODEL
Tire:	Ftire
Damper:	Viscous damper
Bushings:	MXmount in all relevant bushings

The other assembly model that was used is called the [redacted] or ".Vdyn" assembly. This assembly with no replacements has the following original setup:

PART	MODEL
Tire:	PAC2002
Damper:	Viscous damper
Bushings	STD

3.2.2.1 Replacement of bushing model

For the simulations several different bushings were replaced from the MXmount model to a less complex standard model. Bushings subjected to replacement were the top mount, lower control arm and rear subframe bushings. The replacement bushings were as following:

Topmount	[redacted]
LCA	[redacted]
RSF	[redacted]

Redacted

Figure 3. 2: Topmount, LCA_pt3 and RSF bushing positions

The position in the Adams Car assembly of the lower control arm, topmount and rear subframe bushings is shown in Figure 3.2.

3.2.2.2 Replacement of damper model

The replacement of the damper model was done mainly to observe the two different models impact on the vertical mode. To replace the standard damper model “viscous damper” in the VCC model [redacted] to the “advanced damper model” one should add some additional files into the [redacted] and “common.cdb” directory. The files that were changed were the following:

[redacted]

[redacted]

Figure 3. 3: Viscous model vs Advanced model

3.2.2.3 Replacement of tire model

To compare the correlation's dependency of tire models, the two models Ftire and PAC2002 were used. The [redacted] assembly uses the Ftire model and was therefore the assembly chosen for simulating with the Ftire. Likewise another model that uses the PAC2002 tire model was used to investigate its impact called [redacted].

3.2.2.4 Low fidelity model in Tire Damper and Bushings

In order to achieve a more overall view of how the standard bushing model together with the PAC2002 tire model and viscous damper model affect the fidelity levels, the [redacted] assembly was set to bushing setup 1, in other words all the bushing models were changed to the standard model.

3.2.2.5 LCA STD bushing stiffness factor

For evaluating the standard bushing model's capability to take into account frequency and amplitude dependent behaviour and to study how the change in stiffness factor of the LCA bushing affect the results, different scale factors were tested. The stiffness scale factors that were tested were 2,3 and 4 times the normal level for the lower control arm bushing in the y-direction.

3.3 MATLAB

For evaluating the correlation between data collected from the Rig test and Adams/Car simulations the MATLAB functions modalfrf and modalfit were used. The modalfrf function takes time response data as input and calculates a frequency response function. The modalfit function takes the calculated frequency response function provided by modalfrf and calculates estimated modes. For more information regarding the modalfrf and modal-fit functions see 2.3 MATLAB functions

3.3.1 Import data for evaluation

Data was imported from both physical measurements and Adams Car. The data in question are the time response signals for each accelerometer which was extracted from both Testlab and Adams car's post processor into tables. As well as the exciting force signal in order to calculate the frequency response function and modes in MATLAB using modalfrf and modalfit.

4. Results

The results chapter consists of four comparisons, each comparison contains a number of simulations that were done during that time period. The results from these simulations were compared to the reference results and in some cases each other in order to evaluate the changes made for each simulation. Each comparison starts off with a table that describes the modal shaker event setup followed by a table that gives an explanation of the simulations. After these explanatory tables the results are presented with text, tables and bar graphs.

4.1 Comparison 1 - Standard setup and parameterization

Table 4. 1: Simulation event setup comparison 1

Simulation nr	Frequency Start, Hz	Frequency End, Hz	Frequency Sweep Ratio, Hz/s	Force Placement Offset Y	Force Placement Offset Z	Acceleromet er Placement	Input Force
1	0	64	2.0	0	0	Standard	High: 800N Low: 100N

Table 4. 2: Simulation descriptions comparison 1

Simulation nr	Description
Simulation 1	Standard settings

4.1.1 Results of standard simulation

Correlation comparison of the standard setup simulation event and real rig test shows that further modifications need to be done in order to reach sufficient fidelity levels. Mode two is the only mode where the physical measurement result and simulation result frequencies differ within the tolerance of 0.1 however the damping factor’s correlation is not acceptable. Investigation regarding factors that are not directly a part of the Volvo Cars Cooperation assembly [redacted] but more of a simulation event setup will be done, such modifications are accelerometer placement, force placement and input force characteristics.

In Table 4.3 the results show that the estimated modal frequency provided by the MATLAB function modalfit and the modal frequency from using PolyMAX have very good correlation. However, the damping factors found have large differences this could be an result of how the two different methods calculate the damping factor.

Table 4. 3: Estimated modes A100N comparison 1

Simulation nr.	**Frequency, Hz**						**Damping, %**				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	0.5%	0.9%	1%	0.7%	0.3%		2%	30%	89%	38%	36%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 1 vs Testlab	-16%	0%	-1%	-3%	-		-32%	-127%	61%	-42%	-

