

The Application of Dynamic Reduction in Modelling an Exhaust System as a Flexible Body

SIMPACK User Meeting 2001

Andreas Raith, 14.11.01



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- **Flexible Bodies in SIMPACK**

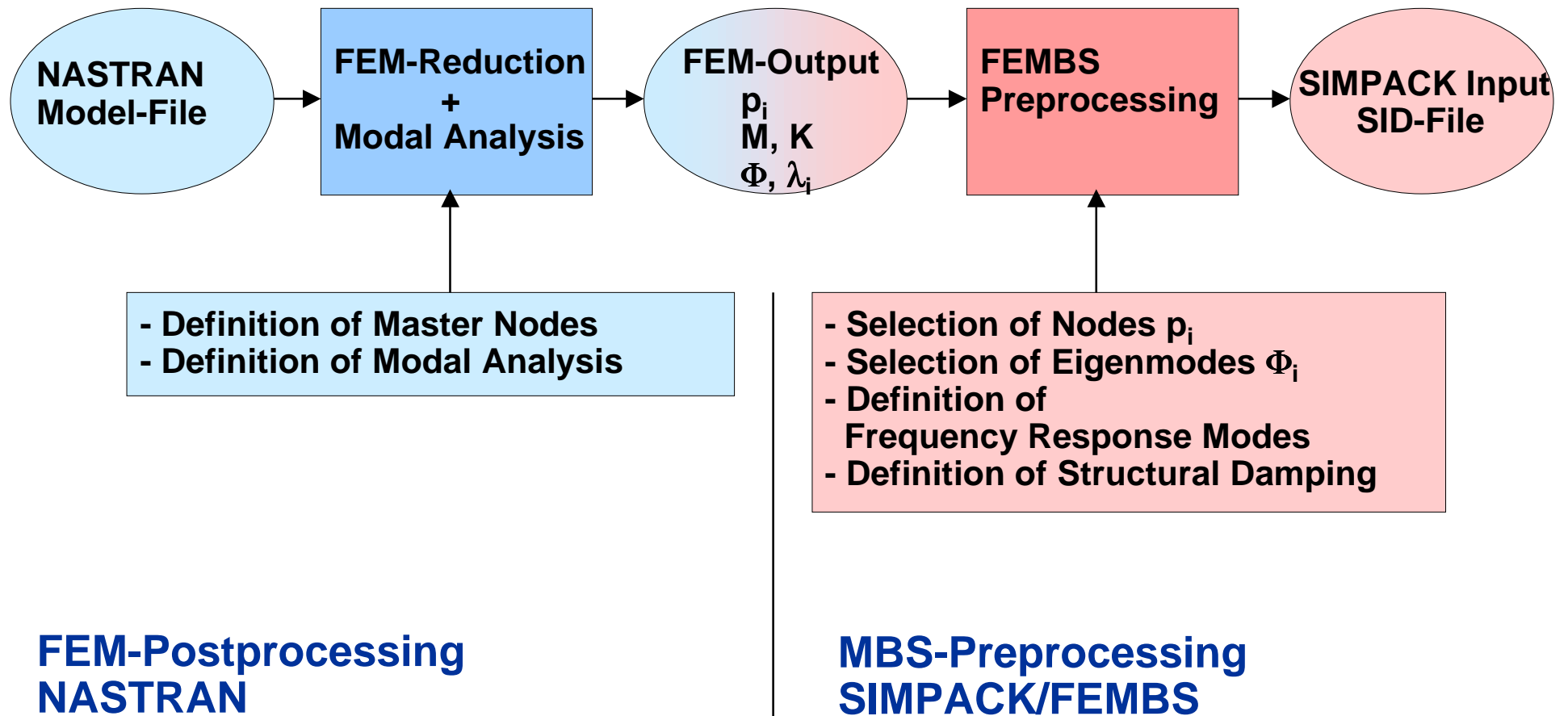
- **Static / Dynamic FEM-Reduction**

- **Exhaust System Example**
 - Reduction
 - Frequency Response Modes
 - Validation

- **Full Vehicle Model Analysis**

Flexible Bodies in SIMPACK

Application Pattern for NASTRAN



Static Reduction

FEM-Model Equations

$$\mathbf{M}_f \ddot{\mathbf{u}}_f + \mathbf{K}_f \mathbf{u}_f = \mathbf{P}_f$$

Dimension n_f : **Physical FEM-Degrees of Freedom**

Separation of **Master Nodes** \mathbf{u}_a from **Internal Nodes** \mathbf{u}_o

$$\begin{bmatrix} M_{aa} & M_{ao} \\ M_{oa} & M_{oo} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{u}}_a \\ \ddot{\mathbf{u}}_o \end{bmatrix} + \begin{bmatrix} K_{aa} & K_{ao} \\ K_{oa} & K_{oo} \end{bmatrix} \begin{bmatrix} \mathbf{u}_a \\ \mathbf{u}_o \end{bmatrix} = \begin{bmatrix} \mathbf{P}_a \\ \mathbf{P}_o \end{bmatrix}$$

