TECHNISCHE UNIVERSITÄT DRESDEN

BOMBARDIER

Fakultät Maschinenwesen Professur für Dynamik und Mechanismentechnik

Numerical Study of the Structure-Borne Sound Transmission of a Bogie using Multibody Simulation

Dipl.-Ing. Johannes Woller

Dipl.-Ing. Claudius Lein

Dipl.-Ing. Robert Zeidler

Prof. Dr.-Ing. Michael Beitelschmidt

Darmstadt, 9th of November 2016

Structure of the presentation

Cooperation between the TU Dresden and Bombardier Transportation GmbH

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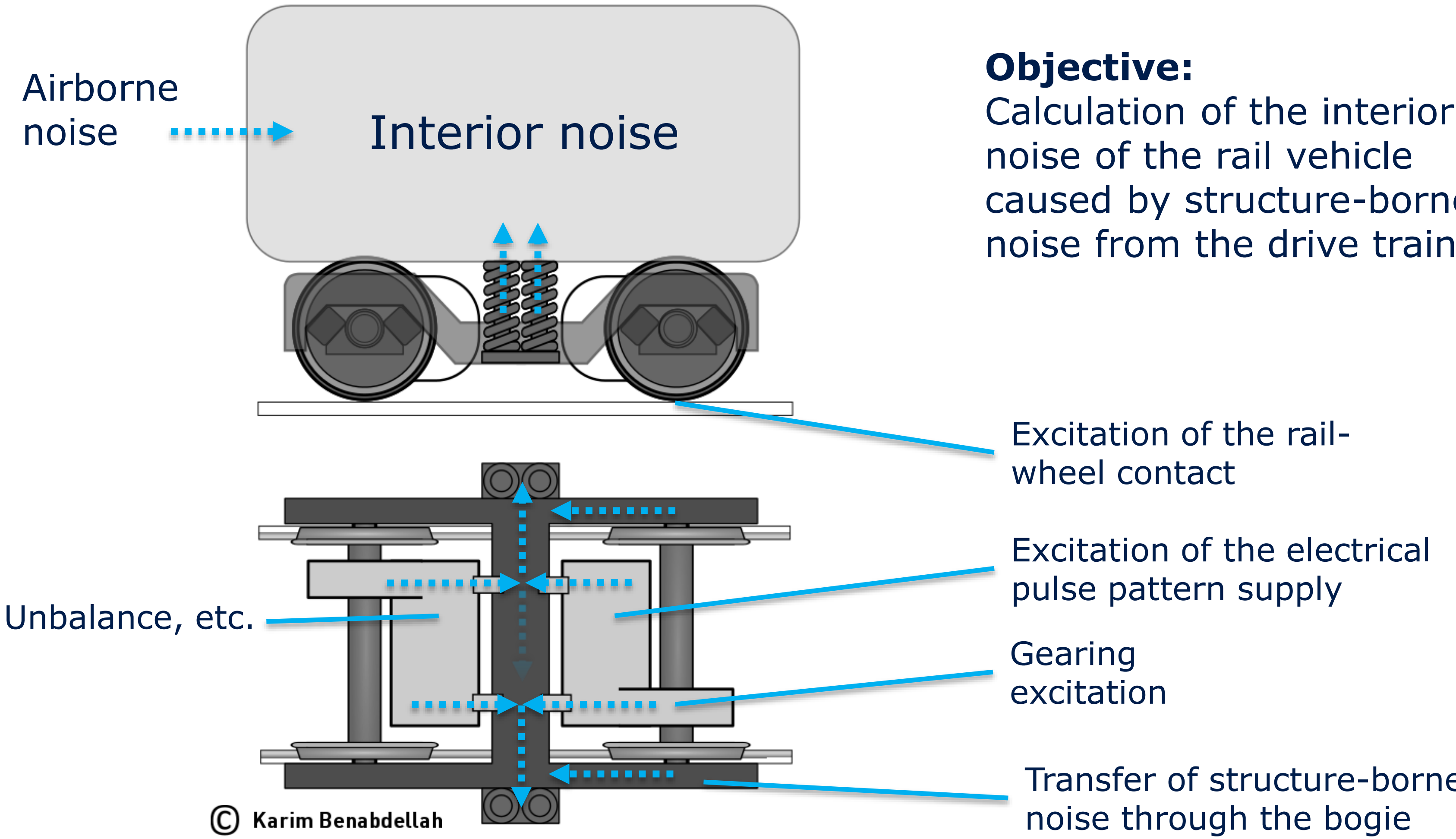
Bombardier
Center of Competence

„Railway Vehicle Engineering Integration Center” (since 2007)

Subsequent results are part of the research project:

Topic:	Seamless integration of standardized Noise–Vibration-Harshness (NVH) calculations in the development process of railway vehicle powertrains
Project head:	Prof. Dr.-Ing. Michael Beitelschmidt
Employee responsible:	Dipl.-Ing. Johannes Woller
Duration of the project:	4 years starting 2014

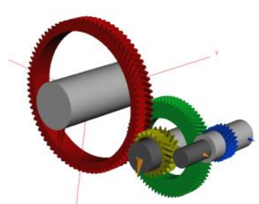
Objective:
Calculation of the interior noise of the rail vehicle caused by structure-borne noise from the drive train

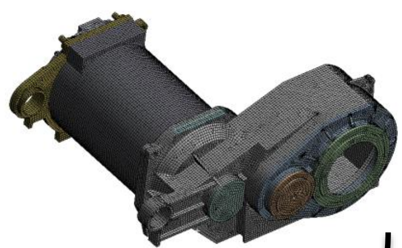


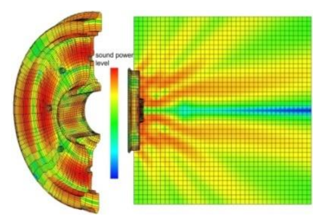
State of the art calculation methods in the vehicle acoustics

