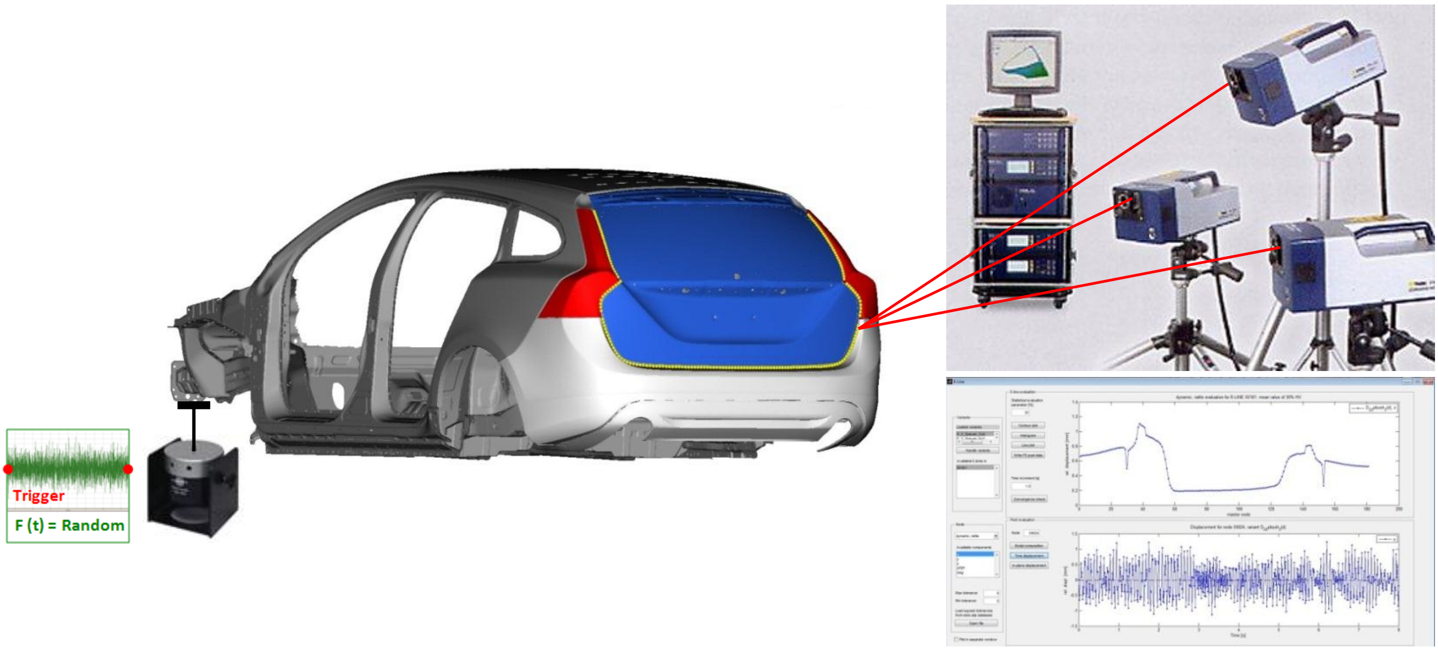

Risk of Chafing and Abrasive Wear between Exterior Part and Car Body –
CAE Simulation and Correlation with Measurement

Master of Science Thesis

Anahita Pakiman
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Department of Applied Mechanics
Division of Dynamics
Chalmers University of Technology
Göteborg, Sweden, 2014
Master Thesis no 2014:19

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Relative displacement of V60 tailgate

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Abstract

Relative displacement in closure gaps can lead to two major undesired phenomena. First, it can cause chafing and abrasive wear between the exterior components close to the visible gap. The second phenomenon is the relative displacement of rubber sealing in the gap which can cause squeak. In order to minimize the risk for these phenomena, the simulation capability has to be increased. In this study the focus is on global damping for car BIW and rubber sealing stiffness, since simulation of relative displacement is very sensitive to these parameters.

Simulations and tests on a car body structure including tailgate, rear lamps and bumper are performed. Three different test setups are used to estimate the damping for BIW and sealing stiffness. Sealing stiffness for all three directions is incorporated in the simulation model. The last test setup investigates the impact of torsional body stiffness on the relative displacement of the tailgate.

One of these test setups investigates the impact of torsional body stiffness on the relative displacement of the tailgate.

The E-LINE approach is used to calculate the relative displacements. An 8-second pseudorandom signal is applied to the structure and the response in time domain both from test and simulation are used for the correlation. The correlation will be between the relative displacement of tailgate and exterior parts, between the test and simulation in time domain.

It is shown that the updated global damping for the BIW and sealing stiffness are increasing the accuracy of relative displacement simulation in the virtual development phase.

Key words: Relative displacement, time domain, E-LINE method, tailgate, exterior parts, test measurement, correlation

Acknowledgements

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Abbreviations

BIW – Body in White: This term refers to the stage in automobile manufacturing in which only the body's sheet metal parts are welded and no components such as window, bumper, motor, doors and so on are assembled.

NVH – Noise Vibration Harshness: It is a department at Volvo Car Corporation that studies the characteristic of noise and vibration in cars.

VCC – Volvo Car Corporation: Car company manufacturing cars since 1927 in Gothenburg- Sweden.

E–LINE – Evaluation line: A method to calculate the relative displacement in local coordinate system in time domain.

ISNVH – International Styrian Noise Vibration & Harshness Congress: It is the European automotive noise conference that takes place in Graz-Austria.

ODS – Operational Deflection Shape: It is a term used for determination of the vibration pattern of a structure due to an excitation.

FRF – Frequency Response Function: The FRF curves describe the input-output relationship between two points on structure as a function of frequency.

SSE – Sum of Square of Error: This is a mathematical approach to achieve the best fitted curve in Matlab to any reference curve. The formula for SSE is as following;

$$\mathrm{SSE} = \sum_{i=1}^{n}(X_i - \overline{X})^2$$

Where the X_i refers to observation values and $\overline{X}$ is the group's mean.

1 Introduction

Squeak and Rattle in a car clearly reduce the premium perception of a car. They are caused by relative displacements [1]. Relative displacement may also cause chafing and abrasive wear between parts. Squeak occurs in case of friction between two surfaces which is the result from slip-stick phenomenon [2]. Rattle occurs when the dynamic relative movement between two components is larger than a defined nominal gap size. It happens when two surfaces along a gap move perpendicular to each other because of different reasons such as lack of stiffness or insufficient tolerances [2].

For tailgate, rattle phenomenon can cause chafing and abrasive wear between exterior components close to the tailgate's visible gap. Furthermore, relative displacement of the rubber sealing in the gap can cause squeak and consequently a friction induced noise.

In order to simulate the risk for these kinds of unwanted phenomena at an early stage, accurate simulation models are needed. A model which can estimate the relative displacement in order to indentify the critical areas in terms of squeak and rattle (and also wear and chafing) risks. When it comes to the tailgate closure gap, the relative displacement simulation is very sensitive to global damping (for the Body in White or BIW) and rubber sealing stiffness. These parameters will here be obtained by correlating the relative displacement from test measurements and simulation in time domain. Relative displacement will be measured due to a random transient excitation which is the same signal in both simulation and test measurement.

Relative displacement between tailgate and exterior parts can be virtually assessed by using a simulation method called E-LINE. This method evaluates the relative displacement in local coordinate system between two parts due to any signal. E-LINE method evaluates the relative displacement in time domain due to the fact that frequency domain does not include phase information which is essential in relative displacement assessment. This method was presented for the first time in 2010 at the ISNVH conference in Graz by the name of SAR-LINE™. How this new simulation method can be correlated to a real squeak or rattle issue was shown in a second paper in 2012 [3] also on the ISNVH conference.

The boundary condition is free-free for the car body in this analysis. The location and direction of the excitation will be optimized in order to get a significant relative displacement in the closure gap without contact (contact cannot be measured in test). A study will be performed to investigate the best position and direction (in terms of achieving an acceptable response from structure) for the excitation. There are several other parameters that can have an impact on sealing behavior such as temperature, humidity and aging which are not within the scope of this project.

1.1 Purpose

The main purpose of this thesis is to estimate the global damping for the BIW and rubber sealing stiffness for the tailgate of a V60 Volvo car. In order to achieve these parameters a correlation in time domain between test measurement and simulation will be performed.

Relative displacement results in time domain will be the mean for correlation. E-LINE method and 3D laser vibrometer [4] are the main tools for simulation and test.

1.2 Problem

In order to avoid wear/abrasion, squeak/rattle in the exterior components already in the virtual phase a reliable model with accurate damping value for the BIW (global damping) and sealing stiffness is needed.

1.3 Methodology

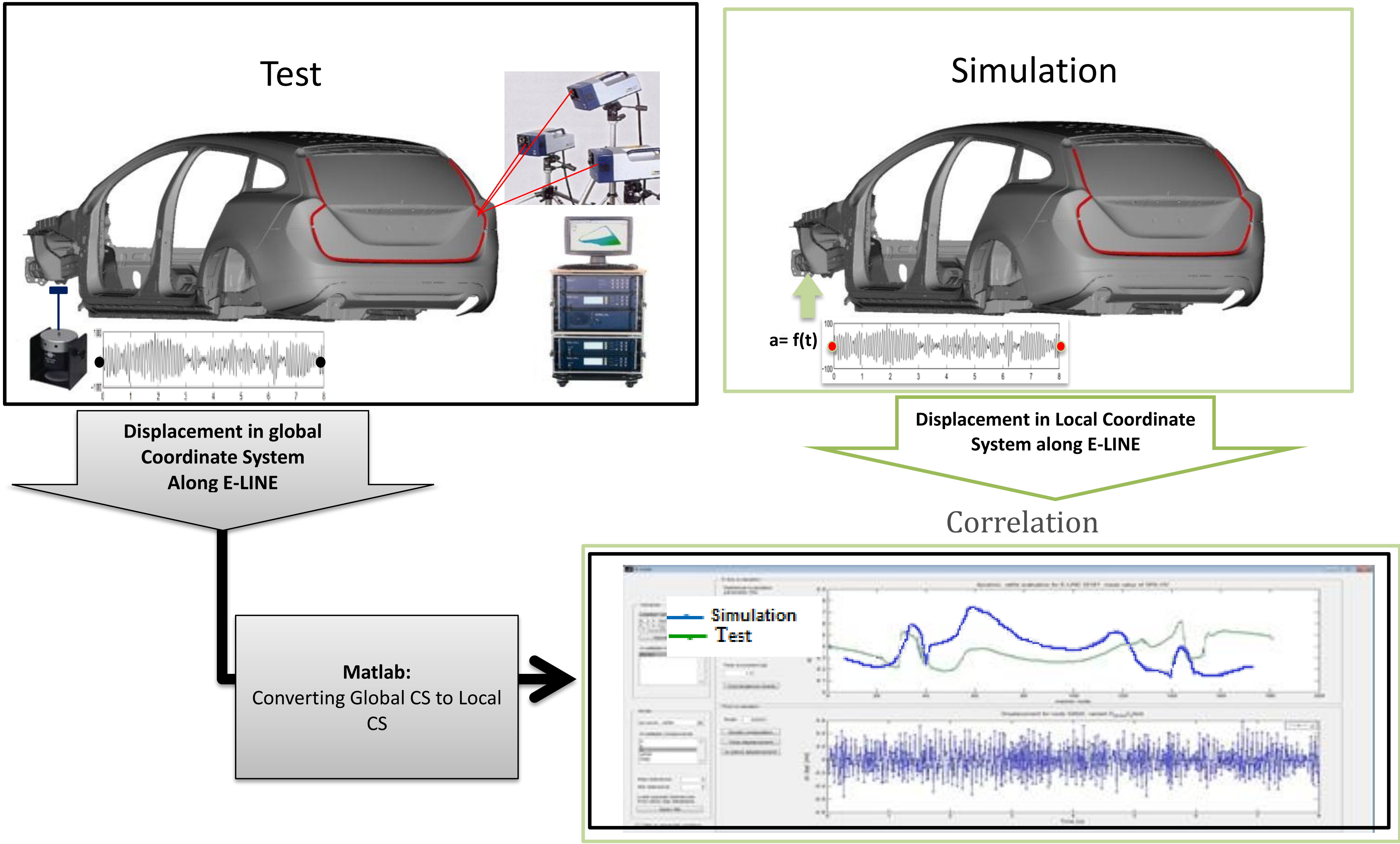
In order to decrease the complexity and also make the project more feasible only the BIW with the assembly parts; a bumper, rear lamps and the tailgate are studied. Therefore, it is important to note that the global damping value (which will be frequently mentioned in the report) refers to the BIW damping and *not* the whole car damping. This master thesis can be divided in to three main parts that are test, simulation and correlation. An outline of the project is shown on Figure 1.