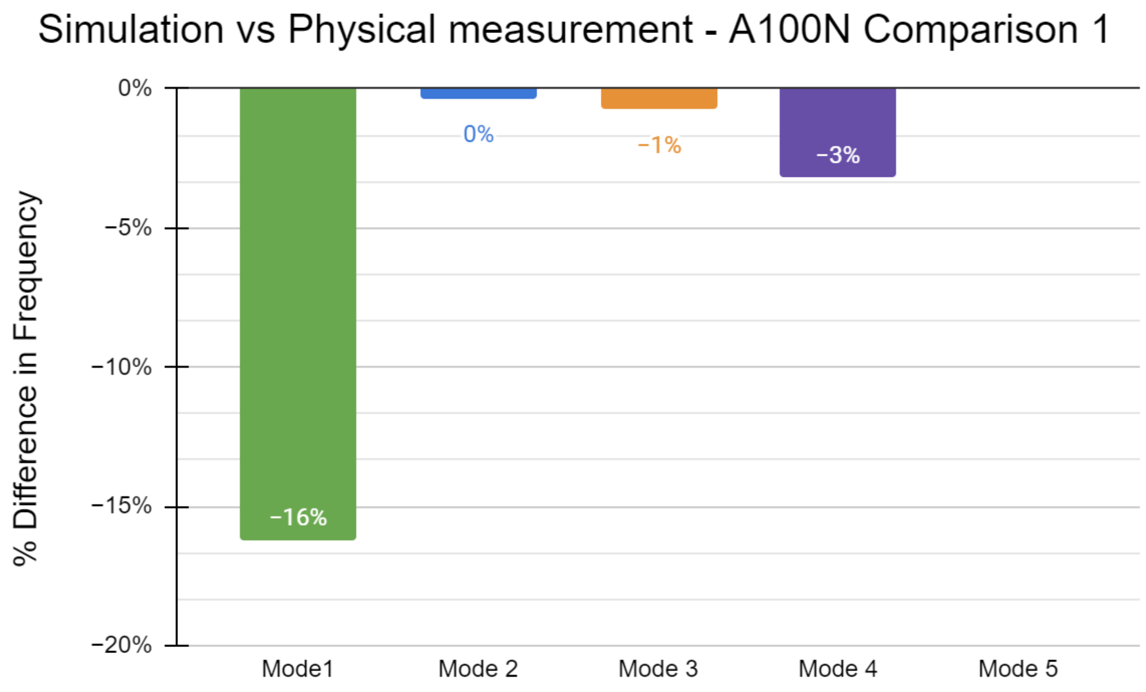

Figure 4. 1: Frequency difference A100N comparison 1

The bar chart specifies the percentage difference for the found mode's frequencies between the first simulation and the reference that is referred to as "physical measurement" which is the modal parameters found for the rig test using MATLAB evaluation.

Table 4. 4: Estimated modes A800N comparison 1

Simulation nr.	**Frequency, Hz**						**Damping, %**				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	1%	3%	-	1%	0.4%		-2%	36%	-	62%	31%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 1 vs Physical measurement	-10%	0%	-2%	1%	28%		8%	2%	78%	-37%	24%

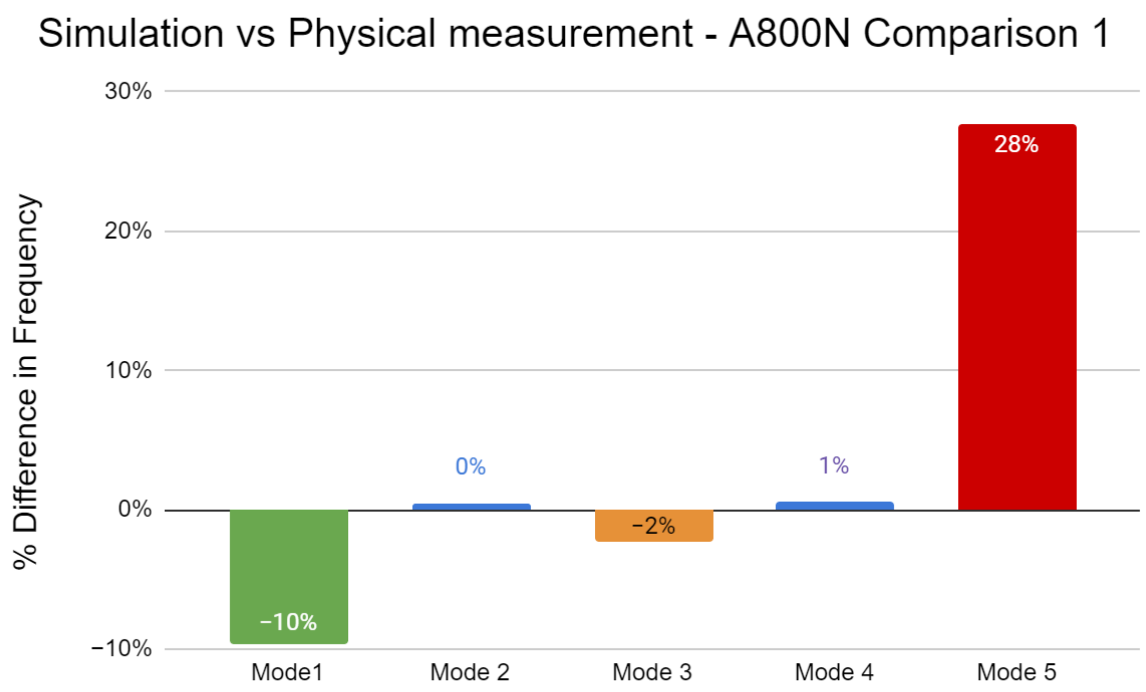

Figure 4. 2: Frequency difference A800N comparison 1

As seen in Figure 4.1 and Figure 4.2 the standard simulation is able to capture a sufficient frequency correlation for the vertical mode (mode 2) which is good. However the other modes are still far off.

4.1.2 Frequency sweep ratio

After conducting simulations with a frequency sweep ratio of 0.5 respectively 2.0 and comparing the modal parameter estimations, the conclusion is that a higher simulation sweep ratio will not affect the outcome in such a manner that it will be considered as a probable cause of divergence. This because the deviations frequency are around 0-1% between the two different ratios. Table 4.5 shows the modes found for the simulated data using the modalfrf and modalfit function in MATLAB at different frequency sweep ratios. This result gave the possibility to use the same frequency sweep ratio during the simulated modal shaker test as in the real rig test.

Table 4.5 specifies the percentual difference between the two different sweep ratios compared to the physical measurements.