Excitation → Transmission → Radiation → Sound propagation

MBS → FEM → FEM/BEM → BEM/FEM/SEA

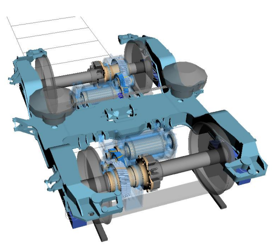






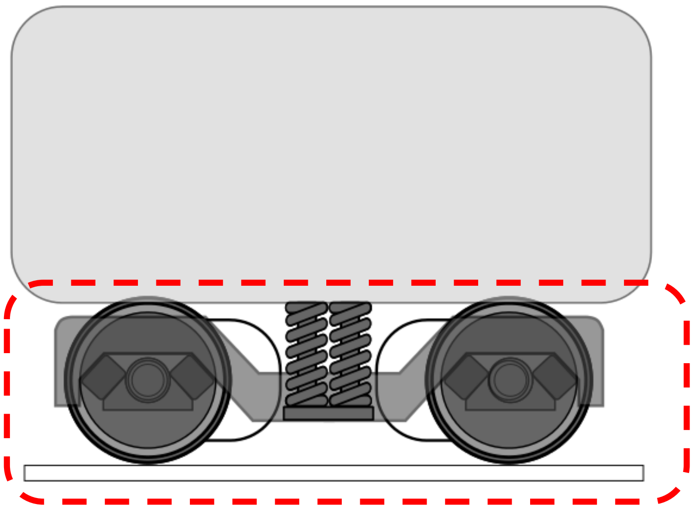
SIMPACK

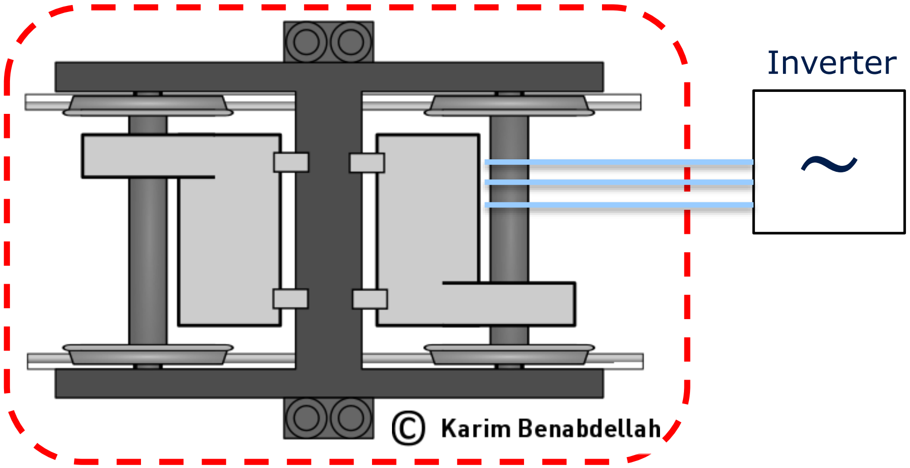
EMBS + MOR



SEA	Statistical Energy Analysis
BEM	Boundary Element Method
FEM	Finite-Element Method
EMBS	Elastic Multi Body Simulation
MOR	Model Order Reduction

Model boundaries:





Modeling approach:

- Simulation of the mechanical vibration origin and transmission of the motor bogie by means of flexible multibody simulation using SIMPACK

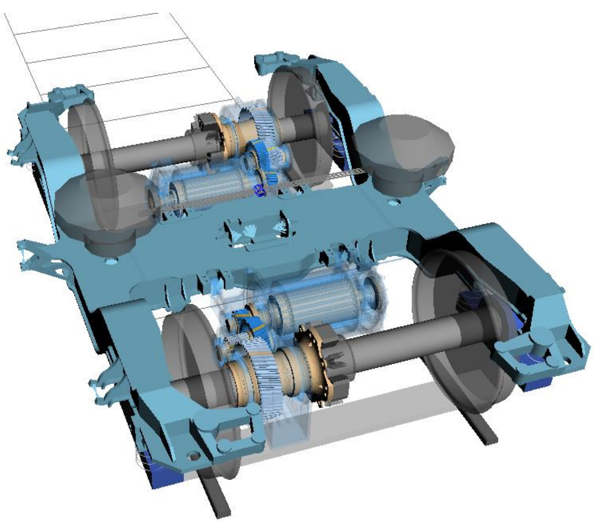
- Investigation of a selected train project at Bombardier

Outcome:

- Calculation of dominant NVH performance of up to 1000 Hz

- Providing quantitative structure-borne sound variables at the coupling points to the car body

In this context the bogie frame is a crucial component:



- Nearly all transfer paths from the structure-borne sound excitation in the traction unit are formed by the bogie frame
- First natural frequency is significantly below 100 Hz
- Low damping (welded structure)
- High mode density
- Many connection points to superstructure and to the drive train
- The bogie frame needs to be considered as an elastic body in the multibody simulation

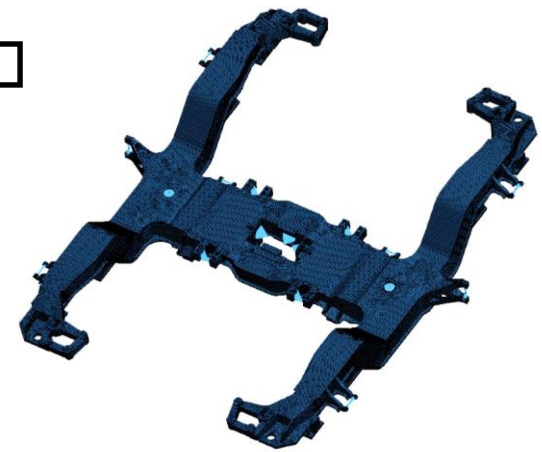
Structure of the presentation

Structural flexibility approximation: Floating Frame of Reference

FE-discrete equation of motion:

$$\underbrace{\mathbf{M}}_{n}\ddot{\mathbf{x}} + \mathbf{D}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \underbrace{\mathbf{B}}_{q}\cdot\mathbf{u} \qquad \boxed{n \in \left[10^4..10^7\right]}$$

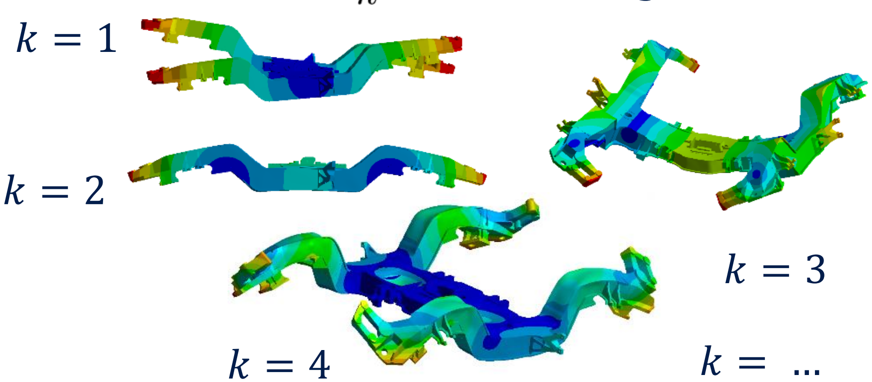
Finite-Element Model



Model Order Reduction (MOR)

$$\underbrace{\mathbf{M}_R}_{m}\ddot{\mathbf{x}}_R + \mathbf{D}_R\dot{\mathbf{x}}_R + \mathbf{K}_R\mathbf{x}_R = \underbrace{\mathbf{B}_R}_{q}\cdot\mathbf{u} \qquad \boxed{m \ll n}$$

Φ_k Reduced eigensolution



Global Ritz Approach

$$\underbrace{\mathbf{u}(\mathbf{r}_P, t) \approx \sum_{k=1}^{n_q} \Phi_k(\mathbf{r}_P) q_k(t)}_{}$$

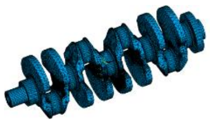
Deformation of the bogie

Standard work flow

FEM–SIMPACK

*_eigen.rst
*_struct.sub
*_cad.cdb

FEM ANSYS → **MOR** ANSYS → **SIMPACK Interface** → *.fbi → **EMBS**



Superelement:

- Guyan / CMS

FlexModal-features:

- e.g. FRM

Alternative Work Flow

FEM–**MORPACK**-SIMPACK

FEM ANSYS → MACRO → **MORPACK Interface** → *.sid, *.obj → **EMBS**

- system matrices + mapping
- nodes + master nodes
- (eigenvalues + eigenvectors)

Main-features:

- alternative MOR
- modal correlation
- minimal model

- Standard MOR-techniques for dynamic reduction :
  - Component Mode Synthesis (CMS)
    - Implemented in commercial FE-software
    - Well established

