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

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Darmstadt, 9th of November 2016
Structure of the presentation

- Motivation
- Question
- Modeling of the bogie frame
- Comparison of the different modeling approaches
- Global evaluation and conclusion
Cooperation between the TU Dresden and Bombardier Transportation GmbH

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

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

- **Interior noise**
- **Airborne noise**
- **Excitation of the rail-wheel contact**
- **Excitation of the electrical pulse pattern supply**
- **Gearing excitation**
- **Transfer of structure-borne noise through the bogie**
- **Unbalance, etc.**

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State of the art calculation methods in the vehicle acoustics

State of the art calculation methods in the vehicle acoustics

- Excitation
  - MBS

- Transmission
  - FEM

- Radiation
  - FEM/BEM

- Sound propagation
  - BEM/FEM/SEA

- EMBS + MOR

Statistical Energy Analysis
Boundary Element Method
Finite-Element Method
Elastic Multi Body Simulation
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
Motivation

Question

Modeling of the bogie frame

Comparison of the different modeling approaches

Global evaluation and conclusion
Structural flexibility approximation: Floating Frame of Reference

FE-discrete equation of motion:

\[ M \ddot{x} + D \dot{x} + K x = B u \]

Model Order Reduction (MOR)

\[ M_R \ddot{x}_R + D_R \dot{x}_R + K_R x_R = B_R u \]

Reduced eigensolution

\[ \Phi_k \]

Global Ritz Approach

\[ u(r_P, t) \approx \sum_{k=1}^{n_q} \Phi_k(r_P)q_k(t) \]

Deformation of the bogie
Standard work flow

**FEM–SIMPACK**

FEM → MOR

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

Superelement:
- Guyan / CMS

SIMPACK Interface

Alternative Work Flow

**FEM–MORPACK–SIMPACK**

FEM → MACRO

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

MORPACK Interface

FlexModal-features:
- e.g. FRM

Main-features:
- alternative MOR
- modal correlation
- minimal model

EMBS

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• 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
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Connections

<table>
<thead>
<tr>
<th>Number</th>
<th>Description</th>
<th>N</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Primary spring</td>
<td>8</td>
</tr>
<tr>
<td>2</td>
<td>Vertical damper</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>Secondary spring</td>
<td>2</td>
</tr>
<tr>
<td>4</td>
<td>Secondary damper</td>
<td>2</td>
</tr>
<tr>
<td>5</td>
<td>Pin</td>
<td>4</td>
</tr>
<tr>
<td>6</td>
<td>Pin damper</td>
<td>1</td>
</tr>
<tr>
<td>7</td>
<td>Anti-roll bar</td>
<td>2</td>
</tr>
<tr>
<td>8</td>
<td>Motor suspension</td>
<td>4</td>
</tr>
<tr>
<td>9</td>
<td>Yaw damper</td>
<td>2</td>
</tr>
</tbody>
</table>

Σ = 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

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.

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Rational Krylov Subspace Method + Mode Truncation by MORPACK

Degree of freedom: 255

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

Degree of freedom: 474

Calculation time $\approx 24\ h$

Finite-Element Model

Number of Elements: 217,629
Degree of Freedom: 2,259,837

Calculation time $\approx 42\ h$

MORPACK Interface
Validation of the reduced models against the full model

- **a) Normalized Cross Orthogonality Check**

\[
\text{NCO}_{ij} = \frac{(\phi^H_{r,i} \cdot W \cdot \phi_{v,j})^2}{(\phi^H_{r,i} \cdot W \cdot \phi_{r,i}) \cdot (\phi^H_{v,j} \cdot W \cdot \phi_{v,j})}
\]

- **b) Normalized Relative Frequency Difference**

\[
\text{NRFD}_i = \frac{|f_{r,i} - f_{v,i}|}{f_{r,i}}
\]