Test: There are three test setups for this project. Two test setups are targeting unknown parameters and the third one is to confirm the results from the first two as well as further investigation. Relative movement of tailgate and exterior parts is of interest therefore points will be defined as pairs along the gap of tailgate. Some more points will be defined to extract mode shapes of the structure. Next, an 8-second random signal will be exciting the car and the responses are recorded by use of 3D laser vibrometers. There will be some post-processing on the test results before evaluating the relative displacement in E-LINE method.

Simulation: After assembling the parts and creating the entire structure including BIW, lamps, bumper and tailgate, an E-LINE is defined as a sealing for the tailgate which includes CBUSH elements with stiffness in three directions. The aim is to measure response of the same points in both test and simulation. Therefore, all the measured points along the tailgate gap are imported to the simulation model. In the simulation the structure will be excited by exactly the same signal as in test. Nastran solution sequence 112 calculates the response for all nodes along the defined E-LINE due to the input signal. At the end, results are evaluated using the E-LINE interface in Matlab.

Correlation: The correlation will be performed by comparing and evaluating the results of relative displacements from test measurements and simulation in time domain. The three test setups can be seen as three independent equations to be used to determine the unknown parameters that are damping for BIW and sealing stiffness in three directions. The first test setup is used to determine the damping value for the BIW. From the results of the second test setup, stiffness in all three directions for the sealing can be obtained. Lastly, the third test setup is performed to provide a confirmation to the achieved results from first and second test setups. Furthermore, the third test setup will investigate the impact of torsional body stiffness on the relative displacement. It is important to note again that the mean for correlation is the relative displacement results in the local coordinate system from test and simulation and the correlation procedure is performed in time domain.

Figure 1: Outline of the project



2 Test object

An initial FE model of V60-BIW including the tailgate was provided from NVH group (Noise Vibration and Harshness) for this project. However, the test model used here does not include all parts of the BIW, since it has been involved in another test prior to this study. Therefore, some steel parts close to the instrument panel (cowl and upper face wall) were removed from the model. This requires modifications in the FE model. To ensure validation for simulation one needs to check the following items.

- Mass of each part and compare them with the reality
- Geometry representation of FE model
- Connections between different parts

2.1 Mass and geometry

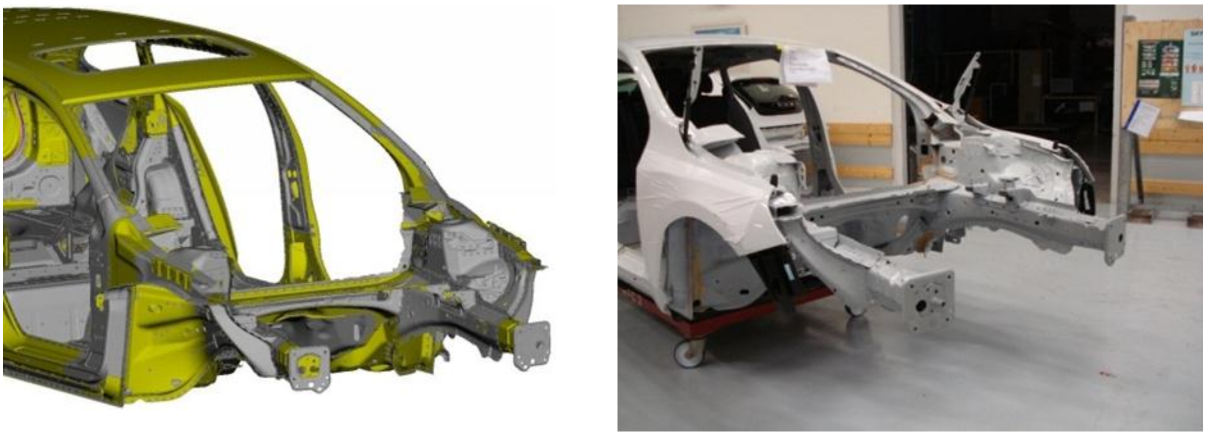
Total mass of the FE model will be adjusted by trimming extra parts from simulation model. In the laboratory total mass of BIW, lamps, bumper and tailgate was measured to get the identical mass in simulation. Results for each part are shown in Table 1.

Table 1: Measured mass for BIW, lamp, bumper and tailgate in test and simulation models

	BIW [kg]	Lamps [kg]	Bumper [kg]	Tailgate [kg]
Test	333.5	3.4	5.4	15.7
Simulation	336	3.4	5.4	15.9

To get an identical mass in simulation, one may tune parameters such as thickness or density of material. This approach was carefully applied to calibrate the mass for the assembly parts (bumper, lamps and tailgate). However, bumper and lamps did not need much calibration adjustment because the initial FE model had previously been used in other simulation. Moreover, for the BIW no thickness or material update has been performed.

The updates were partially performed before the test and partially after the results from the mode extraction were available. The modal correlation allows updating in the material level (elaborated in simulation chapter) however, the main updates before the test include geometry and mass checks.



A *B*

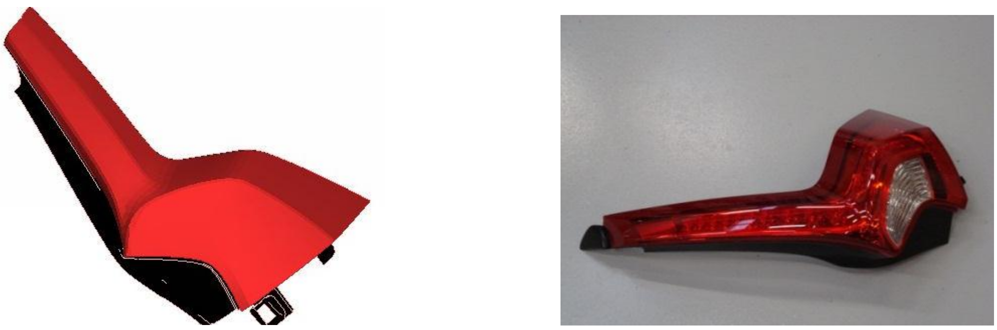
Figure 2: (A) simulation and (B) test model for the BIW.

Figure *2* shows the simulation (A) and test model (B) of the BIW. The cut away part under the windshield opening is indicated.

A B

Figure 3: (A) Simulation model (B) Test model for the bumper

Bumper mass was rather correlated and not a big difference between the simulation model see Figure 3 (A) and the test model (B).



A B

Figure 4: (A) simulation model (B) test model for the lamp

The geometry of the lamp is shown in Figure 4. The final FE model for the lamps contains the housing (inner part) and the lens (outer part). The inner geometry of the lamp however, is not represented in the simulation model. The thickness and the density of both parts are updated in order to validate the simulation model against the test model.

The tailgate mass was calculated based on force required to hold tailgate horizontal and the center of gravity of the tailgate in simulation. This indirect measurement for tailgate mass is due to avoiding errors from tailgate position due to reassembling. In this stage, tailgate mass is adjusted by decreasing the thickness of window from 4.7 to 3.3 mm, see Figure 5. This adjustment is based on the glass thickness measurement results from the laboratory. Later on some more adjustments are implemented according to direct mass measurement of tailgate after the test process.

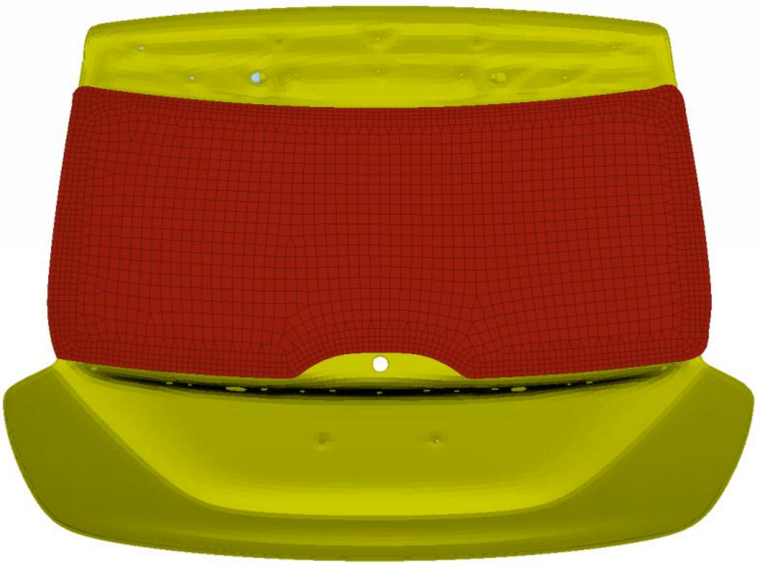


Figure 5: Simulation model – tailgate

2.2 Connection

The connection validation was performed both by carefully considering the physical model with the simulation model and also evaluating the natural frequency of the structures. The three major assembly parts are tailgate, bumper and the two rear lamps. Connections between parts play a significant role for simulation because they have an impact on the mode shapes of the complete assembly. Unconnected parts will result in 6 extra rigid body modes. The Nastran solution sequence 103 which is the normal modes analysis can be used for calculating the mode shapes of any structure and it can be also used to verify the connections. The connections between different parts of the bumper as well as the connections between the bumper and the BIW have been studied.

Figure 6 indicates some of the connections between the brackets and the bumper and the snap connections between the bumper and the BIW. The blue spots on the FE model are RBE2 elements that have infinite stiffness which provide a rigid connection such as a bolt connection.