Elimination of Internal Nodes

$$\mathbf{u}_o = -K_{oo}^{-1} [K_{oa} \mathbf{u}_a + M_{oa} \ddot{\mathbf{u}}_a + M_{oo} \ddot{\mathbf{u}}_o - \mathbf{P}_o]$$

$$\ddot{\mathbf{u}}_o = -K_{oo}^{-1} K_{oa} \ddot{\mathbf{u}}_a$$

Guyan Approximation:

Static Solution 2x differentiated

Reduced Model Equations

$$\mathbf{M}_{aa} \ddot{\mathbf{u}}_a + \mathbf{K}_{aa} \mathbf{u}_a = \mathbf{P}_a$$

Dimension n_a : **Master Nodes-Degrees of Freedom**

Dynamic Reduction

Approximation of Internal Nodes

$$\mathbf{u}_o = -K_{oo}^{-1} [K_{oa} \mathbf{u}_a + M_{oa} \ddot{\mathbf{u}}_a + M_{oo} \ddot{\mathbf{u}}_o - \mathbf{P}_o]$$

$$\ddot{\mathbf{u}}_o = -K_{oo}^{-1} K_{oa} \ddot{\mathbf{u}}_a + \Phi_{oq} \ddot{\mathbf{u}}_q$$

Generalised Coordinates
and Form Functions















Reduced Model Equations

$$\begin{bmatrix} \mathbf{M}_{aa} & \mathbf{M}_{aq} \\ \mathbf{M}_{qa} & \mathbf{M}_{qq} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{u}}_a \\ \ddot{\mathbf{u}}_o \end{bmatrix} + \begin{bmatrix} \mathbf{K}_{aa} & \mathbf{0} \\ \mathbf{0} & \mathbf{K}_{qq} \end{bmatrix} \begin{bmatrix} \mathbf{u}_a \\ \mathbf{u}_q \end{bmatrix} = \begin{bmatrix} \mathbf{P}_a \\ \mathbf{P}_q \end{bmatrix}$$

Dimension n_a+n_q : **Master Nodes - DoF +
Generalised Coordinates**

➡ **The Dynamics of Internal Nodes** is taken into account by **Generalised Coordinates** and **Form Functions**

Properties of the Reduction

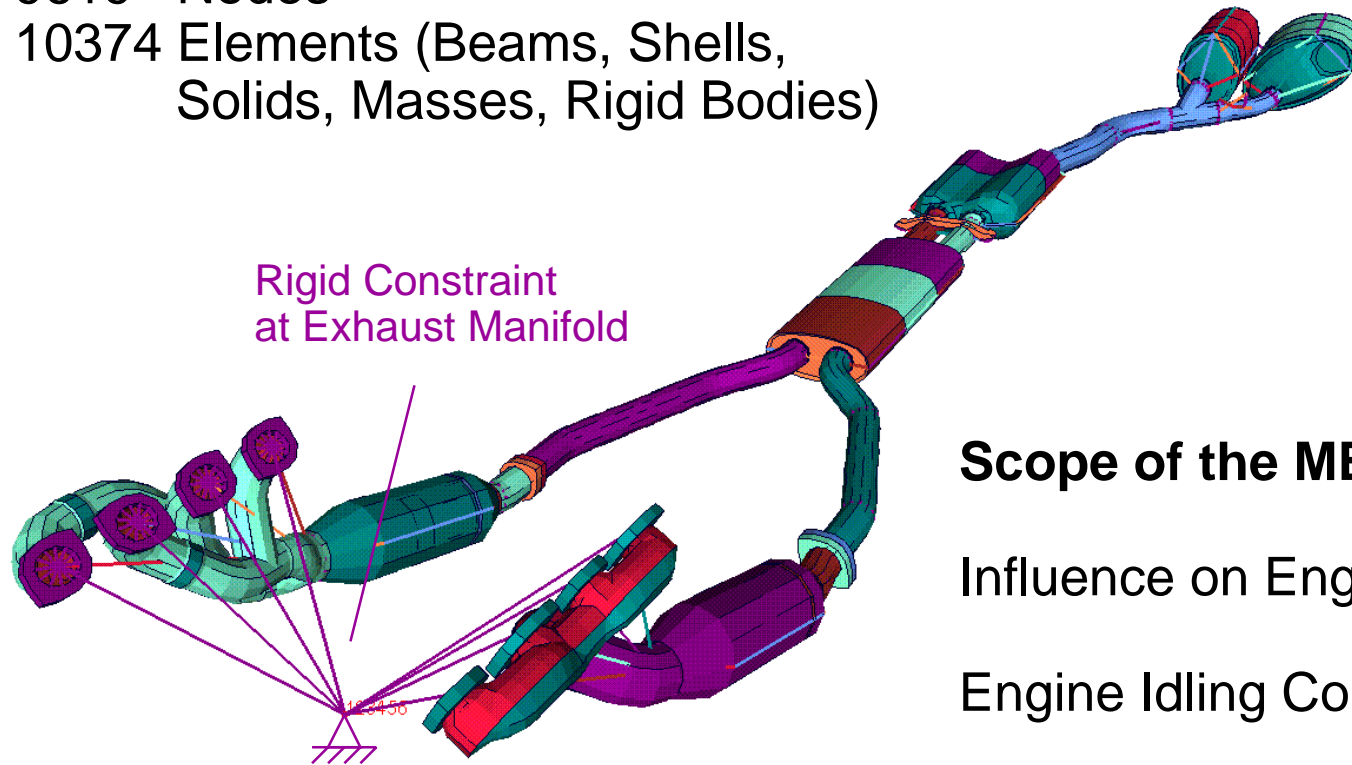
	Static	Dynamic
Stiffness Matrix	Exact  	Exact  
Mass matrix	Rigid body characteristics exact 	Rigid body characteristics exact 
Eigenfrequency,-modes	Dependant on distribution of Master Nodes  	Controllable by the number of frequencies considered  
Modelling Effort	Dense distribution of master nodes and check of the Eigenfrequencies is necessary  	Minimum number of Master Nodes at the coupling nodes to the MBS is possible  