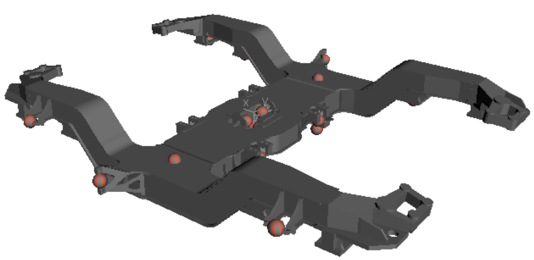
- Modern MOR-techniques:
  - Krylov Subspace Method (KSM)
    - Approximates the frequency response function by a Taylor series
    - Promising better MOR quality while getting smaller models

Following questions will be focused in this presentation:

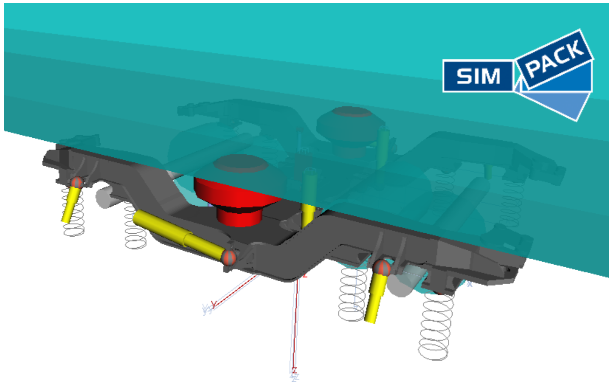
1. Are the results generated with a **modern KSM** similar to those of a **conventional CMS** model, at the example of a bogie frame?

2. What is the **influence of the installation state** ?

unconstrained

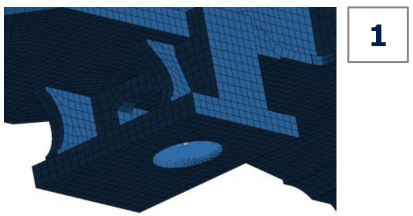


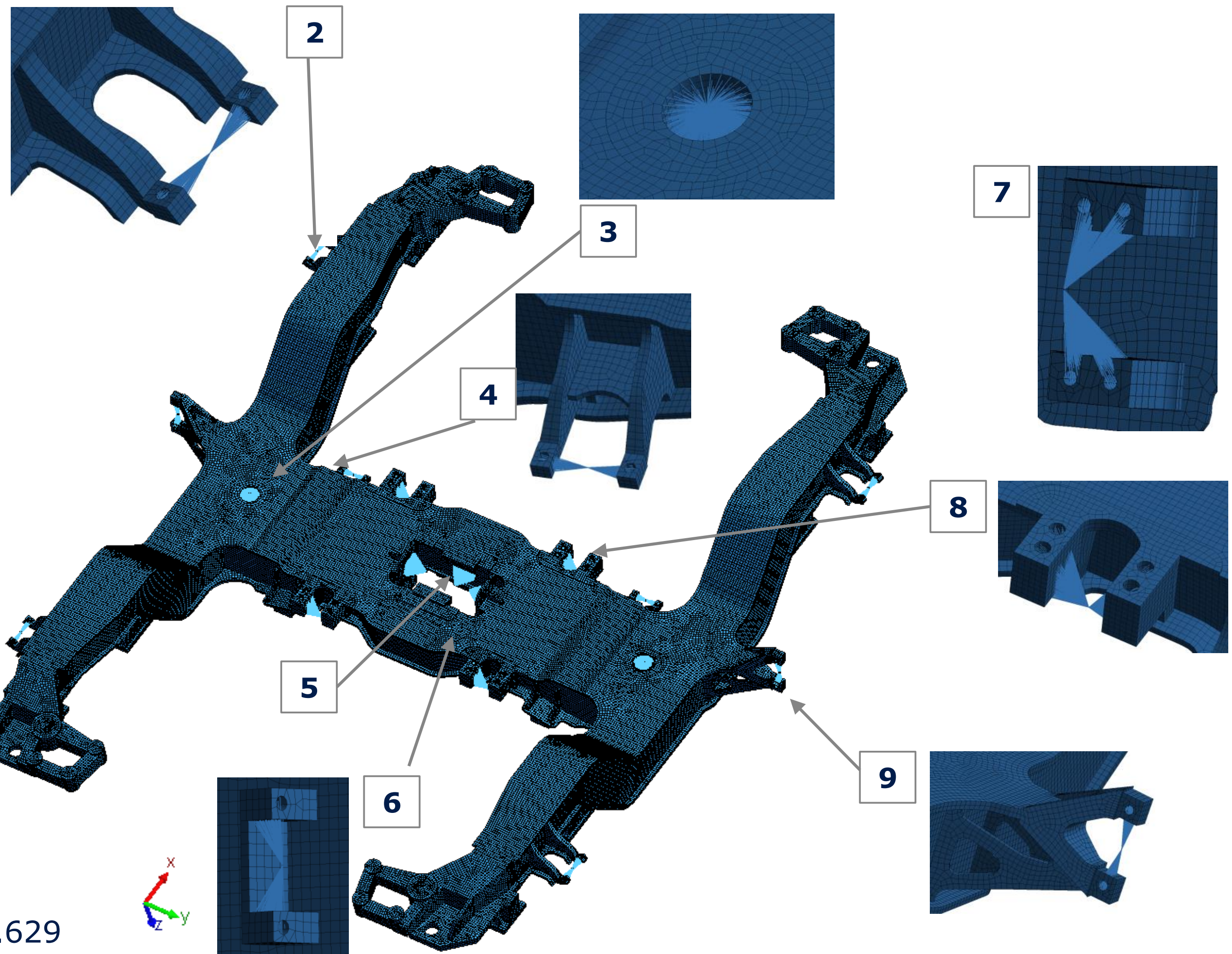
installed in a EMBS



Structure of the presentation

- Motivation
- Question
- **Modeling of the bogie frame**
- Comparison of the different modeling approaches
- Global evaluation and conclusion





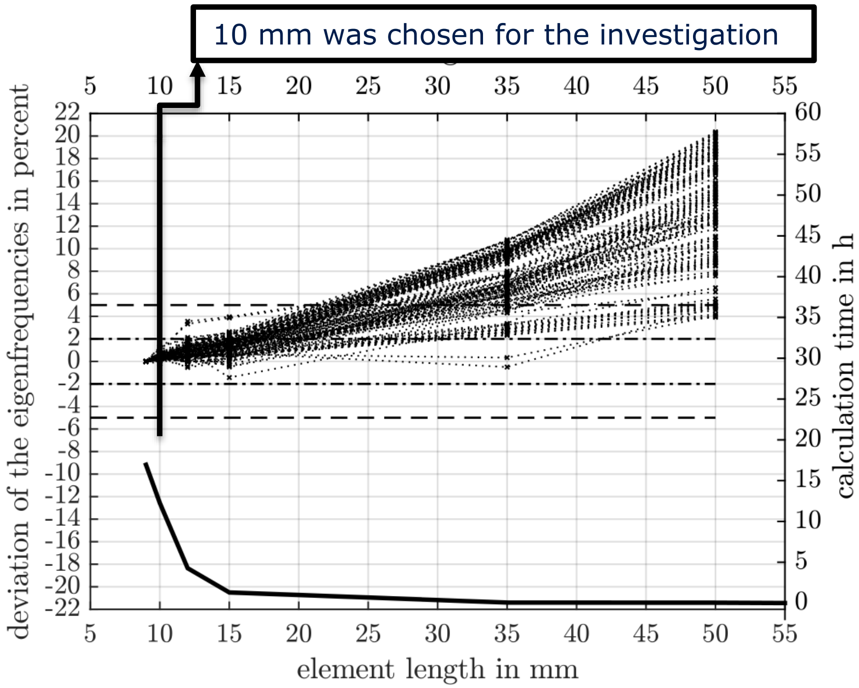
	Connections	N
1	Primary spring	8
2	Vertical damper	4
3	Secondary spring	2
4	Secondary damper	2
5	Pin	4
6	Pin damper	1
7	Anti-roll bar	2
8	Motor suspension	4
9	Yaw damper	2