Table 4. 5: Modes of different frequency sweep ratio

	Adams sweep ratio 0.5		Adams sweep ratio 2.0	
	Frequency, Hz	Damping,%	Frequency, Hz	Damping,%
Mode 1	-9.65%	[redacted]	-9.65%	[redacted]
Mode 2	0.33%	[redacted]	0.44%	[redacted]
Mode 3	-2.7%	[redacted]	-2.3%	[redacted]
Mode 4	0.56%	[redacted]	0.53%	[redacted]
Mode 5	-	[redacted]	27,7%	[redacted]

4.2 Comparison 2 - Modified accelerometer and force placement

Table 4. 6: Simulation event setup comparison 2

Simulation nr	Frequency Start, Hz	Frequency End, Hz	Frequency Sweep Ratio, Hz/s	Force Placement Offset Y	Force Placement Offset Z	Accelerometer Placement	Input Force
2	0	64	2.0	0	0	Moved	High: 800N Low: 100N
3	0	64	2.0	86	55	Standard	High: 800N Low: 100N
4	0	64	2.0	86	55	Moved	High: 800N Low: 100N
5	0	64	2.0	86	55	Moved	Physical measurement data

Table 4. 7: Simulation description comparison 2

Simulation nr	Description
Simulation 2	Moved accelerometer placement to match real testing
Simulation 3	Moved force placement to match real testing
Simulation 4	Moved force and accelerometer placement to match real testing
Simulation 5	Moved force and accelerometer placement to match real testing. Input force replaced with measured force from physical measurement to match real testing.

As shown in Table 4.8 and Table 4.9 the simulation test still variates from the rig test. However when the accelerometers and the force placement were moved, the result was expected to be more similar to the physical measurement results. The actual results received were pretty much indifferent from the previous results generated in the simulations. All the modifications made this far were kept for further work to better match the setup at Hällered proving ground, regardless of its minimal impact on correlation between real and simulated testing.

4.2.1 Placement of accelerometers

In the second comparison the accelerometers were placed to be as close as possible to where they were placed during the rig test. With a more accurate accelerometer placement we can conclude that no significant improvement was gained by moving the accelerometers. The simulated result is as similar to the result from physical measurement as it was before moving the accelerometers.

4.2.2 Input signal

For simulation five the measured input force registered from the rig test was used as input force in the modal shaker event in Adams Car. This resulted in minor frequency changes of the modes.

4.2.3 Force placement

To further make the simulated test in Adams Car similar to the rig test at Hällered proving ground, the location of the applied force into the wheel was moved from the center of the wheel to one of the wheel bolts to replicate the placement at Hällered proving ground. By moving the entry for the force going into the wheel a more similar result to the physical measurement result was expected, but same as for the accelerometers the result was basically the same as before moving the force from the center to one of the wheel bolts.

4.2.5 Model changes

Changes to the model will be made for comparison three to investigate which components or parameters that can be changed for better correlation. The current Ftire model will be swapped to a simpler version called PAC2002 for evaluating the tire model’s impact on correlation. The LCA and top mount bushings will be replaced with a bushing model called STD. The damper model will be changed to an advanced damper model constructed to take more parameters into account.

Table 4. 8: Estimated modes A100N comparison 2

Simulation nr.	Frequency, Hz						Damping, %				
	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5		Mode 1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	0.5%	0.9%	1%	0.7%	0.3%		2%	30%	89%	38%	36%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 2 vs Physical measurement	-16%	-2%	-5%	-3%	-		-33%	-111%	68%	33%	-
Diff: 3 vs Physical measurement	-17%	3%	-1%	-3%	24%		-33%	-154%	74%	-42%	-34%
Diff: 4 vs Physical measurement	-17%	2%	0%	-2%	-		-34%	-152%	75%	-41%	-
Diff: 5 vs Physical measurement	-15%	5%	1%	-3%	-		-55%	-169%	76%	-50%	-

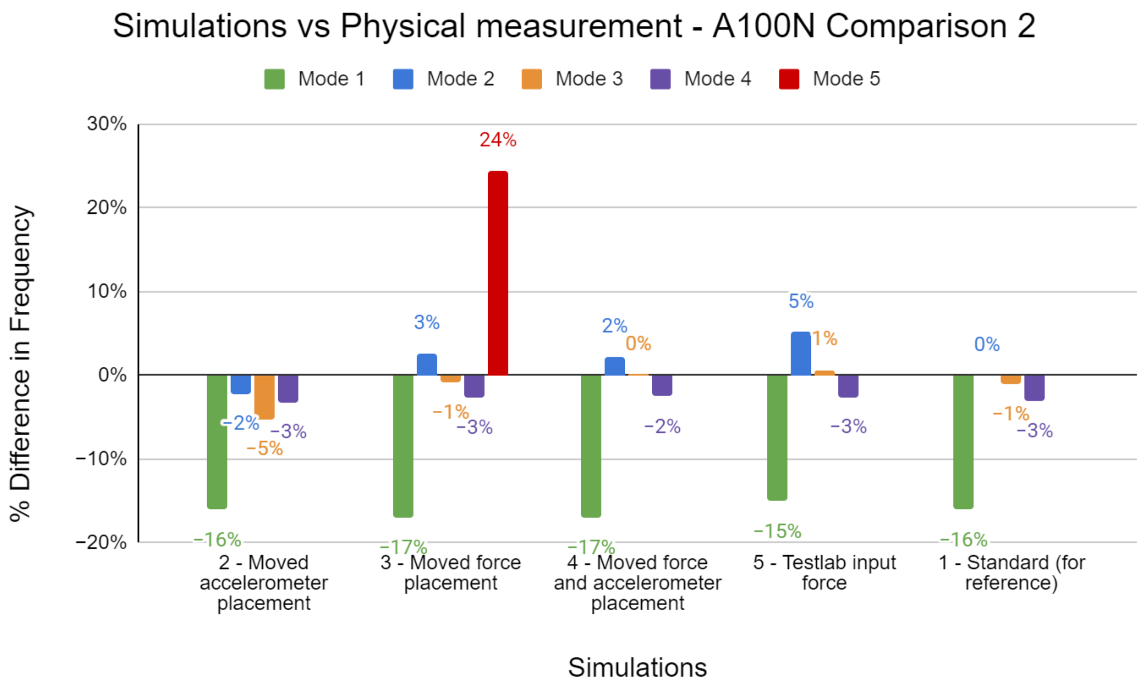

Figure 4. 3: Frequency difference A100N comparison 2

For comparison two the simulated modal event setup modifications only contributed with smaller changes on the longitudinal and vertical mode (mode 1 & mode 2). Combining all the changes and swapping the input force as were done in simulation 5 and gave 1% better correlation in mode 1 but 5% worse correlation in mode 2 for the low force amplitude of 100N.