Exhaust System Example

FEM-Model

9619 Nodes

10374 Elements (Beams, Shells,
Solids, Masses, Rigid Bodies)



Scope of the MBS-Simulation

Influence on Engine Shake (5-30Hz)

Engine Idling Comfort (20-60Hz)

Isolation of Structure Borne Noise
Boom at low Frequencies (< 100Hz)

➔ Dynamics up to 100Hz

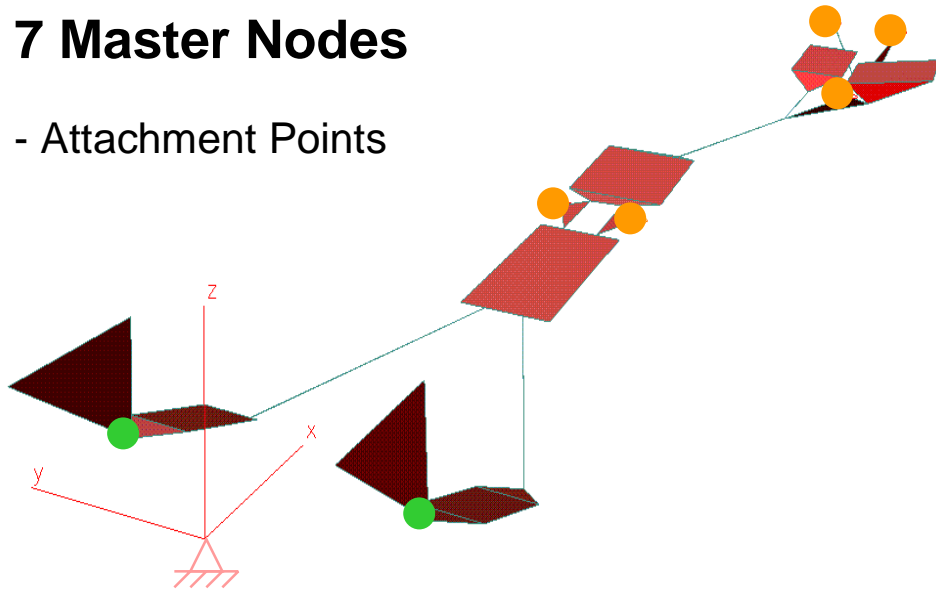
Coupling with the MBS by
Rigid Joint at Exhaust Manifold and
Elastic Force Elements