<table>
<thead>
<tr>
<th>Model</th>
<th>Number of Modes</th>
<th>a) Last mode NCO(&lt;95%)</th>
<th>b) Last natural frequency NRFD(&lt;1%)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>CMS</strong></td>
<td>474</td>
<td>1244 Hz / Mode 99</td>
<td>1259 Hz / Mode 101</td>
</tr>
<tr>
<td><strong>KSM</strong></td>
<td>255</td>
<td>2016 Hz / Mode 251</td>
<td>2027 Hz / Mode 255</td>
</tr>
</tbody>
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Calculation Method: Using State Space Matrices Export to Matlab

→ Linear time independent system (LTI) at linearization point \( x_0 \)
→ Multiple Input Multiple Output (MIMO):

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

State Space formulation:

Laplace Transformation and FRF calculation in MATLAB

\[
\dot{x} = Ax + Bu \\
y = Cx + Du
\]

\[
Y_S = C \left[ s(I - A)^{-1} B \right] + D
\]

Complex admittance matrix of the whole system
Plausibility test $\rightarrow$ Comparison to the Mass admittance (1159 kg)

Admittance (free): V-LL-front-X/F-LL-front-X (ref. $5 \cdot 10^{-8} \frac{m}{Ns}$)

Deviations between CMS and KSM in the antiresonances $\rightarrow$ Faulty higher modes of CMS
Plausibility test → Comparison to the full model

Admittance (free): V-LL-front-X/F-LL-front-X (ref. $5 \cdot 10^{-8} \frac{m}{N_s}$)

- 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:

- Admittance (free): V-SD-left-Z/F-SD-left-Z (ref. $5 \cdot 10^{-8} \frac{m}{Ns}$)

- Deviation from the full model

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

Admittance (installed): V-YD-left-X/F-YD-left-X (ref. $5 \cdot 10^{-8} \frac{m}{N_s}$)

Deviation between the two models

Admittance in dB

Frequency in Hz

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Installed: KSM vs. CMS

- Similarity depends on the transferpath
Installed vs. Free (KSM)

Admittance (installed): V-YD-left-X/F-YD-left-X (ref. $5 \cdot 10^{-8}$ $\frac{m}{Ns}$)

mass vs. stiffness behavior

external damping

Frequency in Hz

Deviation between the two models

Frequency in Hz
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Overall Comparison 174x174 admittance curves KSM vs. CMS

- Mean deviation overall

\[ \Delta Y_{\text{mean}} = \sqrt{20 \cdot \log \left( \frac{Y_1(\omega)}{Y_2(\omega)} \right)} \]

- Maximum mean deviation

\[ \hat{\Delta Y}_{\text{mean}} = \max_i \left[ 20 \cdot \log \frac{Y_{1i}(\omega)}{Y_{2i}(\omega)} \right] \]

- Maximum deviation

\[ \hat{\Delta Y} = \max_i \left[ \max_{\omega} \left| 20 \cdot \log \frac{Y_{1i}(\omega)}{Y_{2i}(\omega)} \right| \right] \]

<table>
<thead>
<tr>
<th></th>
<th>( \Delta Y_{\text{mean}} )</th>
<th>( \hat{\Delta Y}_{\text{mean}} )</th>
<th>( \hat{\Delta Y} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>KSM vs. CMS</td>
<td>free vs. free</td>
<td>1.4±1.8 dB</td>
<td>25 dB</td>
</tr>
<tr>
<td>KSM vs. CMS</td>
<td>installed vs. installed</td>
<td>4.3 ±2.8 dB</td>
<td>22.5 dB</td>
</tr>
<tr>
<td>KSM vs. KSM</td>
<td>installed vs. free</td>
<td>13.2±4.8 dB</td>
<td>36.8 dB</td>
</tr>
<tr>
<td>CMS vs. CMS</td>
<td>installed vs. free</td>
<td>13.5 ±4.1 dB</td>
<td>36.2 dB</td>
</tr>
</tbody>
</table>
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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