Table 4. 9: Estimated modes A800N comparison 2

Simulation nr.	**Frequency, Hz**						**Damping, %**				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	1%	3%	-	1%	0.4%		-2%	36%	-	62%	31%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 2 vs Physical measurement	-10%	1%	-6%	0%	-		-10%	4%	89%	-32%	-
Diff: 3 vs Physical measurement	-9%	0%	-4%	1%	-		-8%	1%	87%	-38%	-
Diff: 4 vs Physical measurement	-9%	1%	-4%	1%	-		-8%	1%	86%	-38%	-
Diff: 5 vs Physical measurement	-9%	-2%	-2%	2%	29%		-30%	-15%	85%	-40%	32%

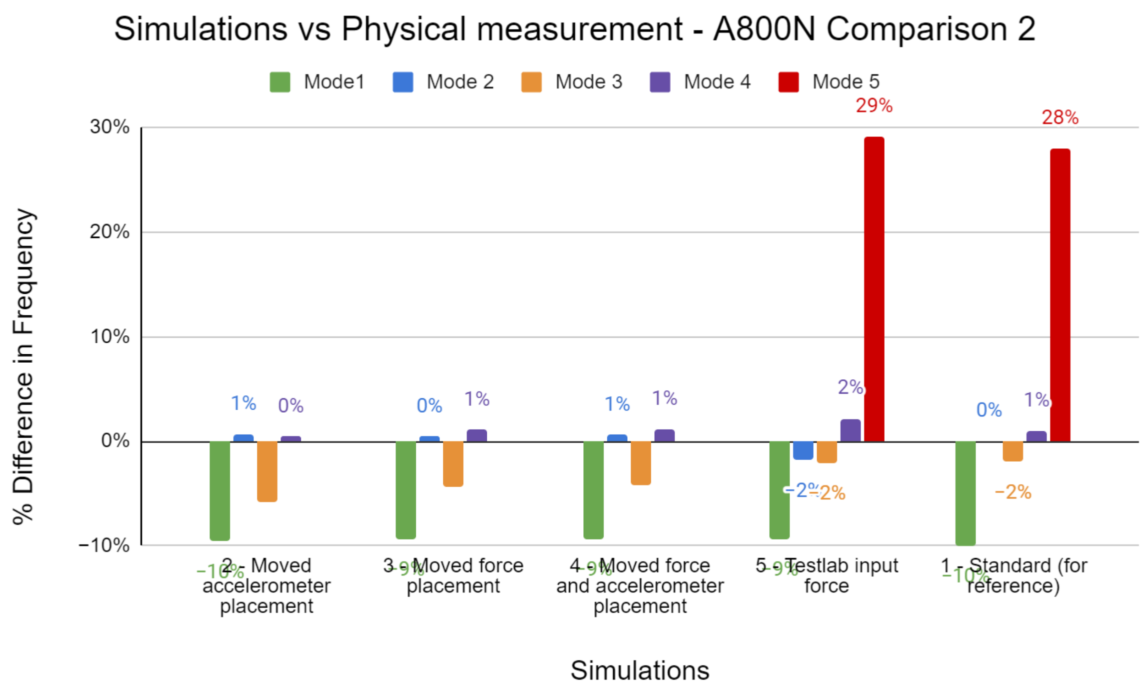

Figure 4. 4: Frequency difference A800N comparison 2

The results for the higher force amplitude of 800N regarding mode 1 are similar to the once received for the lower force amplitude. An interesting finding is that the changes made in simulation 1,2 and 3 did not have any significant impact on mode 2 while changing the input force made a small increase in frequency. All the changes made in simulation 5 were kept for further simulations in order to better match the real rig tests.

4.3 Comparison 3 - Impact of bushing, dampers and tire models

Table 4. 10: Simulation event setup comparison 3

Simulation nr	Frequency Start, Hz	Frequency End, Hz	Frequency Sweep Ratio, Hz/s	Force Placement Offset Y	Force Placement Offset Z	Accelerometer Placement	Input Force
5	0	64	2.0	86	55	Moved	physical measurement data
6	0	64	2.0	86	55	Standard	physical measurement data
7	0	64	2.0	86	55	Moved	physical measurement data
8	0	64	2.0	86	55	Moved	physical measurement data
9	0	64	2.0	86	55	Moved	physical measurement data

Table 4. 11: Simulation description comparison 3

Simulation nr	Description
Simulation 5	Moved force and accelerometer placement to match real testing. Input force replaced with measured force from physical measurement to match real testing.
Simulation 6	Swapped the F-tire model to PAC 2002.
Simulation 7	Top mount bushing changed to std model.
Simulation 8	LCA bushing changed to std model.
Simulation 9	Advanced damper model.

4.3.1 Tire model replacement

Replacing the Ftire model with the less complex model PAC2002 made the most difference in the third mode compared to simulation 5 and had very little impact on the first and second mode.

4.3.2 LCA and top mount bushing replacement

The front lower control arm bushing of model type MXmount was replaced with a STD bushing model to investigate its impact on mode one. When replacing it with the STD bushing model, the correlation between the longitudinal modes (mode one) found with physical measurement data and simulation 5 data got worse. This outcome was expected due to the STD models linear behavior and further investigations regarding the stiffness scale factor will be conducted to find a reasonable level for better correlation

4.3.3 Advanced damper model implementation

The advanced damer model made the biggest impact on the vertical mode (mode two) as one might expect but not necessarily a more accurate result. There is clearly a need for further investigation regarding its parameterization in order to fully exploit its beneficial features. The advanced damper model was specially made to produce good correlation between the four post shaker test and simulation. With that said, it should be a more suitable model for this type of correlation work with the right settings.