FEM-Model Reduction

Alternatives with Different Number of Master Nodes

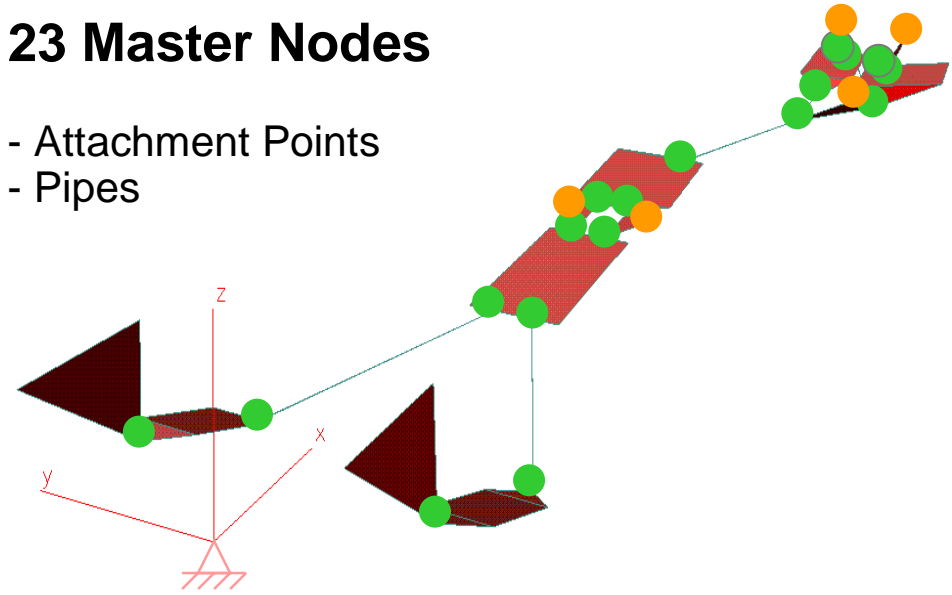
7 Master Nodes

- Attachment Points



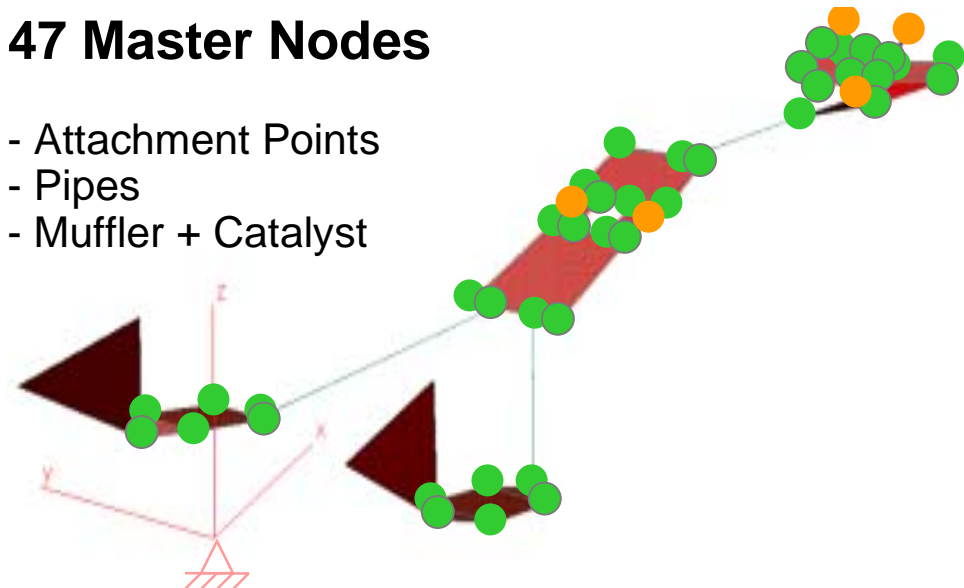
23 Master Nodes

- Attachment Points
- Pipes



47 Master Nodes

- Attachment Points
- Pipes
- Muffler + Catalyst



Master Nodes

● Coupling Nodes

● Additional Master Nodes

Model Quality of the FEM-Reduction Constrained Exhaust System without Mounts

		FEM	Static Reduction			Dynamic Reduction EigenFrequencies to 300Hz		
Nodes		9619	7	23	47	7	23	47
Number of Eigen Frequencies	1	1,6	1,6	1,6	1,6	1,6	1,6	1,6
	2	3,5	3,5	3,5	3,5	3,5	3,5	3,5
	3	9,7	9,8	9,7	9,7	9,7	9,7	9,7
	4	15,8	16,1	15,8	15,8	15,8	15,8	15,8
	5	24,0	27,6	24,1	24,0	24,0	24,0	24,0
	6	28,8	31,8	28,8	28,8	28,8	28,8	28,8
	7	32,1	36,0	32,2	32,1	32,1	32,1	32,1
	8	41,7	49,0	41,9	41,7	41,8	41,7	41,7
	9	58,0	70,5	58,5	58,2	58,1	58,1	58,1
	10	74,6	99,8	75,1	75,0	74,8	74,7	74,7
	11	85,3	135,4	87,3	85,7	85,5	85,4	85,4
	12	87,3	151,3	88,3	87,5	87,4	87,4	87,4
	13	110,4	172,4	112,6	110,9	110,7	110,5	110,5
	14	116,2	242,9	126,5	116,9	116,5	116,3	116,3
	15	129,1	261,3	133,1	130,4	129,6	129,3	129,3
	16	148,4	340,5	153,7	149,6	150,5	148,8	148,7