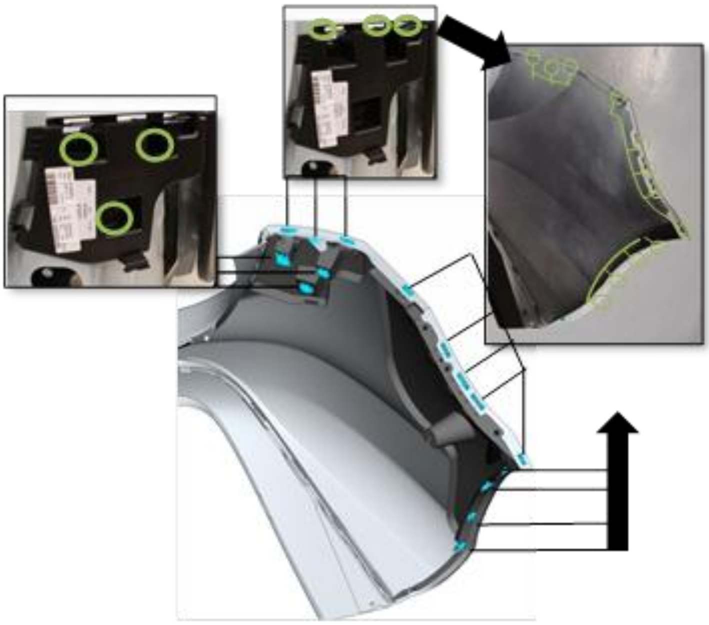
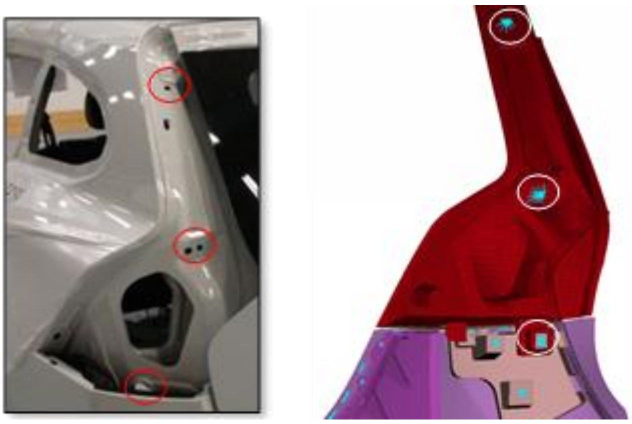


Figure 6: Connections between the different parts of the bumper and between the bumper and the BIW

The lamps connections to the BIW are also studied. For each lamp two connections to the BIW and one to the bumper were provided, see Figure 7.



A *B*

Figure 7: Connections between assembly parts and BIW, (A) test and (B) simulation

3 Test

The test was performed at LeanNova laboratory in Trollhättan.

Figure *8* shows the test configuration where the 3D laser vibrometer is placed behind the car to measure response of points along the visual gap between the tailgate and the surrounding components. The car is located on four air-pillows to be modeled as a free-free, see

Figure *9* and is excited in the front left end in vertical direction. The car is painted in white by a developer flaw detection spray to increase the reflectivity. Absolute displacements of defined points along visible gap are measured for three different test setups.

The three test setups are,

1) Tailgate without rubber sealing.
2) Tailgate with rubber sealing.
3) Tailgate with rubber sealing and an added beam system to increase the torsional stiffness, see Figure 23.

The first and second test setups target the global damping value and the sealing stiffness respectively. The third test setup is to evaluate the impact of the torsional stiffness of the BIW on relative displacement and also to confirm the previous results.

Figure 8: Test configuration

Figure 9: Free-free boundary condition approximation with air-pillows

3.1 Test preparation

Three main parameters of the force are studied here to find the best test configuration,

1) Force location
2) Force direction (angle)
3) Force level

All of these parameters have impact on the relative displacement. The aim was to find an optimal configuration in terms of dynamic response of the complete structure.

3.1.1 Force location

Four different candidate force locations are chosen, two in front (right and left of the BIW), one in the middle and one in the back, see Figure 10.

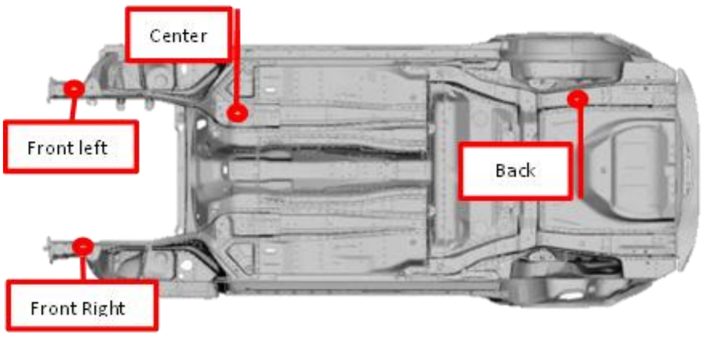
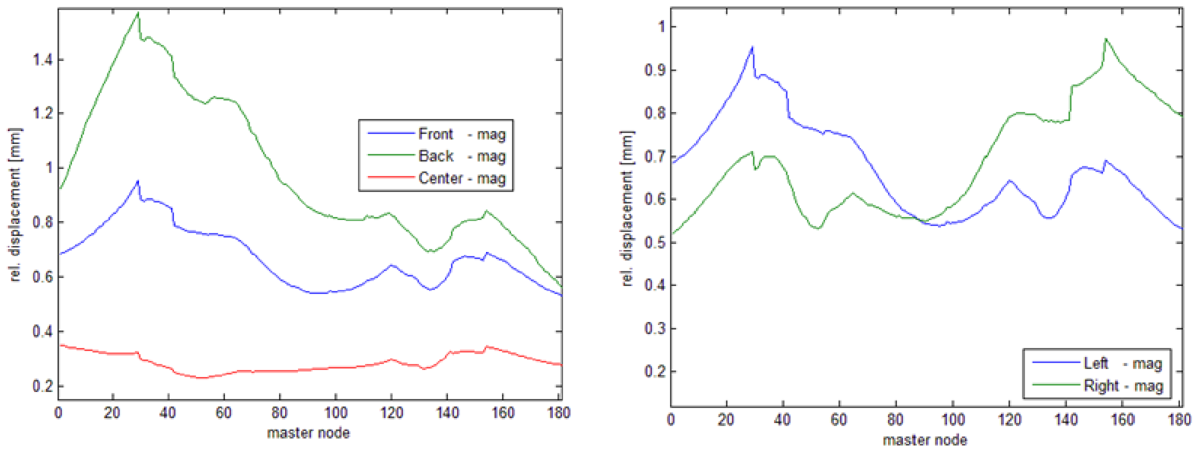


Figure 10: Force locations

To verify the best location, FE simulation were performed. Two kinds of results were studied,

1) Relative displacement along the tailgate gap line
2) Maximum total displacement in the BIW in all nodes

For the relative displacement both amplitudes and curve shapes are evaluated and compared to each other. The amplitude should be maximized to have sufficient relative displacement with limitation of gap size to avoid contact. The curve shape should be as equal as possible, which means that the difference between the mean value and the max/min value is minimized. According to these criteria the front excitation was preferable, see Figure 11 (A).

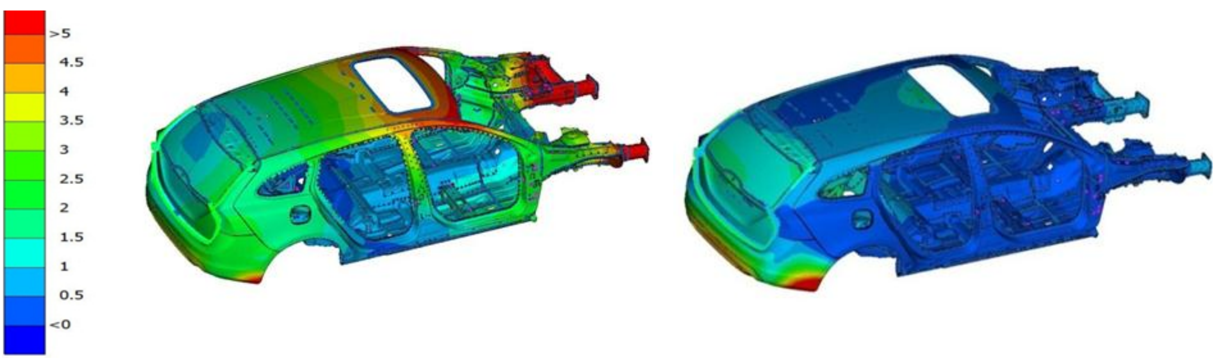


A *B*

Figure 11: Force locations. (A) Front excitation is the best choice regarding to amplitude and standard deviation. (B) Exciting left side or right side has just the symmetry difference.

Figure 11 (B) indicates that the left and right location has no significant difference due to the car symmetry thus, left position is chosen.

Figure 12 shows the max value in each point during the complete response which means the values shown in each point can occur at different points in time. The maximum total displacement of the body due to front and back excitation is similar. After a comparison of these two excitations (front/back), the front location is chosen because it excites the BIW more evenly and involves more eigenmodes than the excitation in the back. According to the results from both comparisons the front-left excitation location is preferred.

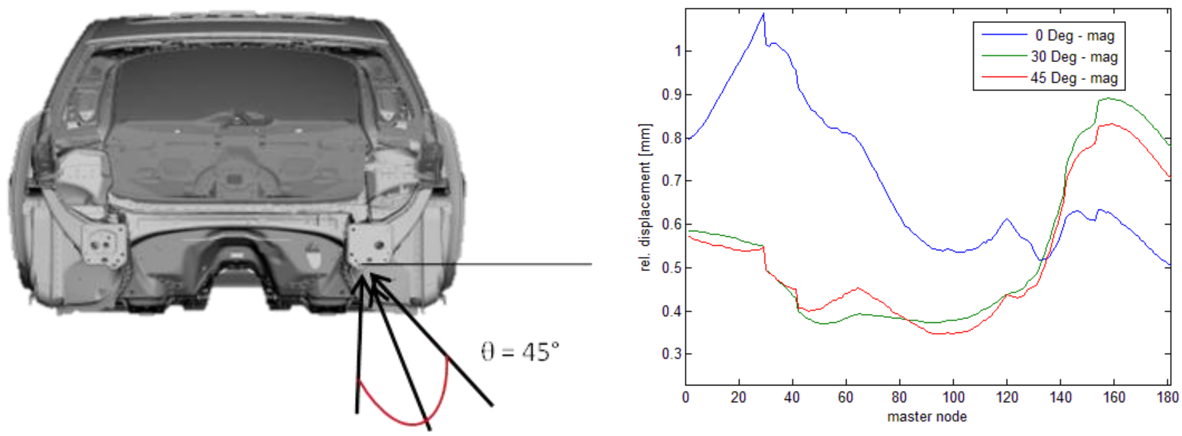


A *B*

Figure 12: Maximum value (mm) of total displacements along the BIW-exciting the (A) front, (B) Back

3.1.2 Force direction

Another parameter to study is the direction or angle of applied force. For this, three excitations with 0, 30 and 45 degrees are applied on front-left of the model as shown in Figure 13. The relative displacements due to the zero degree (vertical arrow (A) blue curve (B)) excitation reveal the highest amplitude in comparison with the response due to excitation in the two other directions.