$\Sigma = 29$

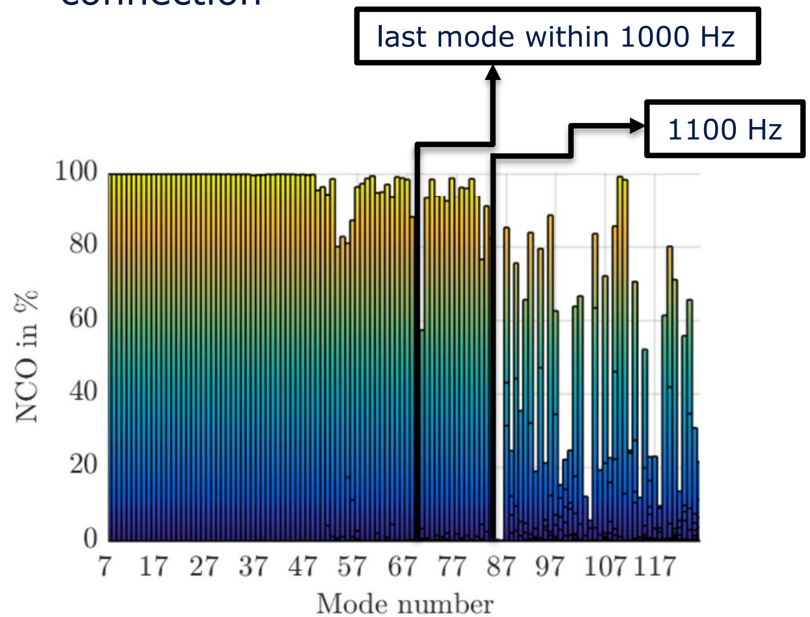
Number of Elements: 217.629
Degree of Freedom: 2.259.837

Validation of the finite element model

a) Mesh sensitivity to the eigenfrequencies



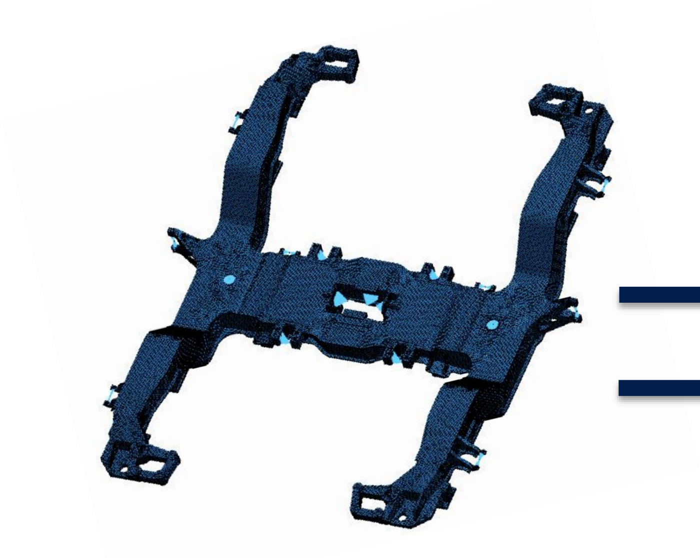
b) Influence of the master node connection



Normalized Cross Orthogonality Check
(model with masternodes and without)

The calculation was performed with the FE-software ANSYS on a workstation with Intel Core i7-2600 and 16 GB RAM. Due to the limited RAM the calculation was performed out of core.

Finite-Element Model



Number of Elements: 217.629
Degree of Freedom: 2.259.837

MORPACK Interface

Calculation time $\approx 42\ h$

Rational Krylov Subspace Method
+ Mode Truncation by MORPACK

Degree of freedom: 255

Calculation time $\approx 24\ h$

Component-Mode-Synthesis with Fixed-Interface algorithm by MORPACK

Degree of freedom: 474

Validation of the reduced models against the full model

a) Normalized Cross Orthogonality Check

$$\mathrm{NCO}_{ij} = \frac{\left(\boldsymbol{\phi}_{r,i}^{H} \cdot \mathbf{W} \cdot \boldsymbol{\phi}_{v,j}\right)^2}{\left(\boldsymbol{\phi}_{r,i}^{H} \cdot \mathbf{W} \cdot \boldsymbol{\phi}_{r,i}\right) \cdot \left(\boldsymbol{\phi}_{v,j}^{H} \cdot \mathbf{W} \cdot \boldsymbol{\phi}_{v,j}\right)}$$

b) Normalized Relative Frequency Difference

$$\mathrm{NRFD}_i = \frac{|f_{r,i} - f_{v,i}|}{f_{r,i}}$$

Model	Number of Modes	a) Last mode NCO(<95%)	b) Last natural frequency NRFD(<1%)
CMS	474	1244Hz / Mode 99	1259 Hz / Mode 101
KSM	255	2016 Hz / Mode 251	2027 Hz / Mode 255

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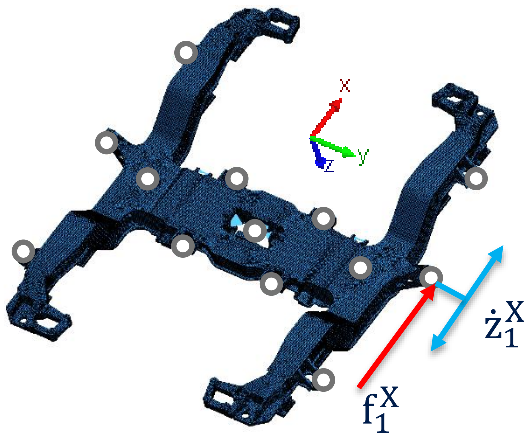
Calculation Method: Using State Space Matrices Export to Matlab

→ Linear time independent system (LTI) at linearization point $\mathbf{x_0}$

→ Multiple Input Multiple Output (MIMO):

Inputs/Outputs: $\mathbf{u} = \mathbf{f}$ (forces), $\mathbf{y} = \dot{\mathbf{z}}$ (velocities)

State Space formulation:



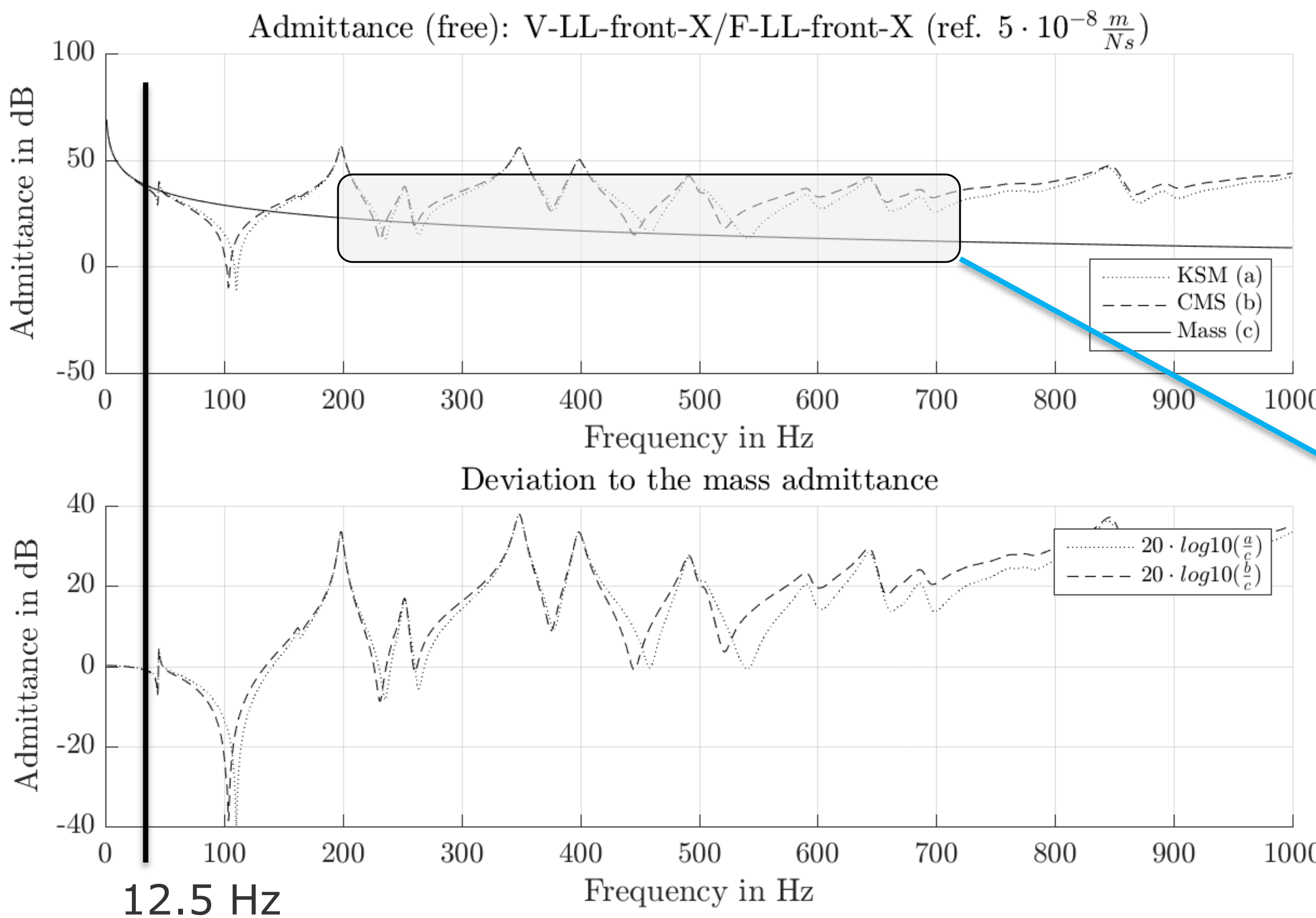
Laplace Transformation and FRF calculation in MATLAB

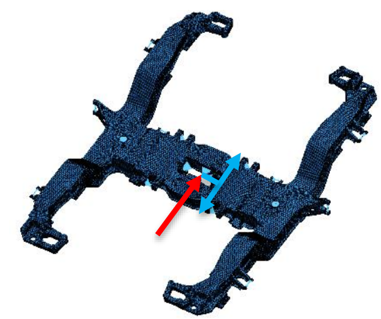
$$\begin{aligned}\dot{\mathbf{x}} &= \mathbf{Ax} + \mathbf{Bu} \\ \mathbf{y} &= \mathbf{Cx} + \mathbf{Du}\end{aligned} \quad \Rightarrow \quad \underline{\mathbf{Y}}_{\mathbf{S}} = \mathbf{C}\left[\underline{\mathbf{s}}(\mathbf{I} - \mathbf{A})^{-1}\mathbf{B}\right] + \mathbf{D}$$

⇓

Complex admittance matrix of the whole system

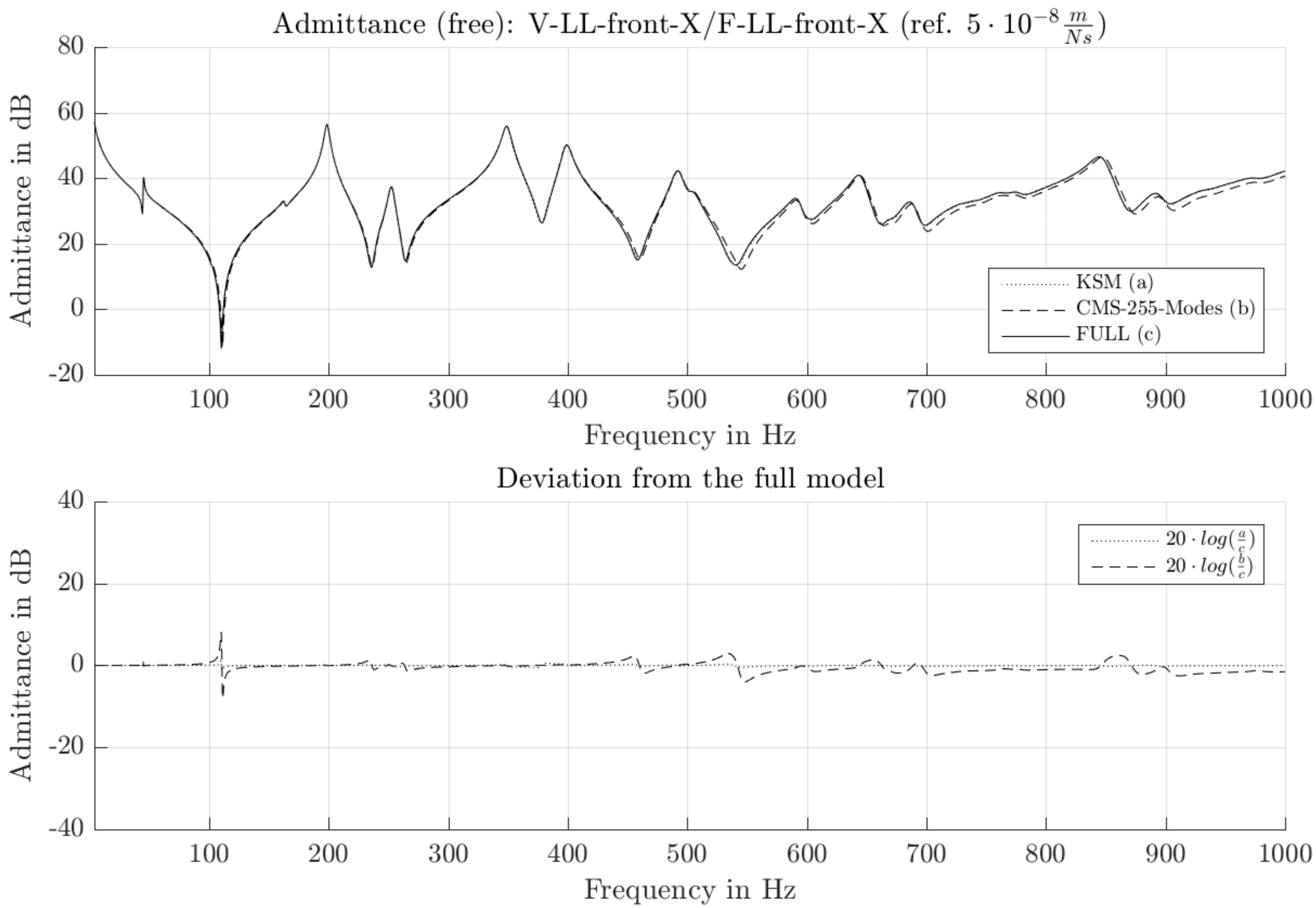
Plausibility test → Comparison to the Mass admittance (1159 kg)