4.3.4 STD bushing stiffness parameterization

After conducting several model changes the lower control arm bushing replaced with a STD model seemed to be the configuration with the largest negative impact of mode one. This suggests that the bushing's parametrization might need some further modification in order to find factors that have negative influence on the frequency correlation. And that the LCA bushing is an important aspect for accomplishing good fidelity in the longitudinal mode.

Table 4. 12: Estimated modes A100N comparison 3

Simulation nr.	**Diff vs. Testlab Frequency, %**						**Damping, %**				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	0.5%	0.9%	1%	0.7%	0.3%		2%	30%	89%	38%	36%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 5 vs Physical measurement	15%	-5%	-0.6%	3%	-		-55%	-169%	76%	-50%	-
Diff: 6 vs Physical measurement	-18%	-8%	-26%	-5%	4%		-34%	-104%	88%	-17%	-19%
Diff: 7 vs Physical measurement	-15%	-5%	0%	-3%	24%		-37%	-48%	-78%	-45%	-34%
Diff: 8 vs Physical measurement	-25%	-8%	-1%	-6%	24%		-50%	-134%	79%	-82%	10%
Diff: 9 vs Physical measurement	-15%	10%	-2%	-4%	15%		-54%	-265%	56%	-86%	-871%

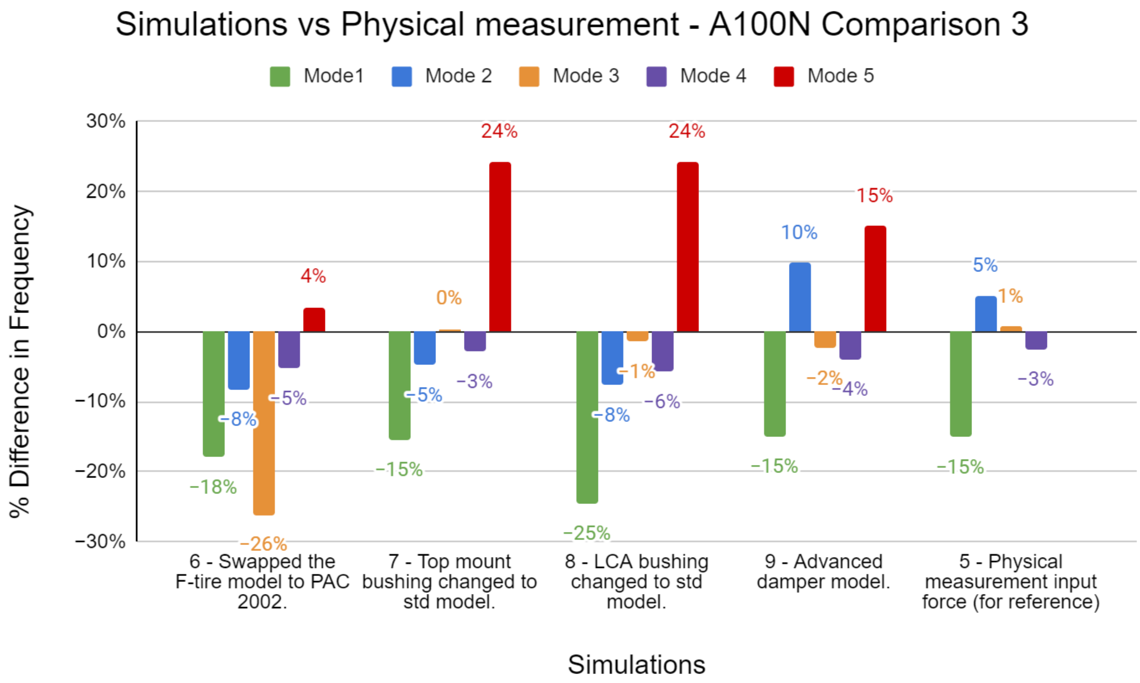

Figure 4. 5: Frequency difference A100N comparison 3

Figure 4.5 tells us that the expectations that the LCA bushing of type MXmount replaced with the STD model would have a large impact on the longitudinal mode, mode number 2 was verified, thus the frequency difference increased by 10%.

Table 4. 13: Estimated modes A800N comparison 3

Simulation nr.	Frequency, Hz						Damping, %				
	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	1%	3%	-	1%	0.4%		-2%	36%	-	62%	31%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 5 vs Physical measurement	9%	2%	2%	-2%	-29%		-30%	-15%	85%	-40%	32%
Diff: 6 vs Physical measurement	-9%	-6%	-26%	1%	7%		-48%	-7%	93%	-88%	5%
Diff: 7 vs Physical measurement	-9%	-1%	-2%	2%	30%		-28%	29%	85%	-39%	32%
Diff: 8 vs Physical measurement	-13%	-3%	-4%	0%	-		-33%	-2%	84%	-58%	-
Diff: 9 vs Physical measurement	-9%	5%	-3%	2%	-		-32%	-21	83%	-46%	-