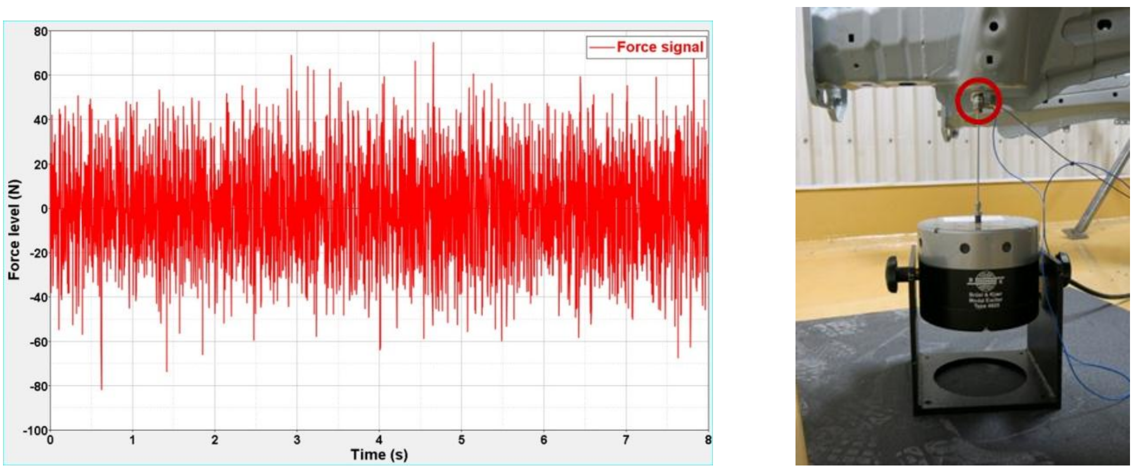


A B

Figure 13: (A) Three force directions (0,30,45 degree) applied on the front-left, (B) Relative displacement curves along E-LINE in local z direction-response due to the three different force angel

3.1.3 Force level

An 8-second pseudorandom signal was requested from Polytec for this measurement. A plot from the signal is presented in Figure 14 (A). Figure 14 (B) illustrates the shaker and its position which is on the front-left side of the BIW. The same signal is applied to the simulation model. The duration of signal is chosen 8-second make sure that the complete frequency content is applied on the structure.



A B

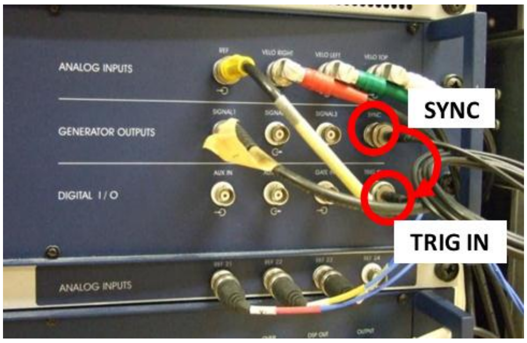
Figure 14: (A) An 8-second force signal, (B) Modal exciter from Brüel & Kjær, (specification, Rated Force [random (RMS)] 70 – Max 100N)

3.2 Test procedure

Laser vibrometry is a rather new type of measurement technique. Some of the advantages of using this method are,

1) High precision measurements in comparison to testing with accelerometers.
2) No-mass loading on structure as well as no contact tools procedure.
3) Wide range of frequency for the measurements up to 20 MHz.

3D laser Vibrometers camera is a test instrument that allows measurements in all three directions simultaneously in one measurement point at a time. To apply the same excitation signal for each measurement point, a trigger function is used. This feature can be implemented to the measurement system using an external trigger function, as shown in Figure 15 (A). The trigger function results in an identical response for single point measurement. In Figure 15 (B) the same point has been measured two times to illustrate the repeatability.



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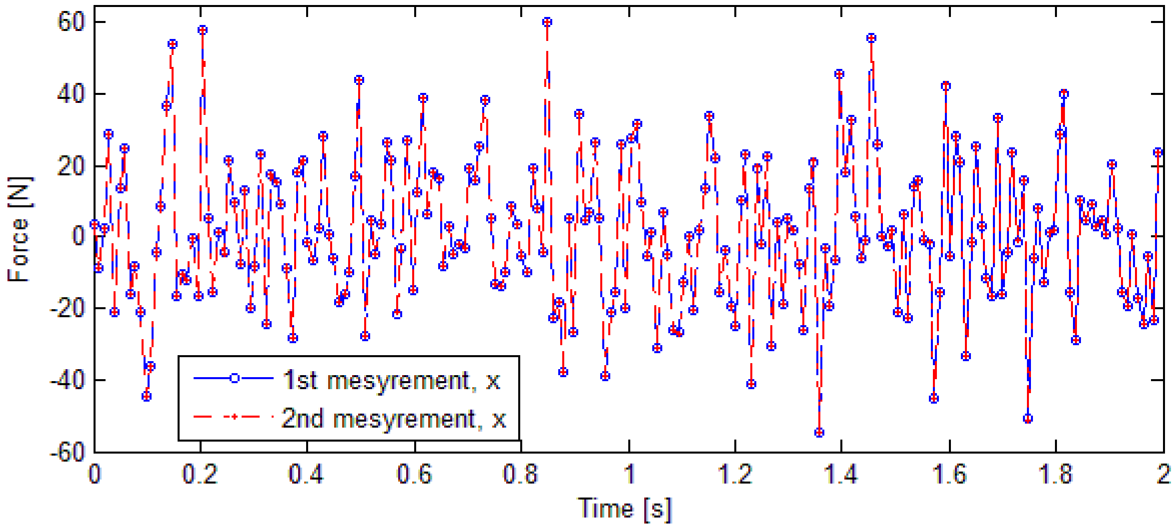


B

Figure 15: (A) Trigger function applied in the acquisition system (using two cables), (B) Verification of trigger function operation.

Figure 16 illustrates all the measurement points that have been defined for the three test setups. The lasers will measure one point after the other. Hence, the trigger function will assure that each point will be excited for 8 seconds with the same signal.

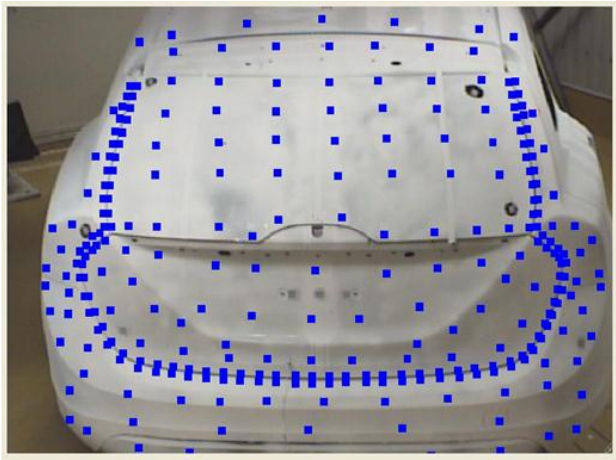


Figure 16: All defined measurements point

Three test setups have been used during the measurement and for each test setup two measurements were performed. The measurements are,

1- FFT run
2- Time domain run

First, the FFT measurement was performed to extract the modes and the corresponding modal damping. The duration of the FFT measurements was 4 seconds. The velocity was requested as output for the FFT run. Later, the time domain measurement was performed with duration of 8 seconds for each measurement point. For the time domain measurement, the displacement was requested from the Polytec test system which means that the system integrates the velocity signal from the laser Doppler.

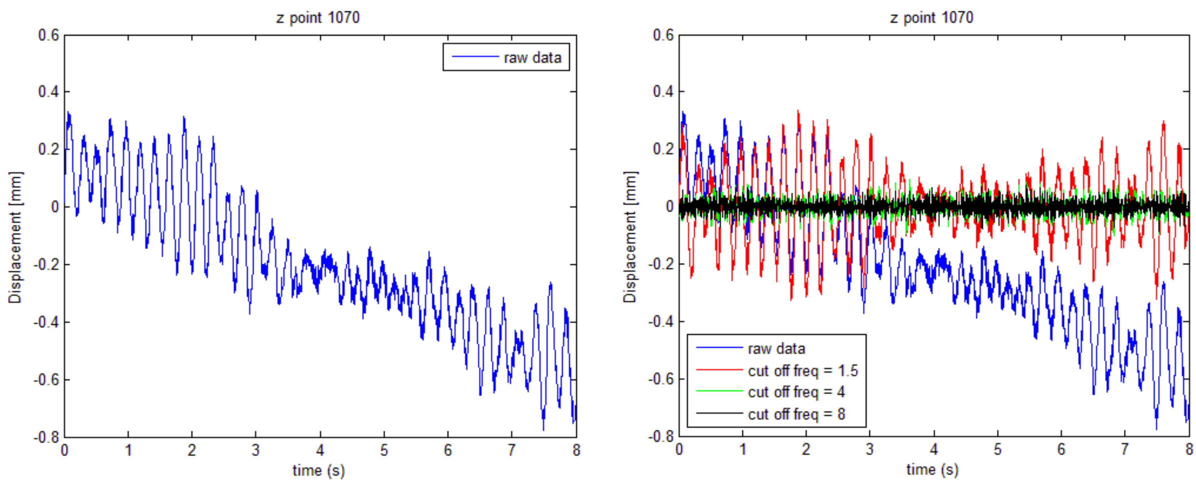
In the FFT run, all information will be recorded in frequency domain (FRFs) which can be used to extract the mode shapes and modal damping values for each mode. Modal damping refers to the damping for each specific mode in frequency domain. Measurement points (see Figure 16) are defined according to mode shape and natural frequency extraction of tailgate and bumper. These results can be achieved by use of Polymax method in the LMS software [5]. Results from FFT can be already implemented in Polytec measurement software to achieve the Operational Deflection Shape (ODS) feature in Polytec, see Figure 17. In the figure the tailgate mode and ODS at 21 Hz is visualized. The detailed information about ODS and its difference to eigen-mode can be read in [6].

The results from both ODS-Polytec measurement and the LMS-Polymax are showing almost exactly the same results. One advantage of using the LMS- Polymax is that the modal damping can also be extracted [5]. For each mode a specific damping value is calculated.

A B C

Figure 17: (A) Undeformed tailgate, (B) tailgate torsional ODS in Polytec, (C) tailgate torsional mode in Polymax-LMS

In time domain analysis for comparing the results from test with the simulation results, some post-processing on data is required. First time domain data from measurements are filtered to exclude rigid body response from data. The contribution from rigid body modes are eliminated by using a high-pass filter with a cut-on frequency set to 10 Hz. Also, the filtered data are transferred from the global coordinate system of measurement to the local coordinate system of each point (defined according to E-LINE method).