Relative Error

5% < f

2% < f < 5%

f < 2%

Eigenfrequencies of FEM-Model and Reduced Models evaluated with NASTRAN

- ➡ **Static: Model quality dependant on number and distribution of master nodes**
- Dynamic: Model quality independant of the master nodes**

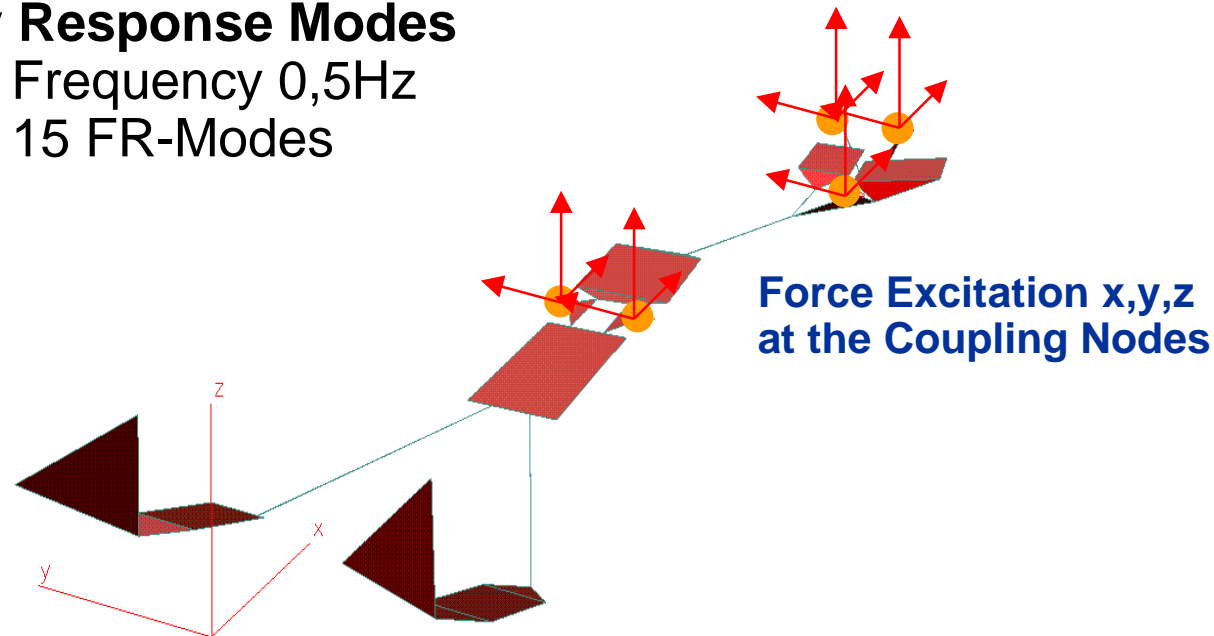
FEMBS Preprocessing

Eigenmodes

- $f_e < 155\text{Hz}$

Frequency Response Modes

- Excitation Frequency 0,5Hz
- Maximum 15 FR-Modes



- Maximum Frequency of the Flexible Body

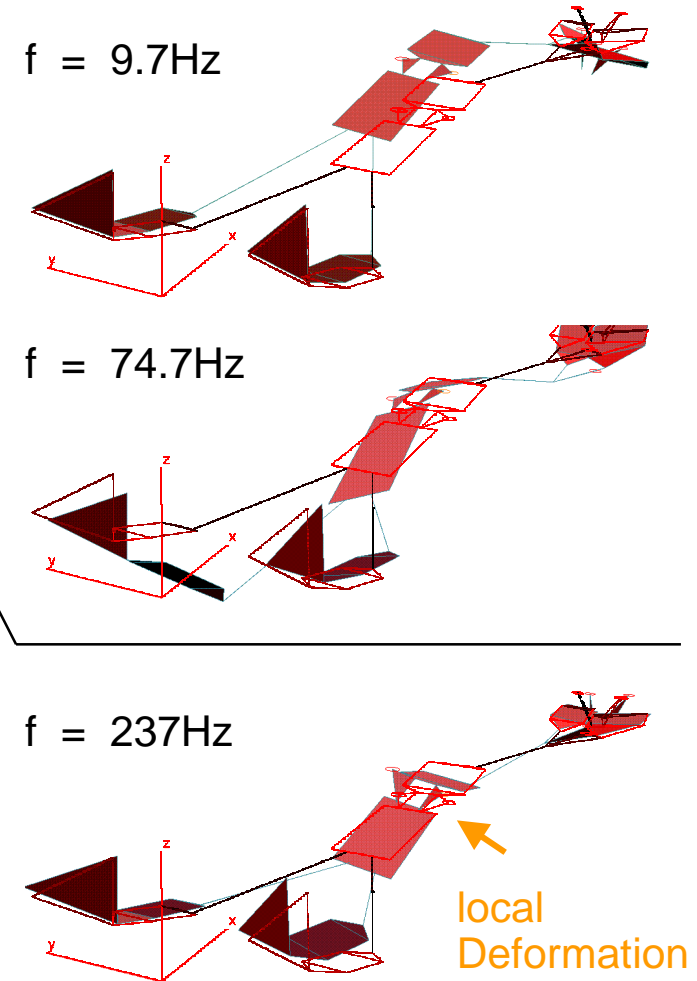
$f_r < 500\text{ Hz}$ -> 5 FR-Modes

$f_r < 1000\text{ Hz}$ -> 11 FR-Modes

$f_r < 1800\text{ Hz}$ -> 15 FR-Modes

Influence of Frequency Response Modes Constrained Exhaust System without Mounts