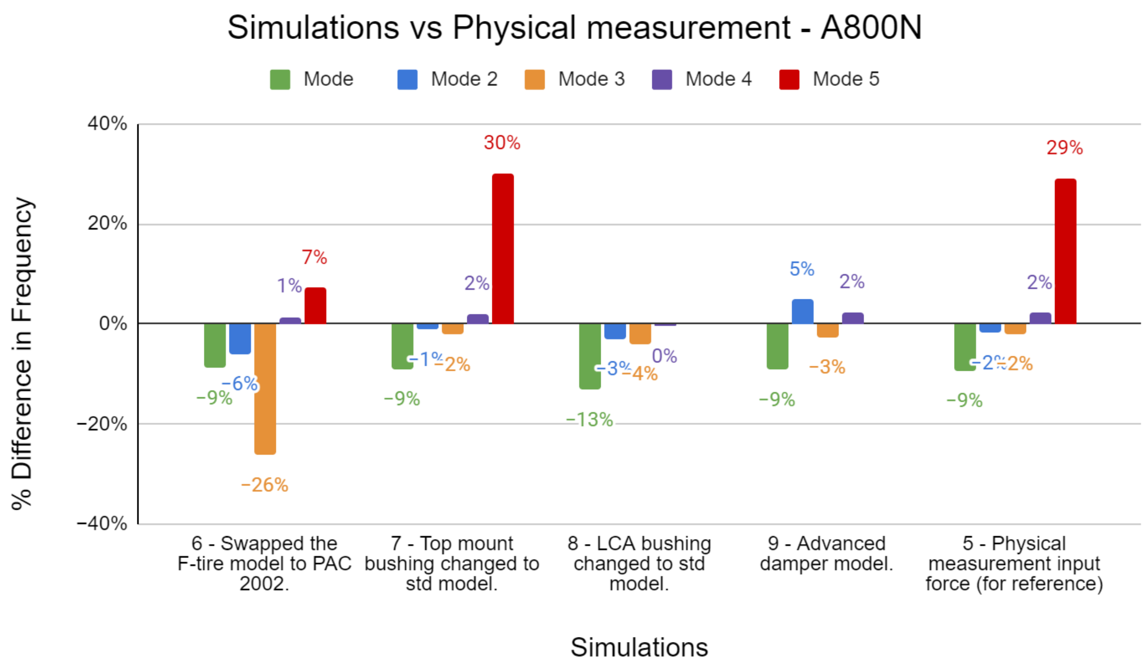

Figure 4. 6: Frequency difference A800N comparison 3

The advanced damper model and PAC2002 model were expected to change the results for the vertical mode which Figure 4.6 indicates. The PAC2002 tire model gave a higher frequency for mode 2 while the advanced damper model gave a lower one. The topmount bushing was expected to have a larger impact on the vertical mode than it did.

4.4 Comparison 4 - Stiffness parametrization of LCA bushing

Table 4. 14: Simulation event setup comparison 4

Simulation nr	Frequency Start, Hz	Frequency End, Hz	Frequency Sweep Ratio, Hz/s	Force Placement Offset Y	Force Placement Offset Z	Accelerometer Placement	Input Force
5	0	64	2.0	86	55	Moved	physical measurement data
8	0	64	2.0	86	55	Moved	physical measurement data
12	0	64	2.0	86	55	Moved	physical measurement
13	0	64	2.0	86	55	Moved	physical measurement
14	0	64	2.0	86	55	Moved	physical measurement
15	0	64	2.0	86	55	Moved	physical measurement
16	0	64	2.0	86	55	Moved	physical measurement

Table 4. 15: Simulation description comparison 4

Simulation nr	Description
Simulation 5	Moved force and accelerometer placement to match real testing. Input force replaced with measured force from physical measurement to match real testing.
Simulation 8	LCA bushing changed to std model.
Simulation 12	Rear subframe bushing changed to std model
Simulation 13	Vdyn assembly with bushing setup 1
Simulation 14	LCA bushing changed to std model and stiffness factor : 1.2.1
Simulation 15	LCA bushing changed to std model and stiffness factor : 1.3.1
Simulation 16	LCA bushing changed to std model and stiffness factor : 1.4.1

4.4.1 Stiffness scale factor impact.

The lower control arm bushings replaced with the standard model of bushings appeared to have the largest impact on the longitudinal mode (mode one) which led to further simulations with different levels of stiffness in the longitudinal y-direction. This was done to see if a higher stiffness could contribute to better frequency correlation of the first mode, and to establish how sensitive the modal results are to the LCA bushing stiffness. A higher stiffness led to a higher frequency of the first mode as seen in Table 4.16 and Table 4.17.

4.4.2 Vdyn model with bushing setup 1

The Vdyn assembly model was used to capture the outcome of setting all bushings to standard bushings together with the tire model PAC2002 and damper model Viscous damper. When doing this one could capture to what degree the frequency correlation changes when using only models that are not frequency and amplitude dependent. The results would come to show that the MXmount bushing model and its ability to capture amplitude and frequency dependencies is important to achieve sufficient fidelity levels, even though there is a need for further investigation regarding parameterization for bushings of relevance.

4.4.3 Standard rear subframe bushings

The rear subframe bushing model has almost the same amount of impact on the longitudinal mode frequency as the LCA bushing but produces fairly good frequency correlation on the vertical mode.

Table 4. 16: Estimated modes A100N comparison 4

Simulation nr.	**Frequency, Hz**						**Damping, %**				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	0.5%	0.9%	1%	0.7%	0.3%		2%	30%	89%	38%	36%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 1 vs Physical measurement	16%	0%	1%	3%	-		-32%	-127%	61%	-42%	-
Diff: 5 vs Physical measurement	15%	-5%	-0.6%	3%	-		-55%	-169%	76%	-50%	-
Diff: 8 vs Physical measurement	-25%	-8%	-1%	-6%	24%		-50%	-134%	79%	-82%	10%
Diff: 12 vs Physical measurement	-20%	-1%	1%	-4%	-		-46%	-158%	70%	-73%	-
Diff: 13 vs Physical measurement	-32%	15%	-27%	7%	23%		83%	-82%	70%	-119%	-46%
Diff: 14 vs Physical measurement	-15%	-6%	-1%	-4%	24%		-54%	-167%	76%	-100%	-3%
Diff: 15 vs Physical measurement	-10%	0%	0%	-2%	24		-67%	-198%	74%	-204%	-11%
Diff: 16 vs Physical measurement	-6%	9%	0%	-1%	24%		-74%	-205%	73%	-202%	-18%