A B

Figure 18: (A) Raw data, (B) filtering different frequencies from raw data

Figure 18 (A) shows the raw data from response of a single point during the 8 seconds test measurement. The drift in Figure 18 (A) is due to the mathematical integration of velocity (the impact of DC bias is one of the reasons) that occurs due to the system internal measurement procedure [7]. After filtering the data and coordinate system transformation, the results became comparable with the simulation results. . Figure 18 (B) illustrates the impact of filtering at different frequencies from the raw data. Frequency 4 and 8 Hz are the rigid body modes frequencies which have a significant impact on the amplitude of raw data.

4 Simulation

4.1 Introduction

In the simulation, the main aim is to bring up the model as close to reality model as possible. For the simulations, ANSA is used as a pre-processor for Nastran [8] which is used for finite element analyses. Two Nastran solution types are used in this project, Sol 103 (normal mode analysis) and Sol 112 (modal transient response) [9]. The results from the analyses are visualized and evaluated in the post-processing software μETA.

4.1.1 E-LINE

The E-LINE method was first presented in 2012 in an SAE paper [3]. The focus of E-LINE is on computing and evaluating the relative displacement between two parts [3]. The relative displacement measurement is performed in local coordinate system to have all values in the rattle direction or squeak plane [3]. The evaluation of relative displacement in E-LINE is performed in time domain to include the phase formation.

Figure 19 shows an overview of the simulation process. The process can be divided into three main steps which result in calculation of relative displacement. The first step is to create an E-LINE in ANSA. The second step is to run solution 112 (modal transient analysis) in order to calculate the response (displacement) of the structure, output on a .pch file. The last step is to compute the relative displacement by applying the E-LINE method via an interface on Matlab program to evaluate the relative displacement along the defined line.

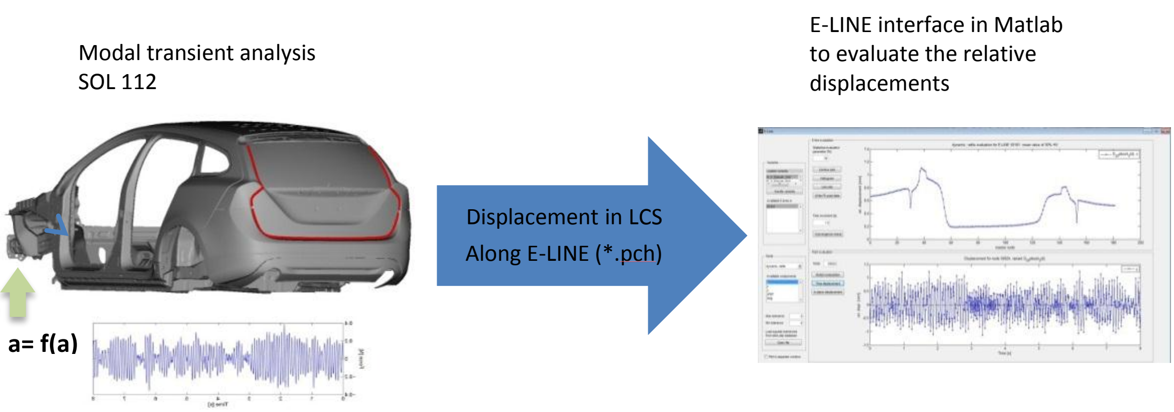


Figure 19: Overview of the simulation procedure

4.2 Model updating

The three main items that have impact on the natural frequency of any structure are stiffness, mass and boundary conditions [10]. Likewise the dynamical behavior of any structure is connected to the mentioned parameters. This explains the importance of fidelity of the FE model.

To ensure the validity for simulation one needs to check and correct the stiffness and material properties of components.

Visual modal correlation is an approach to validate and update the model. After ensuring the fidelity of FE model, a first loop of modal validation can be run. In simulation, mode shapes and natural frequencies of the complete assembly are computed from Nastran Sol 103. From the FFT test results, mode shapes and natural frequencies can be extracted by use of Polymax. The visual modal correlation between the extracted modes from Polymax and simulation results from Sol 103 can be used to ensure that the updating is trustworthy. In order to have a validated model all parameters such as mass, material properties, geometry need to be updated to be in accordance with the physical test object.

4.2.1 Bumper

The frequency of the first bumper mode was different from the simulation model. Hence, the material properties of the plastic parts were adjusted by modifying the Young's modulus from 2000 MPa to 3200 MPa which is still in the appropriate range [11], see Table 2.

Table 2: Influence of E modulus on mode shape of bumper

	Initial-FE Model	Updated-FE Model	Test
E [MPa]	2000	3200	-
Frequency [Hz]	14.84	18.4	18.5

4.2.2 Tailgate

The behavior and natural frequency of the tailgate depends on different parameters. Therefore, updating becomes more complicated than the bumper and lamps assembly parts. The bump stops were removed from both test and simulation to decrease the uncertainty.

The model for the tailgate glass has been modified in different steps. The density was decreased from 2500 Kg/m^3 to 2200 Kg/m^3 which was the only uncertain parameter (other parts are steel part with a defined density) and finally the Young's modulus was modified from 70 to 73 GPa.

Another item that has influence on the behavior is the bonding of the glass. The bonding is modeled by solid elements. In order to validate the mode shapes of tailgate between the test and simulation, the shear modulus of the bonding was raised from 2.7 to 6.7 MPa. The influence of modifying the mentioned parameters on mode shape of tailgate is given in Table 3.

Table 3: Influence of modifying bonding for the glass as well as glass density on mode shape of tailgate

Mode	Initial Frequency [Hz] FE-Model	Updated Frequency [Hz] FE-Model	Frequency [Hz] Test-Model
Tailgate Torsion	29.3	19.5	21.0

4.3 E-LINE creation

In order to compare the relative displacement results from test and simulation an E-LINE should be created which has master and slave points. The same points should be used both in test and simulation. For this reason all the test points geometry were input in ANSA. In the next step all the transferred points were projected to the surface of material because points of the test were not exactly on the surface. To select the closest grid on the simulation model, the mesh refinement was performed and the closest grid was chosen as the corresponding simulation points. These procedures ensure that all the E-LINE points in the simulation are the same or very close to the test measurement. After defining all the points (masters and slaves) on the model a CBUSH element with a specific local coordinate system was created between each master and its corresponding slave. The local coordinate has the x axis along the gap of the tailgate, y axis is flush and z axis is normal to the gap, see Figure 20. In total 70 CBUSH elements were created which means that the resolution of the E-LINE plot on x axis will 70 points, see Figure 26.

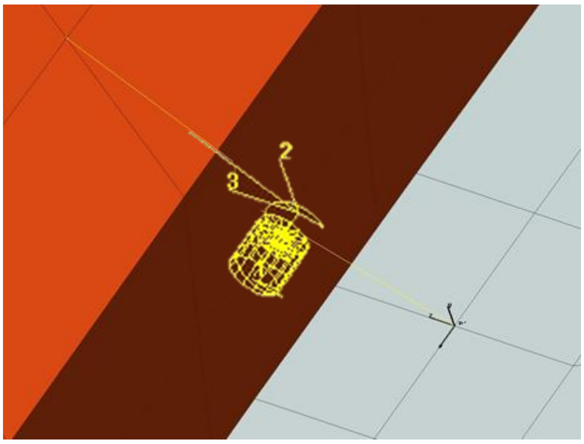


Figure 20: CBUSH element coordinate system direction, which is local coordinate system for relative displacement evaluation

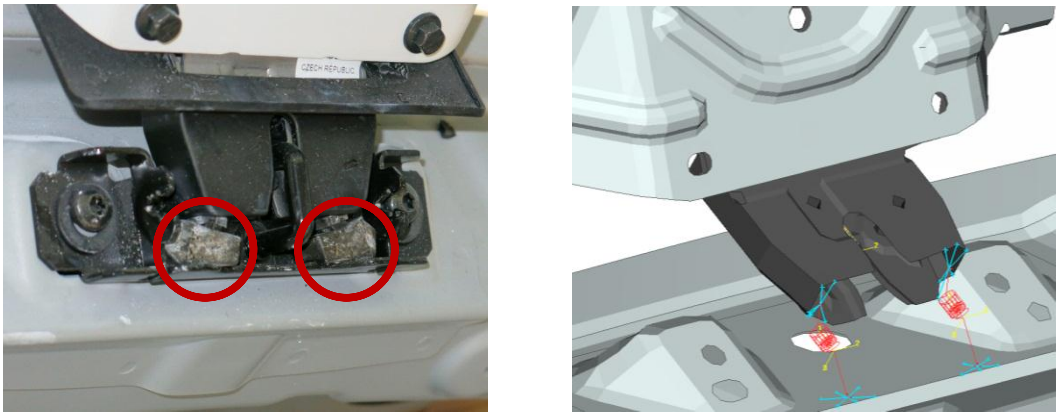
The E-LINE model is then created and ready to use for all three updated simulation models. Later, Sol 112 for different test setup will be run and the relative displacement can be determined via E-LINE interface in Matlab. Finally, the unknown parameters (damping and sealing stiffness) will be calculated.

4.4 Test setups

Three test setups are defined. The first two test setups target the unknown variable (global damping, stiffness). The third test setup (adding a beam system to the BIW) is to confirm the results from the first two test setups. Moreover, the third test setup investigates the impact of torsional stiffness on the relative displacement.

4.4.1 Test setup 1

In the first test setup, the rubber sealing in the tailgate's gap is removed to focus on the effect of damping and calculate the global damping of the BIW. When the sealing was removed, the latch was not in the defined position (latch was resting against the BIW). This was prevented by adding rubber cubes between the stop and the latch, see Figure 21 (A). In the simulation model these rubber cubes were modeled by a CBUSH element with the stiffness of 100 N/m only in the opening direction, see Figure 21 (B).



A B

Figure 21: (A) Tailgate latch in first test setup (B) in the simulation model

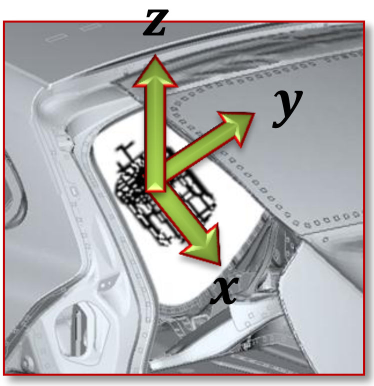
Another important update was concerning the latch modeling. Initially, in the FE model the latch is modeled by CBUSH element that includes stiffness in closing and lateral direction equal to 2500 N/m and 40 N/m in the radial direction. According to the results from extracted mode (specifically the tailgate sliding mode), it could be obviously concluded that the radial stiffness also needs to be increased. Hence, the radial stiffness for the latch model was also given 2500 N/m in order to validate the second tailgate mode (sliding mode). Results from Table 4 show a good correlation between test and simulation for the first test setup. In another words, the first test setup is ready for time domain correlation of relative displacement.