		FEM	MBS - Dynamic Reduction (47 Nodes)				
FEMBS Modes			fe < 155Hz	fe < 155Hz fr < 500z	fe < 155Hz fr < 1000z	fe < 155Hz fr < 1800z	
Number of Eigen Frequencies	1	1,6	1,6	1,6	1,6	1,6	Eigenmodes
	2	3,5	3,5	3,5	3,5	3,5	
	3	9,7	9,7	9,7	9,7	9,7	
	⋮	⋮	⋮	⋮	⋮	⋮	
	14	116,2	116,3	116,3	116,3	116,3	
	15	129,1	129,3	129,3	129,3	129,3	
	16	148,4	148,7	148,7	148,7	148,7	
	Frequency Response Modes	17	157,7		215,5	215,5	215,5
		18	171,0		236,9	236,9	236,9
		19	192,4		345,8	345,8	345,8
		20	217,2		412,8	412,8	412,8
		21	260,8		479,6	479,6	479,6
		22	264,3			579,1	579,1
		23	272,7			⋮	612,1
24		288,6			960,7	⋮	
25		304,7				1789,8	
⋮		⋮					

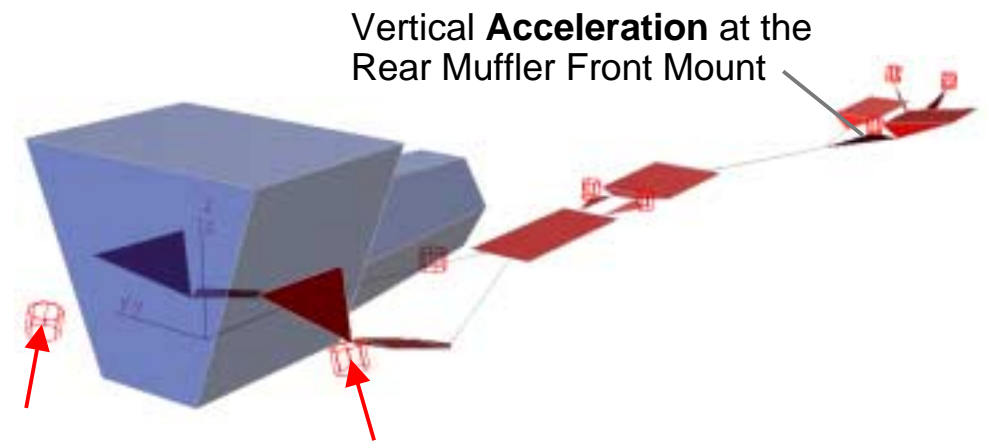
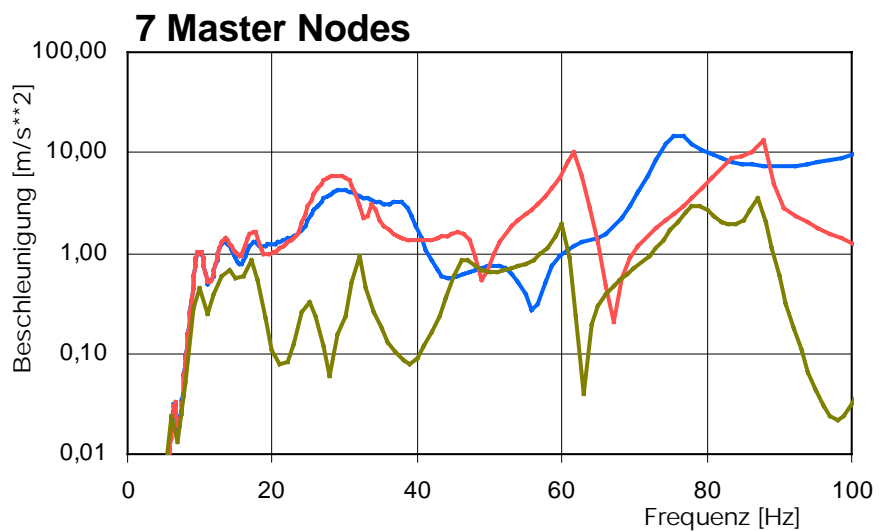