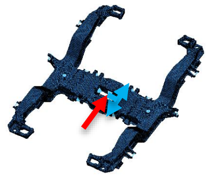




Deviations between CMS and KSM In the antiresonances → Faulty higher modes of CMS

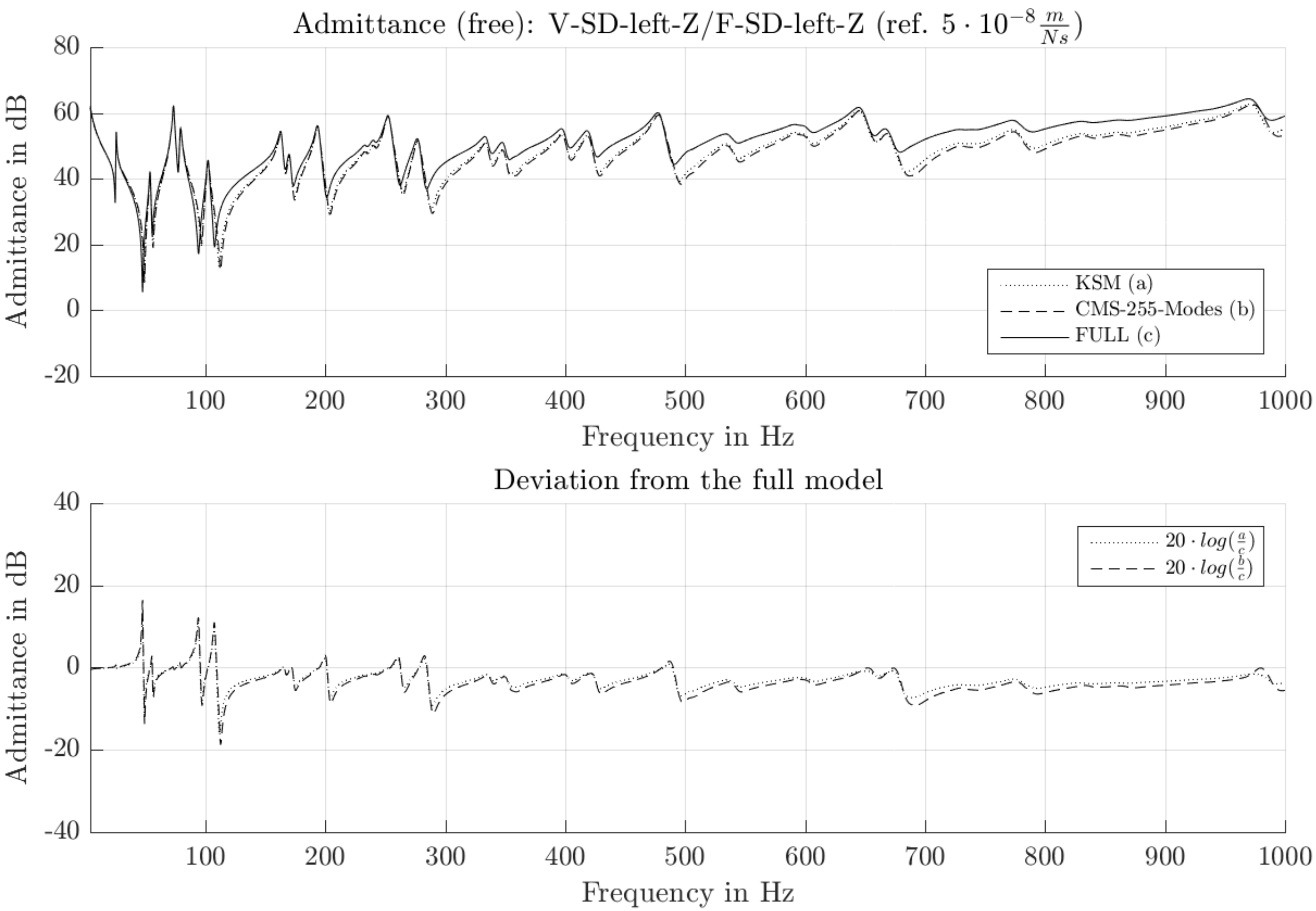
Plausibility test → Comparison to the full model

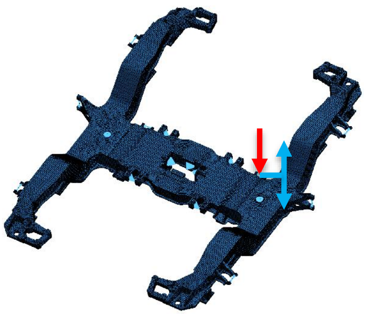




- Good agreement with full model
- KSM performs slightly better especially between the eigenfrequencies
- Both models show a similar behavior

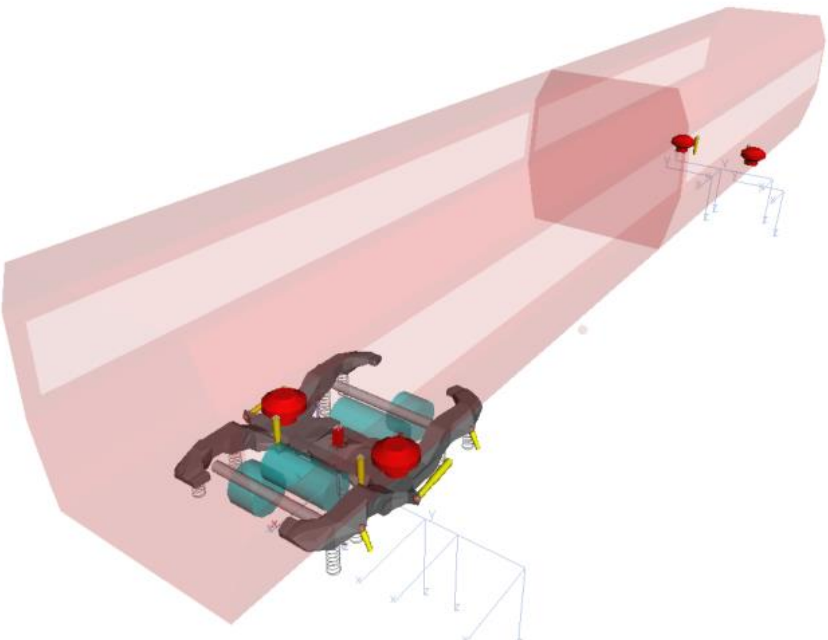
Fit of the reduction depends on the transferpath:





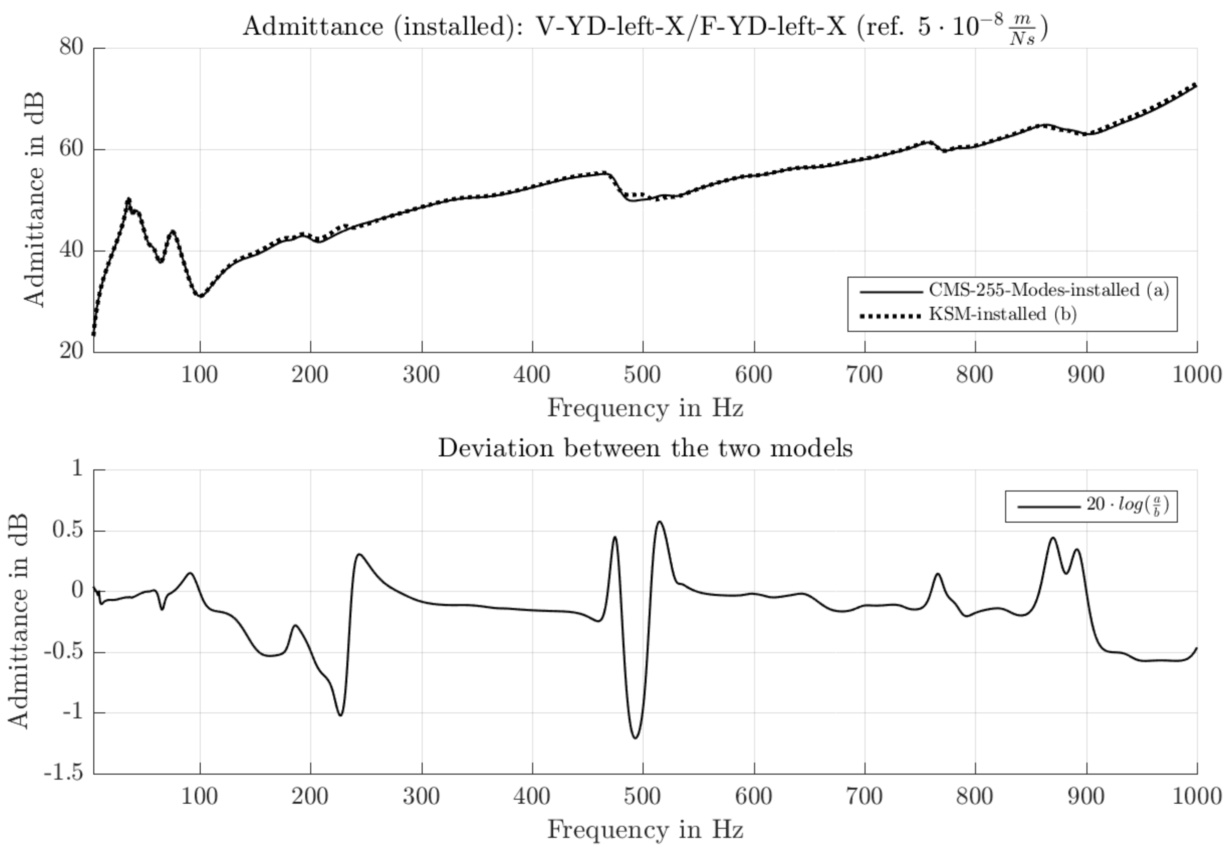
Bad agreement between full and reduced model due to lack of higher modes

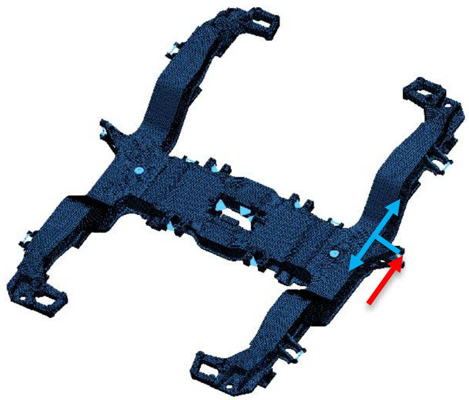
The installed state



- Equipped with all relevant spring and damper elements within the EMBS
- Lower model boundary is given by the primary spring stage
- Car body is supported on the bogie
- Based on state of the art MBS modeling in the area of derailment safety and comfort calculation
- The wheelset is assumed to be rigid