Table 4: Natural frequencies comparison between test/simulation for test setup 1

Mode	Initial Frequency [Hz]	Updated Frequency [Hz]	Test Frequency [Hz]
Bumper	14.8	18.4	18.5
Tailgate Torsion	29.3	19.5	21
BIW Torsion	25	25.8	27
Tailgate Sliding	25	36.7	35

4.4.2 Test setup 2

The rubber sealing was assembled in the second test setup. Likewise, in the simulation model a sealing was created using the E-LINE method. This allows the stiffness definition in all three directions, see Figure 22 (A). Sealing is generated by CBUSH elements that are placed every 20 mm along the tailgate, see Figure 22 (B). The unit for the stiffness of sealing is normally expressed in N/mm, however according to this sealing model stiffness is given to each CBUSH element. This means that the unit for sealing stiffness is in N/mm/20mm, which will be used later in correlation.



A *B*

Figure 22: (A) indicates the sealing simulated by CBUSH elements, (B) specified direction for sealing stiffness.

For the second test setup, some modes are exactly the same as for the first test setup such as the BIW torsion and bumper modes. Modes related to the tailgate are totally different. The reason is that the sealing adds to the stiffness and increases the tailgate's natural frequencies especially those related to torsional mode. Table 5 shows the comparison between some of the mode shapes of test and simulation for the second setup.

Table 5: Natural frequencies comparison between test/simulation for test setup 2

Mode	Updated Frequency [Hz]	Test Frequency [Hz]
1st Tailgate Torsion	36.7	35
Tailgate Sliding	37.2	36
2nd Tailgate torsion	40.8	43

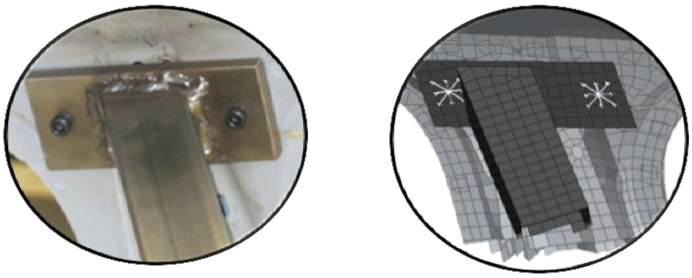
4.4.3 Test setup 3

The last test setup has a beam system with a mass of 9.3 kg added to the BIW in order to increase the torsional stiffness. This has been modelled in the simulation, see Figure 23. The mass of the beams in the simulation model is 10.3 kg. The torsional mode increased due to the added beam system from 26.3 Hz to 32.5 Hz in the simulation model. Test showed an increased in the torsional mode from 27.0 Hz to 33.0 Hz.

A B

Figure 23: Adding beam to increase torsional stiffness in third setup (A) indicates simulation (B) test.

Likewise, for the third test setup the connections were studied and modeled, see Figure 24. The bolt connections were modelled as RBE2 elements with infinite stiffness.



A B

Figure 24: the top left connection of the beam to the BIW (A) test (B) simulation

For the third test setup as in the second test setup, some modes are similar to the second setup including the bumper and tailgate torsion (similar to the second one). Table 6 however compares the other modes such as the BIW torsion mode which increases by 6 Hz compared to what it was in the second test setup.

Table 6: Natural frequencies comparison between test/simulation for test setup 3

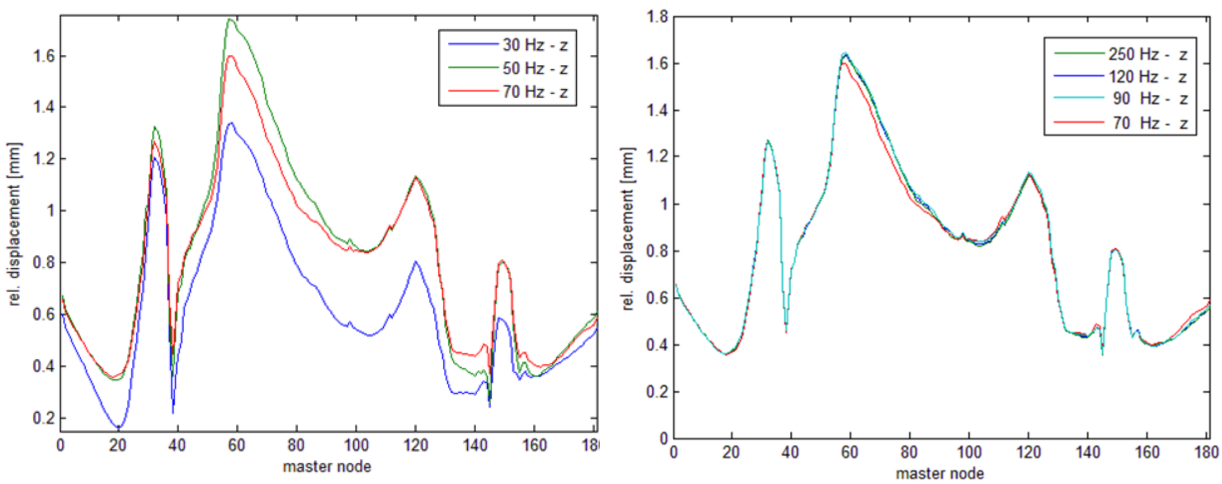
Mode	Updated Frequency [Hz]	Test Frequency [Hz]
BIW Torsion	32.4	33.5
Tailgate Sliding	43.1	39
2^{nd} BIW Torsion	32.4	47.5

4.5 Parameter's study

The study of parameters partially had a focus on the excitation configuration which was described in Chapter 2. The other focus of the parameter study is the number of modes that will contribute to the response.

4.5.1 Number of modes

Nastran Solution sequence 112 calculates the transient response of a structure by use of modal superposition. Hence, the number of modes will influence the response of the structure. This influence has been studied by including frequencies from 30 Hz up to 250 Hz. Figure 25 (A) shows that the relative displacement plot goes up by increasing the frequencies from 30 to 70 Hz. Figure 25 (B) however, indicates that the relative displacement curves for 120 Hz and 250 Hz are almost the same. In another words, the results of the relative displacements are converging from 120 Hz. Therefore, 120 Hz will be used during the simulation procedure for the calculation of relative displacement.



A B

Figure 25: (A) Relative displacement in local z due to including frequencies from $\mathbf{30}$ *to* $\mathbf{70}$ $\boldsymbol{Hz}$ *(B) Relative displacement results due to including frequencies from* $\mathbf{70}$ *to*$\mathbf{250}$ $\boldsymbol{Hz}$

5 Correlation

The last part of this study is to correlate the results from test and simulation. It is important to note that the benchmark for this correlation is the relative displacements plot for time domain data, see Figure 26. Relative displacement correlation necessitates using time domain response. The relative displacement is plotted based on a statistical approach in E-LINE method where the mean value of 30% of highest value is plotted instead of the maximum values. The statistical approach in the time domain is a more robust evaluation in comparison with the maximum vales [3].

The global damping value for the BIW and sealing stiffness in three directions are the unknown parameters which are verified due to a correlation. The correlation assessment principles are first visual correlation and mathematical approach which is based on the sum of square due to error (SSE) between test and simulation. Figure 26 shows how the SSE approach gives us the most accurate results. Figure 26 also illustrates where the relative displacement plot starts along the tailgate gap and where it ends. The plot between the vertical lines shows the relative displacement along the bumper.

Global damping and sealing stiffness are obtained from the first and second test setups and the third test setup correlation is a substantiation of the obtained values from the two first setups.

5.1 Test setup1

In the first test setup the only variable is the global damping. Since the rest of the model has been validated through its eigenfrequencies a global damping value can be set in the Nastran input file. Different values have been given and the results of relative displacements have been evaluated by use of the E-LINE method. A damping value of 0.5% shows the most accurate results, see Figure 26.

Figure 26 also indicates the magnitude of the relative displacement along the closure gap. The results show the precision of the simulation model. The damping value of 0.5% can be used as a global damping value for the BIW in any similar analysis. This damping value is an accurate value that has been achieved through the correlation in time domain. This means that the superposition of modal damping value for each specific mode can be substituted by the presented value.

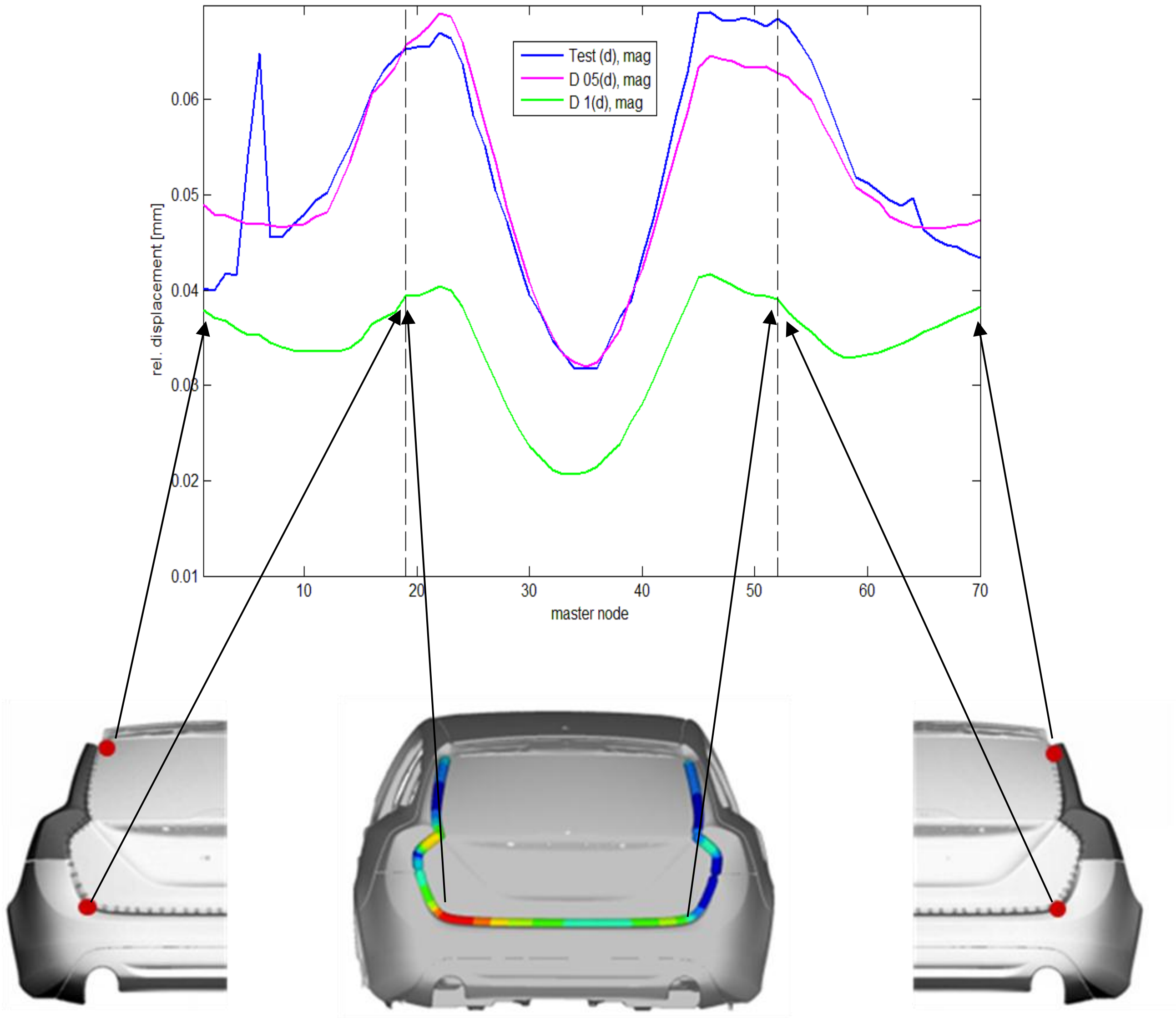


Figure 26: Impact of curve fitting approaches on magnitude of relative displacement from test setup 1 and simulation as well as the relative displacement plot illustration along the tailgate gap