Eigenfrequencies of the Flexible Body in Comparison to FEM

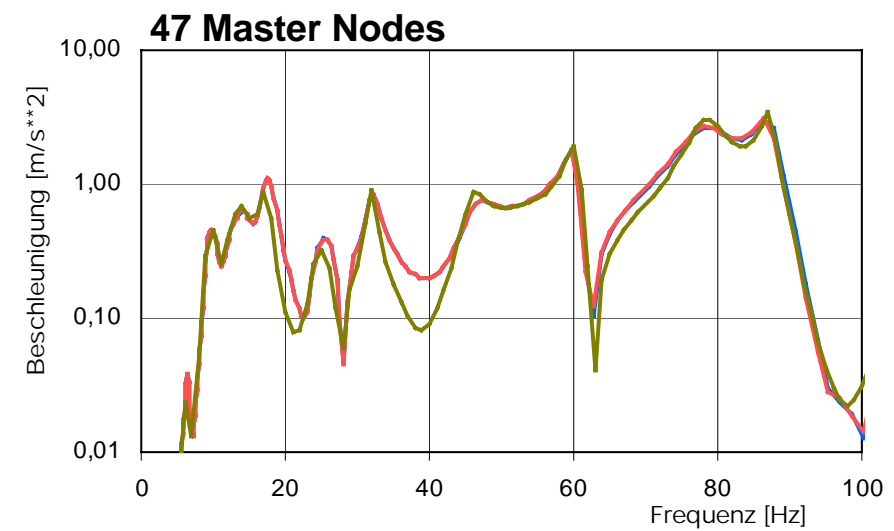
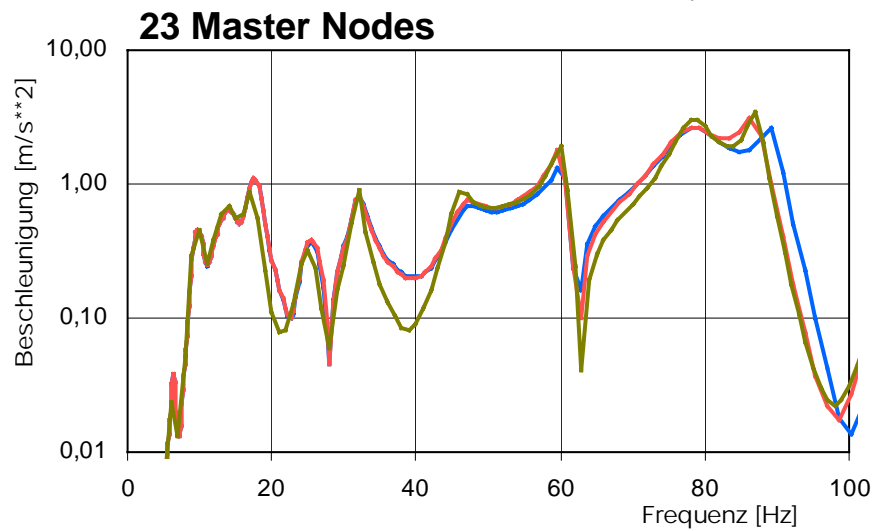
➡ **Frequency Response Modes** create **High Frequency Eigenmodes** as correction for missing FEM-Modes to describe **Local Deformations** at the coupling nodes.