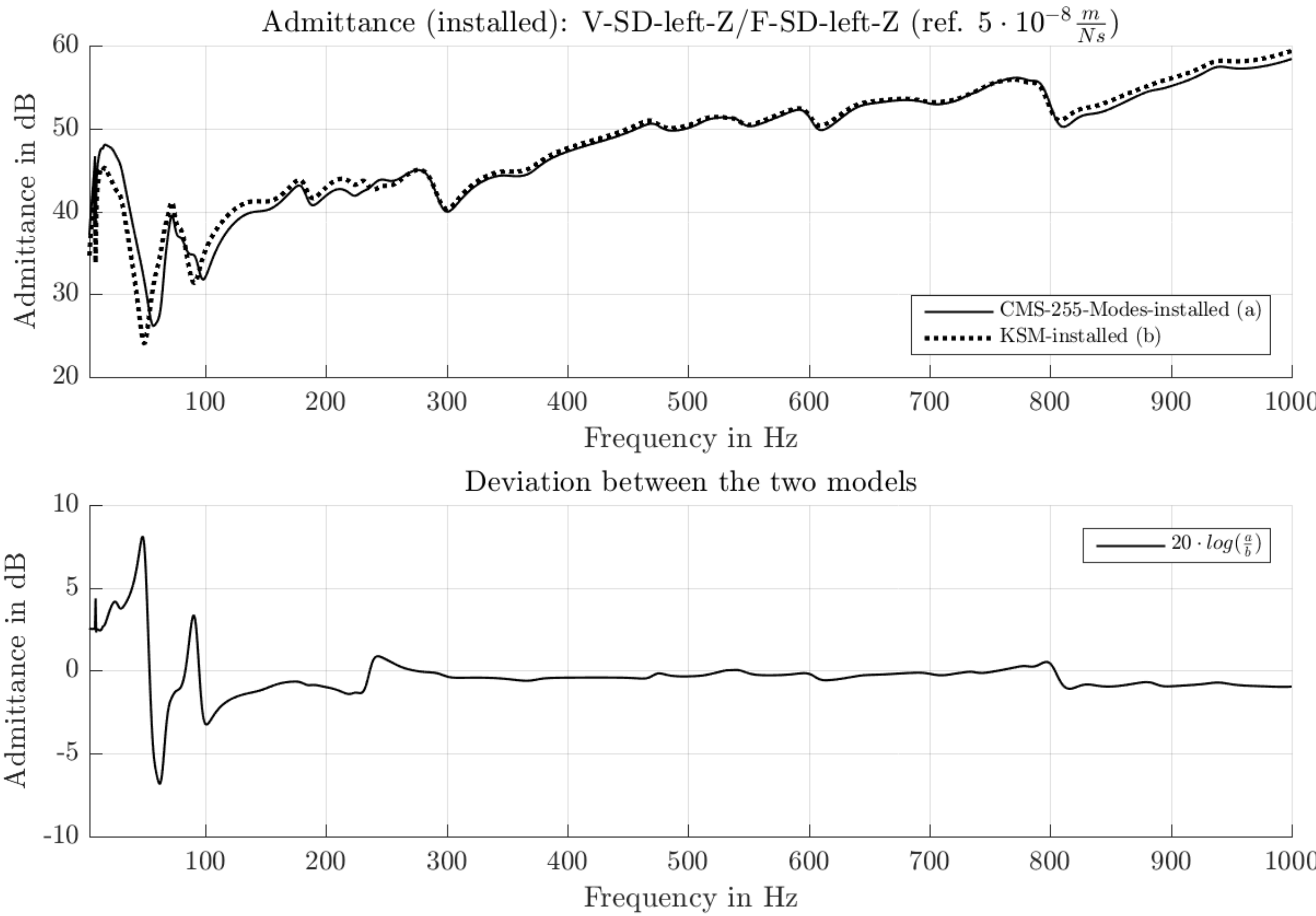
Installed: KSM vs. CMS

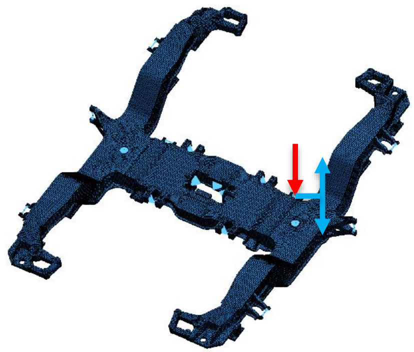




- Both models show a similar behavior

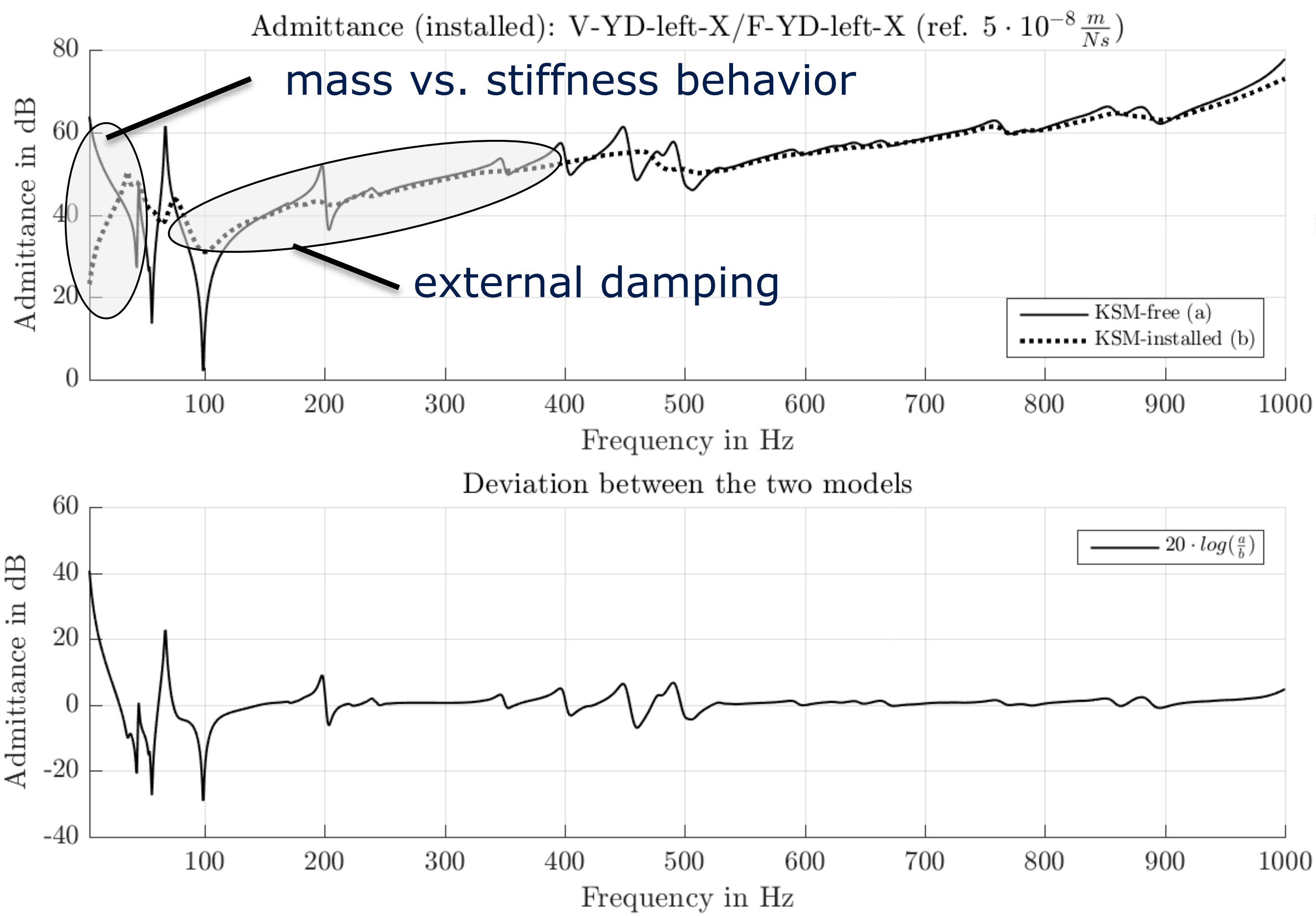
Installed: KSM vs. CMS

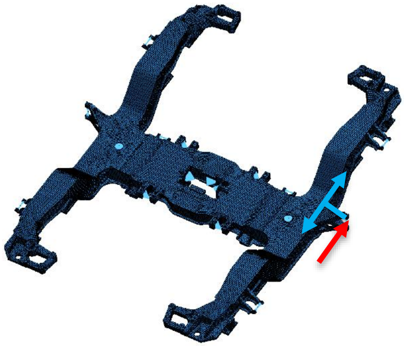




- Similarity depends on the transferpath

Installed vs. Free (KSM)





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Overall Comparison 174x174 admittance curves KSM vs. CMS

- Mean deviation overall $$\Delta Y_{mean} = \overline{\left|20 \cdot \log \frac{Y_1(\omega)}{Y_2(\omega)}\right|}$$
- Maximum mean deviation $$\Delta \hat{Y}_{mean} = \max_i \overline{\left|20 \cdot \log \frac{Y_{1i}(\omega)}{Y_{2i}(\omega)}\right|}$$
- Maximum deviation $$\Delta \hat{Y} = \max_i \left[\max_\omega \left|20 \cdot \log \frac{Y_{1i}(\omega)}{Y_{2i}(\omega)}\right|\right]$$

		ΔY_{mean}	$\Delta \hat{Y}_{mean}$	$\Delta \hat{Y}$
KSM vs. CMS	free vs. free	1.4±1.8 dB	25 dB	74.3 dB
KSM vs. CMS	installed vs. installed	4.3 ±2.8 dB	22.5 dB	74.4 dB
KSM vs. KSM	installed vs. free	13.2±4.8 dB	36.8 dB	36.9 dB
CMS vs. CMS	installed vs. free	13.5 ±4.1 dB	36.2 dB	36.3 dB

Conclusion

- KSM250 model approximates the full model better than the CMS300
- Significant errors arise by poorly correlated modes from the higher frequency range by the CMS model
- Both models provide comparable results for the free and for the installed state
- KSM reduced models can replace the CMS models
- The bogie in installed state has a totally different low frequency behavior in some points and the external damping smoothes the transfer functions
- Nevertheless the global trend of the transfer functions is maintained comparing the installed and the free bogie

Thank you for your attention!

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