Figure 27 shows the results for the first test setup (test versus simulation) in the local z direction (gap direction). A comparison between the test (green curve) and the simulation shows a good correlation between the results which substantiates the global damping value of 0.5%

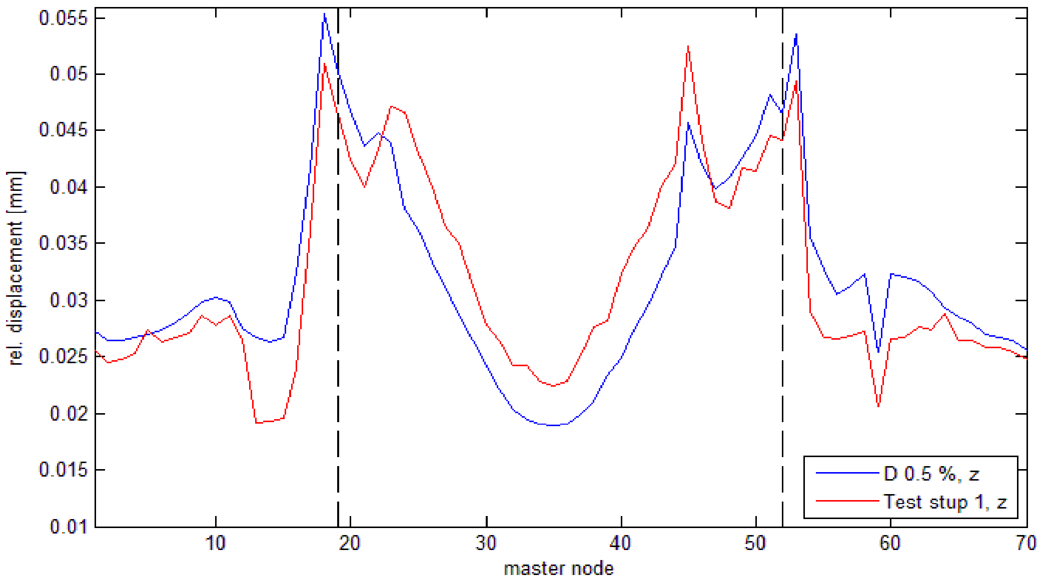


Figure 27: Relative displacement in local z direction from test setup 1 and simulation

5.2 Complexity regarding the stiffness

To start with, the impact of single stiffness value on the relative displacement was investigated.

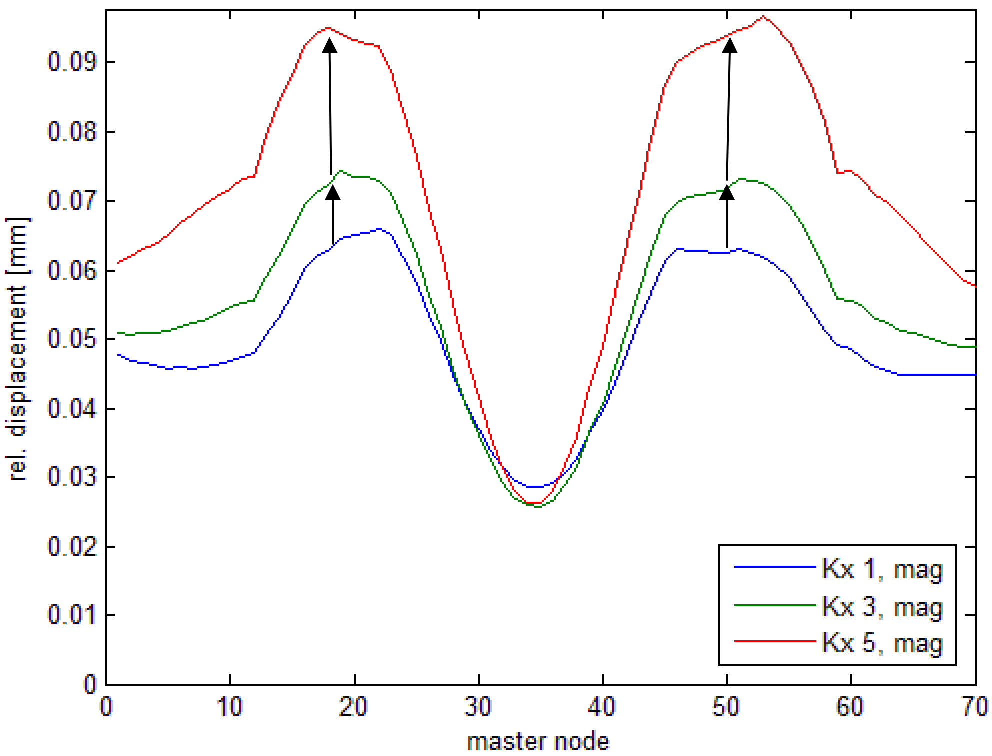


Figure 28: Magnitude of relative displacement due to the stiffness in local x direction

Figure 28 shows the impact of increasing k_x on the relative displacement. The results show that increasing the k_x causes the increase in the relative displacement which is not expected.

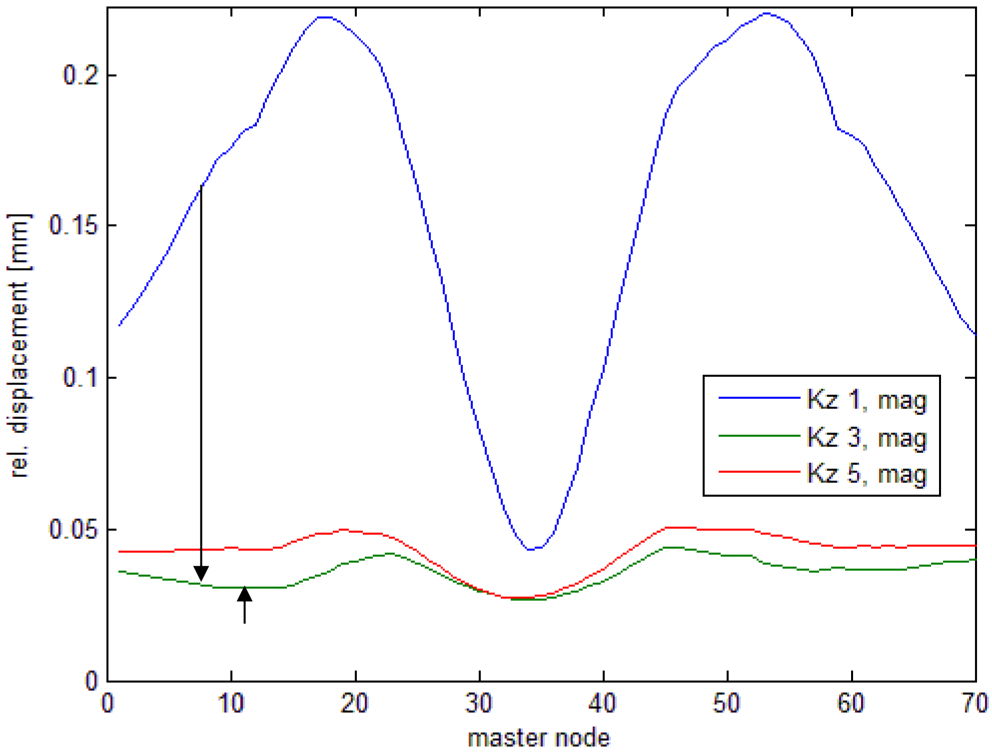


Figure 29: Magnitude of relative displacement due to the stiffness in local z direction

Stiffness only in z direction was also evaluated, see Figure 29. The figure shows that by increasing the stiffness in z direction, first the plot of relative displacement goes down and then it goes up which is even more unexpected. In another words, relative displacement plot due to the change and varying the stiffness is unpredictable. The reason is due to the fact that the absolute displacement (response) is determined from the result of modal transient analysis and varying the stiffness will vary the eigenfrequencies and phase, thus interpretation of the behavior is not easy.

In order to systematically investigate the impact of sealing stiffness on the second and third test setup, the results from different stiffness configuration were analyzed. A table of simulation analysis constructed for studying the impact of single stiffness in all three directions as well as the combination of two direction or all three on the relative displacement plot.

Evaluating the results from the table gave an idea of the sealing stiffness cube (space). Figure 30 illustrates the sealing stiffness cube where each axis shows stiffness in a single direction.

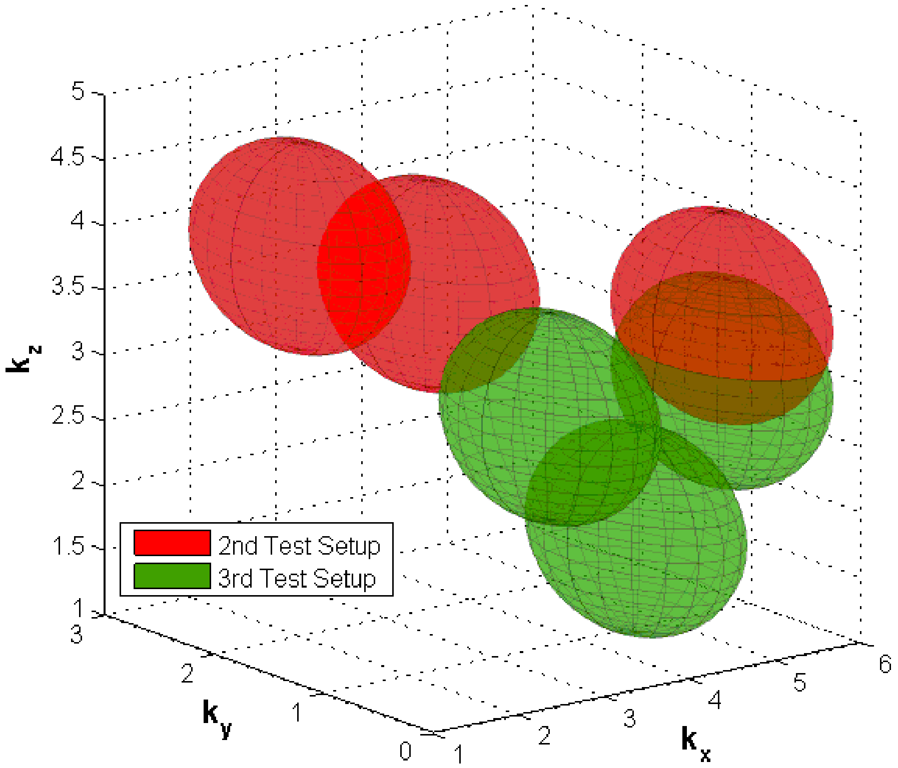


Figure 30: Sealing stiffness cube which indicates criteria used for stiffness in each direction

In Figure 30, the red spheres indicate sealing stiffness configurations that give good results for the second and the green spheres show the most accurate results for the third test setup. By looking at the common space (common space between green and red spheres) the best sealing configuration that gives a correlated result for both test setups can be achieved. Sealing stiffness cube helps to constrain and limit the space to look for the most appropriate sealing stiffness configuration.