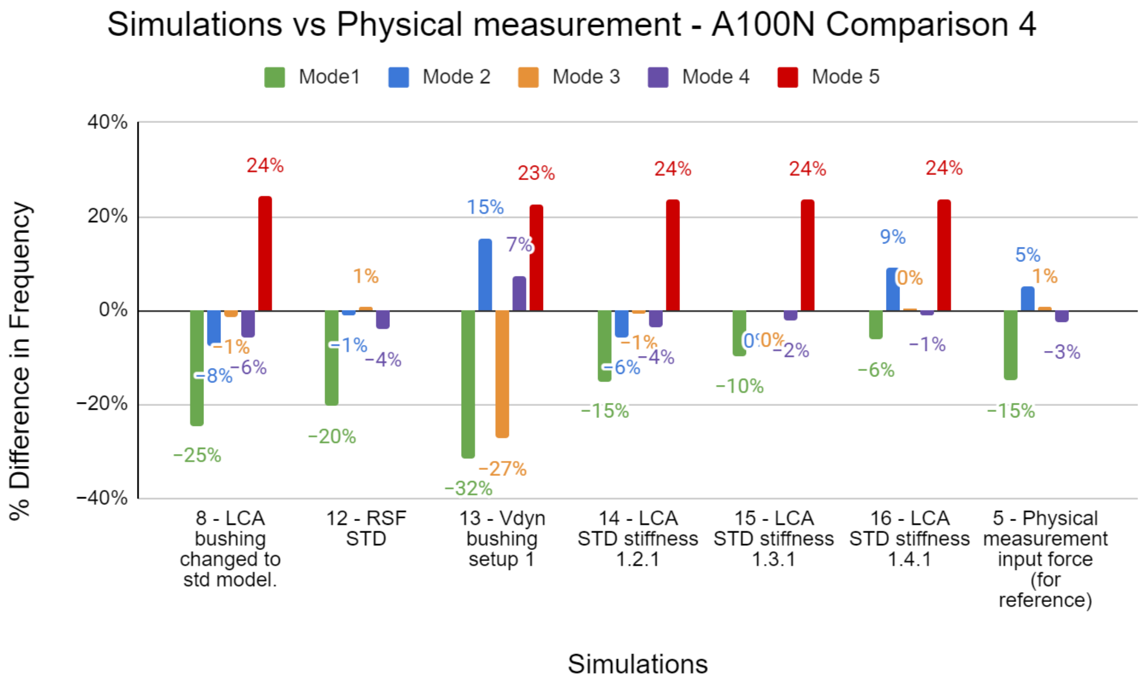

Figure 4. 7: Frequency difference A100N comparison 4

In comparison 3 the LCA bushing was established to have effect on the longitudinal mode. Therefore, further simulations with increasing stiffness scale factor in the y-direction were conducted to find the level of which the frequency of mode 1 had good correlation for the STD bushing model.

Table 4. 17: Estimated modes A800N comparison 4

Simulation nr.	**Frequency, Hz**						**Damping, %**				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5		Mode1	Mode 2	Mode 3	Mode 4	Mode 5
PolyMAX	1%	3%	-	1%	0.4%		-2%	36%	-	62%	31%
Physical measurement	0%	0%	0%	0%	0%		0%	0%	0%	0%	0%
Diff: 1 vs Physical measurement	10%	0%	2%	0%	-27%		8%	2%	78%	-37%	24%
Diff: 5 vs Physical measurement	9%	2%	2%	-2%	-29%		-30%	-15%	85%	-40%	32%
Diff: 8 vs Physical measurement	-13%	-3%	-4%	0%	-		-33%	-2%	84%	-58%	-
Diff: 12 vs Physical measurement	-13%	-1%	-2%	1%	13%		-24%	-9%	82%	-46%	-306%
Diff: 13 vs Physical measurement	-19%	23%	-27%	14%	29%		-50%	0%	82%	-101%	0%
Diff: 14 vs Physical measurement	-1%	-3%	-3%	2%	30%		-34%	-15%	82%	-75%	36%
Diff: 15 vs Physical measurement	5%	-1%	-2%	3%	29%		-38%	-32%	81%	-84%	33%
Diff: 16 vs Physical measurement	9%	3%	-2%	5%	29%		-40%	-46%	78%	-86%	31%