Another conclusion based on the evaluation the relative displacement due to varying the stiffness is that, stiffness in y direction has no significant impact on the relative displacement. Therefore, k_y was excluded from the sealing stiffness configuration for both second and third test setup.

5.3 Test setup 2

In the second test setup, the rubber sealing was installed and all measurements were made with the sealing mounted. The challenge to get correlated results for the second test setup was greater than for the first test setup as it was mentioned. The sealing stiffness in all there directions needed to be taken into account. The lateral stiffness k_y was found to have a minor impact on the relative displacement and therefore it was excluded. Figure 31 and Figure 32 show the magnitude and z component of relative displacement for test setup 2. The most accurate results for the second test setup were achieved by $k_x = 5$ and $k_z = 3$ N/mm/20 mm.

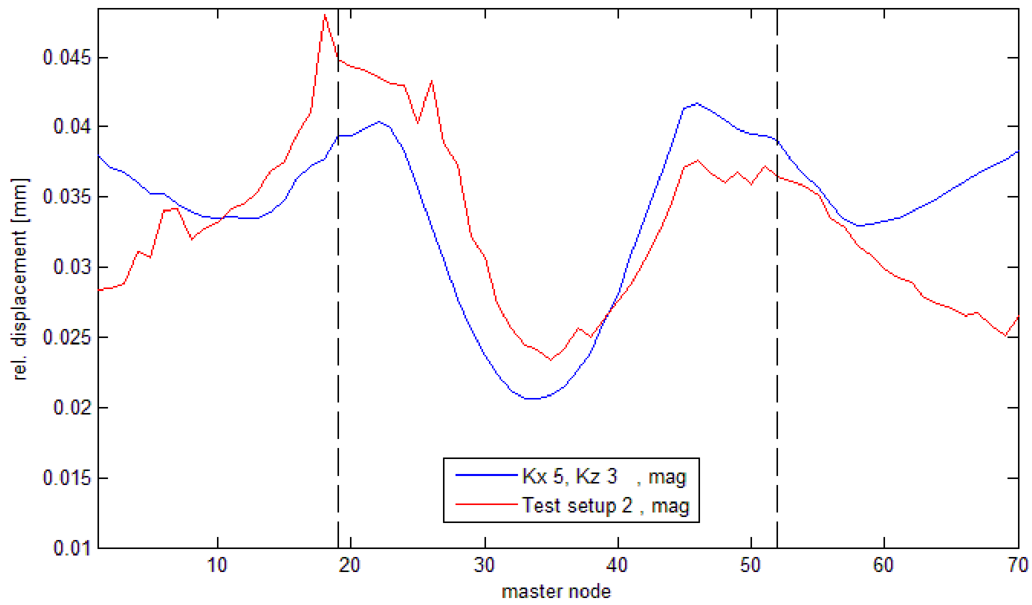


Figure 31: Magnitude of relative displacement from test setup 2 and simulation

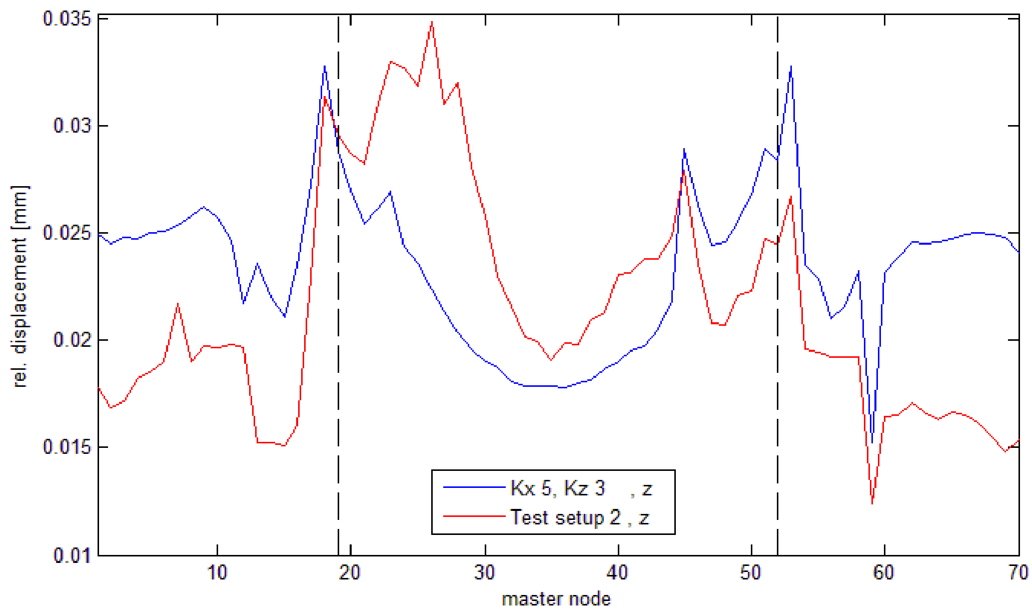


Figure 32: Relative displacement in local z direction from test setup 2 and simulation

5.4 Test setup 3

The last test setup was performed not only to investigate the impact of torsional stiffness on the relative displacement but also to validate the result from the first and second test setups (damping and sealing stiffness). Also for the third test setup the lateral stiffness ky was set to zero. Relative displacements from test setup 3 (including the beam system) and from simulation are shown in the Figure 33 (magnitude) and in Figure 34 (z component). The best results were achieved by following configuration $k_x = 5$ and $k_z = 3.5$ N/mm/20 mm.

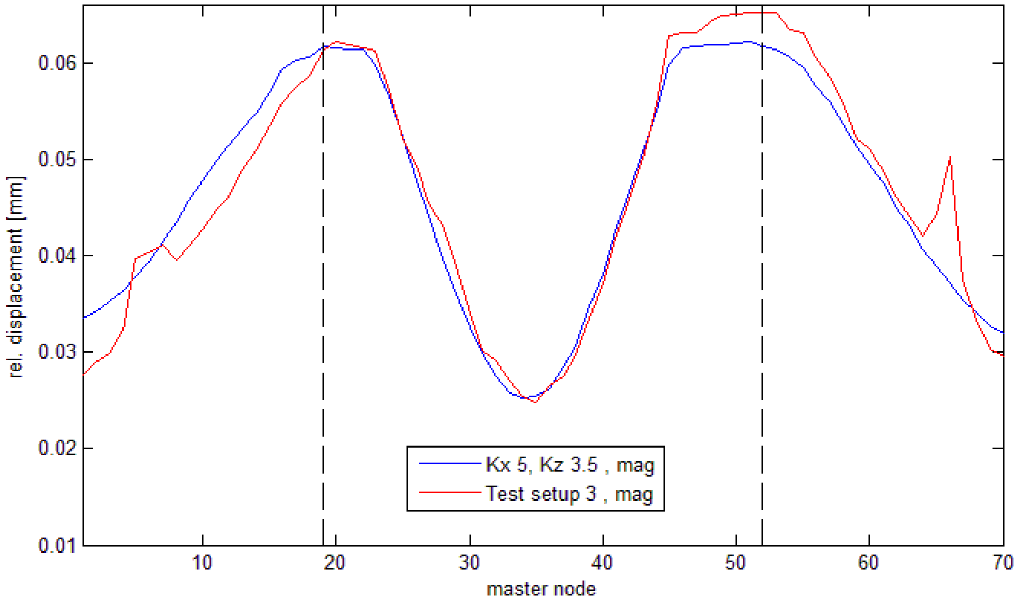


Figure 33: Magnitude of relative displacement from test setup 3 and simulation

Results from the last test setup show that the relative displacement does not necessarily decreases when increasing the torsional stiffness. In some areas along the gap the relative displacement increases due to the higher torsional stiffness. This is also seen in the simulation model which indicates the fidelity of the model and precision of the test measurements.

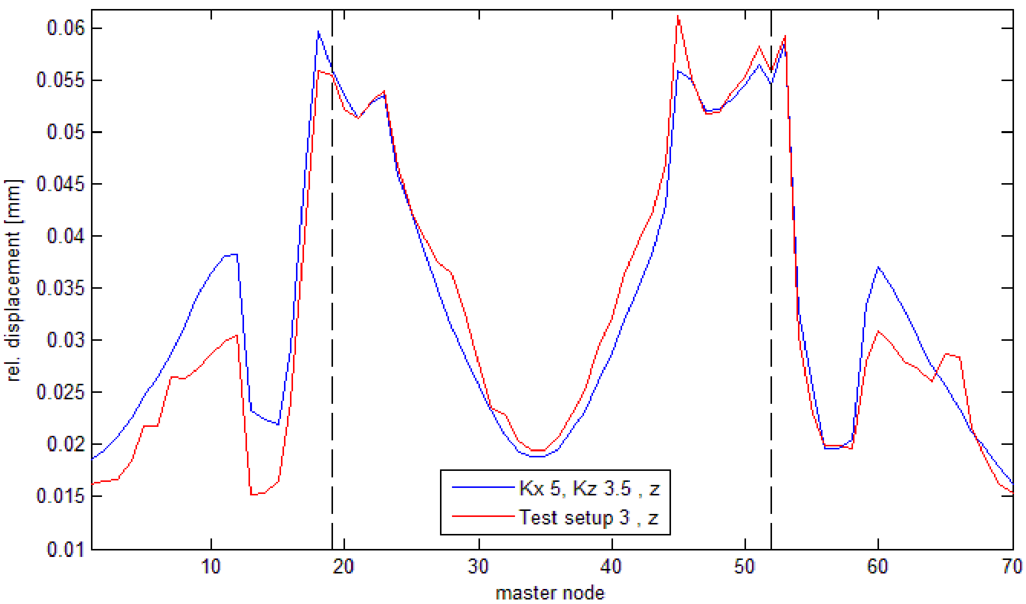


Figure 34: Relative displacement in local z direction from test setup 3 and simulation

6 Conclusion and future work

A correlation between test and simulation has been performed to update the simulation model. The main aim was to improve the value of the global damping for the structure as well as the rubber sealing stiffness considering all three directions. The benchmark for the correlation was the relative displacement results. It is shown that the updated damping and sealing stiffness values are clearly increasing the capability of the relative displacement simulation during the virtual development phase.

Conclusion

- The damping value for the BIW is within the range of $0.4\% - 0.6\%$
- For the stiffness value there can be different configurations giving accurate results
- All three stiffness direction influence the relative displacement
- The best configuration is achieved by setting $k_x > k_z > k_y$
- Stiffening up the BIW will not necessarily decrease the relative displacement

Future work

Suggestions for future work are,

- The correlation can be done on the whole body using the robovibrometer cameras versus body rig test
- The whole car including all components is tested to evaluate the global damping for the entire car

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