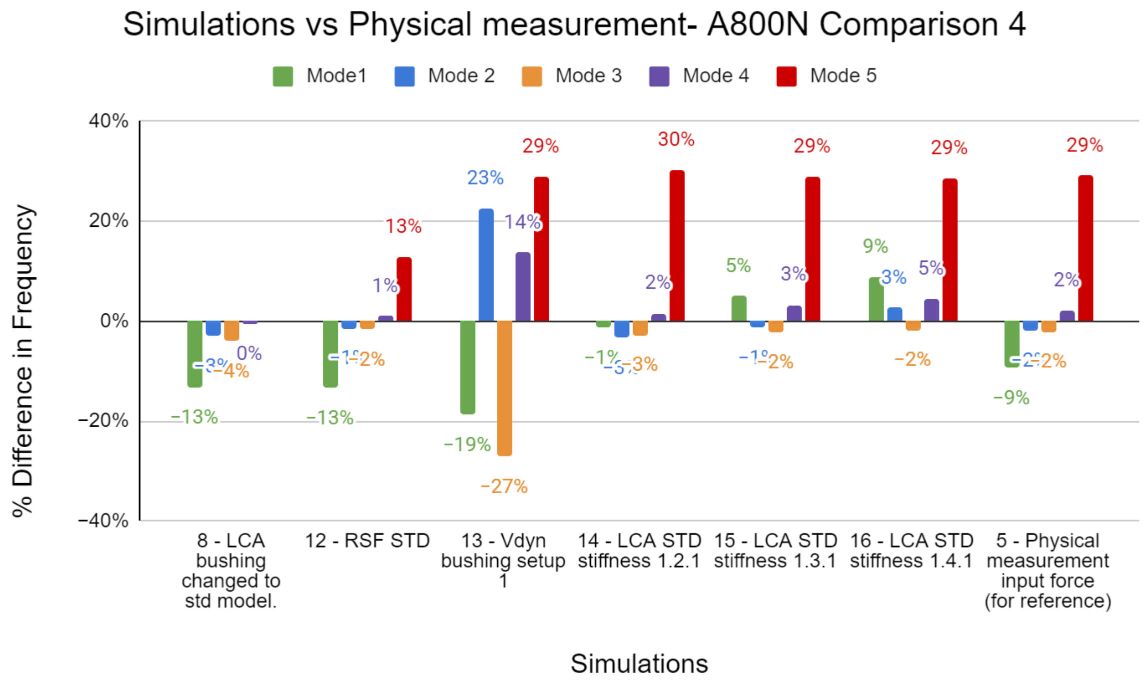

Figure 4. 8: Frequency difference A800N comparison 4

Bushing models in general were anticipated to play a large role during the thesis work and in simulation 13 all bushings in the assembly model were replaced with the STD model and results in Figure 4.7 verifies that this is the case. The rear subframe bushing and its model type is also an important aspect for good correlation in the longitudinal mode's frequency. The sweet spot was about 5 times the original stiffness factor for the low force amplitude and 2 times for the higher force amplitude. This shows the STDs incompatibility to fully model a real bushing at different force amplitudes. These results show that the model is sensitive for changes in the LCA bushing's stiffness and that with parametrization of the LCA bushing using the MXmount model good correlation in the longitudinal mode would be possible. But also, that further work needs to be done on component level to establish where and how the model should be changed.

5. Discussion

The discussion segment will be executed in the same order as the results were presented. The modal shaker test that was performed at the beginning of the thesis work was used as a reference point for all the comparisons done. In the first comparison, comparison 1 the reference values were compared against the first simulation. In the first simulation standardized tires, bushings and damper models for the ride assembly were used, the simulated results captured some amplitude and frequency dependent behaviour but not with such fidelity that it was not in need of further work. During comparison 2 some of the event setup parameters were changed with the expectation to generate more similarity to the reference results and to evaluate the simulated event setup parameters impact on the correlation of modal parameters. During the second comparison, simulation 2-5 was conducted with a few tweaks and changes. For simulation 2 the placement of the accelerometers were moved to more accurately represent the original modal shaker test done at Hällered Proving Ground. The third simulation used the original placement of accelerometers but the point where the force that is applied into the wheel was moved from the centre to one of the wheel bolts. In simulation 4 and 5 both accelerometers and the applied force were moved as they were in simulation 2 and 3. The difference of simulation 5 was that the measured force extracted from the modal shaker test at Hällered was used instead of the fixed values of 800N/100N. To further investigate and find important aspects in order to gain fidelity in the results, the simulations in comparison 3 were done with different bushing, tire and damper models. In simulation 6-9 the changes done were directed towards these models. Simulations 6, 7 and 8 were all done to be able to establish each model's respective impact on the frequency correlation. So, these 3 simulations were done with a simpler version. In Simulation 6 the tires were changed from the F-tire model to the simpler PAC2002. For the next Simulation, the Tires were changed back and this time the Top mount bushings were changed to a standard model. The last simulation done this way was simulation 8 where the LCA bushing was swapped to the standard model. The last simulation for the third comparison was done in the opposite way as the previous simulation in this comparison. Instead of swapping to a less complex model, simulation 9 was conducted with the advanced damper model. For comparison 4 we continued to use this method to see which models that should be used and their impact on modal parameters. The difference from comparison 3 was that this time changes were also made to the stiffness scale factor of the LCA bushing. Simulation 12 was done with the standard bushing type on the rear subframe bushings to verify the expectation that the rear subframe bushings model type has a large impact on the longitudinal mode. In simulation 13 the whole model was changed to the Vdyn assembly with bushing setup 1, this means that all the bushings in the assembly were swapped to standard bushings together with the PAC2002 tire model. The three last simulations were all executed with a standard LCA bushing at different stiffness scale factors to see how high the stiffness had to be in order to provide better frequency correlation.

6. Conclusion

After conducting this work there is no doubt that with the right parameters and models of important parts such as bushings, dampers and tires and further work it is possible to achieve sufficient fidelity levels with the simulated modal shaker event. Modal event setup parameters did not affect the results as expected but should be used in order to match the real modal shaker test as well as possible.

With that said, the simulated modal shaker event should not be used as a substitute to the real modal shaker test but with the right setup it can be used as a compliment for better understanding how different parts and parameters affect the natural frequency and damping.

The force placement of the simulated event is in need of further investigation regarding its expected ability to create momentum with the current position being on the top wheel bolt not crossing centre of rotation. The Ftire model provides a better representation of a tire's non-rolling properties than the PAC2002 and is recommended for further work. The advanced damper model has not been evaluated to its full potential and should be subjected to more simulations with various parametrization. The bushing model MXmount is superior to the STD model regarding capturing frequency and amplitude behaviours, this is shown during the last comparison where the stiffness scale factor for the STD model must be set differently for the low and high force amplitude level in order to produce a better correlation. Finally important modeling and setup aspects have been found and their impact has been evaluated. Especially noticeable is the lower control arm's and rear subframe's effect on the longitudinal mode and the damper model's effect on the vertical mode.

Even though the results only cover merely a fraction of all the parameters and different models that can be evaluated, this thesis work will be a good ground to further continued work in this field. The conclusion that can be drawn when evaluating these results is that the standard ride assembly with no changes is a good starting point. It provides an overview of the modal parameters but is in need of further parameterization of assembly parts that have a large impact on the modal parameter's correlation to the real test in order to provide good fidelity levels.

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Appendices

A Force Placement

*Redacted

B Placement of accelerometers

*Redacted

C Create accelerometer script

*Redacted

D Change accelerometer placement script

*Redacted

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