Validation of the Coupled MBS

Comparison of the Reduction Alternatives



Force Excitation at Engine Mounts in Phase 200N

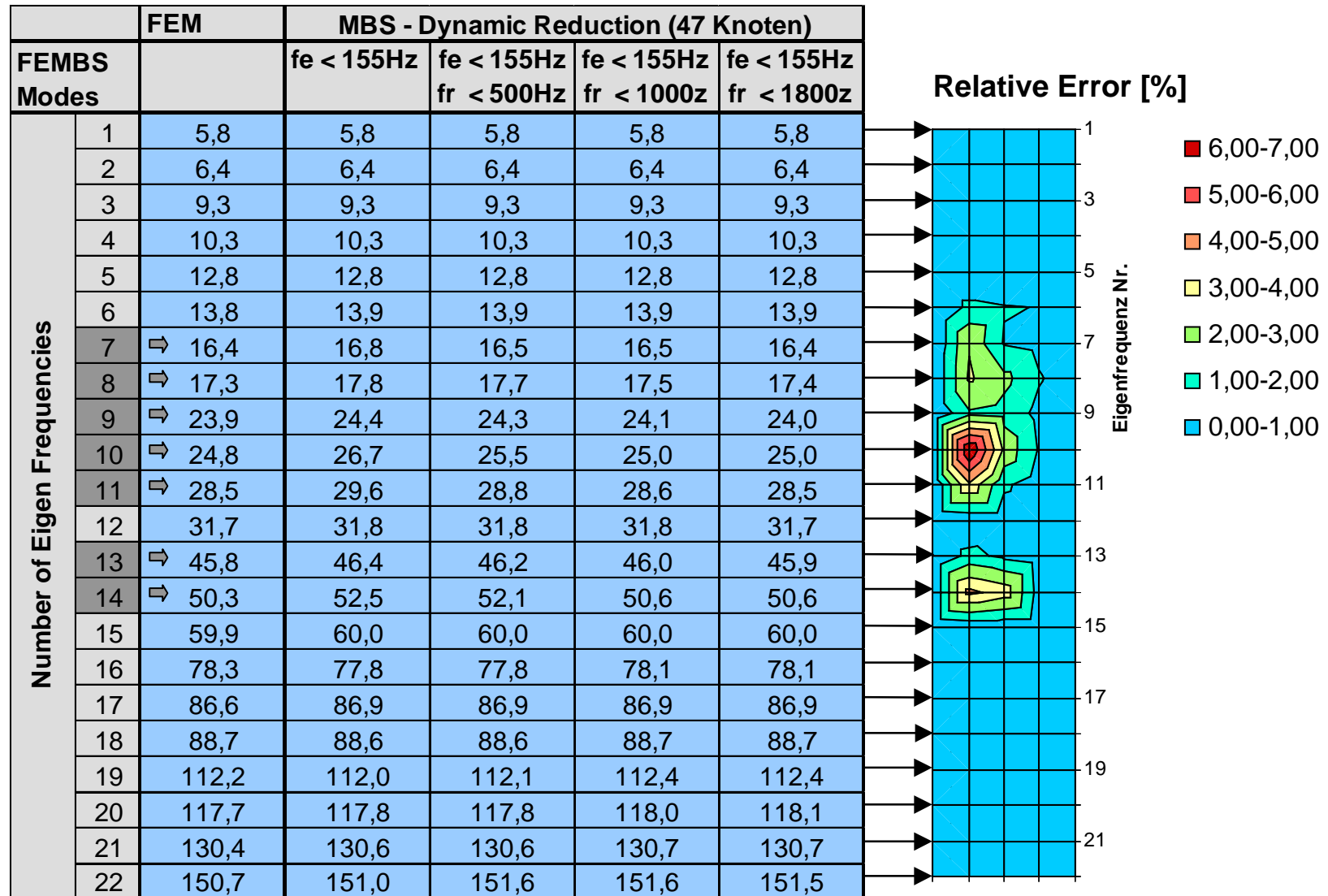


Linear Response Analysis Comparing Reduction Alternatives (without Frequency Response Modes)

— FEM — Static Reduction — Dynamic Reduction

Validation of the Coupled MBS

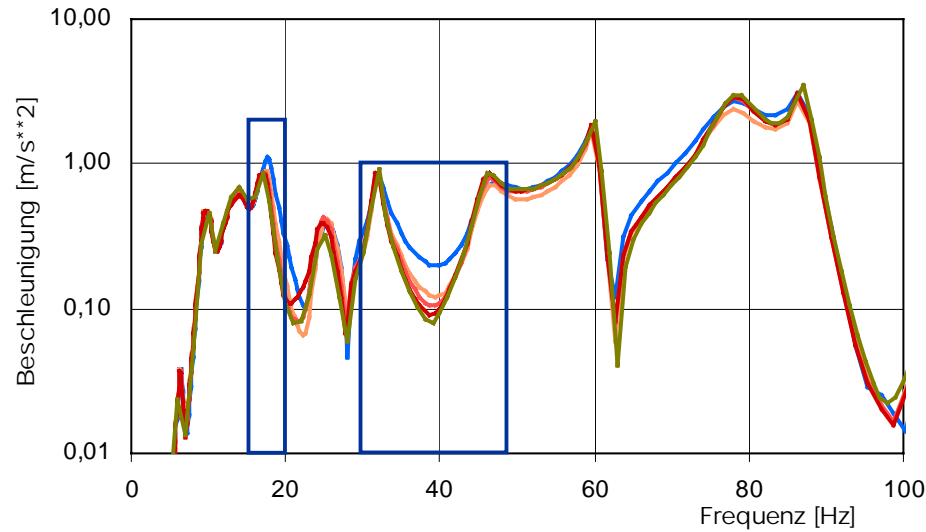
Influence of Frequency Response Modes



Eigenfrequencies of the Coupled MBS in Comparison to FEM

Validation of the Coupled MBS

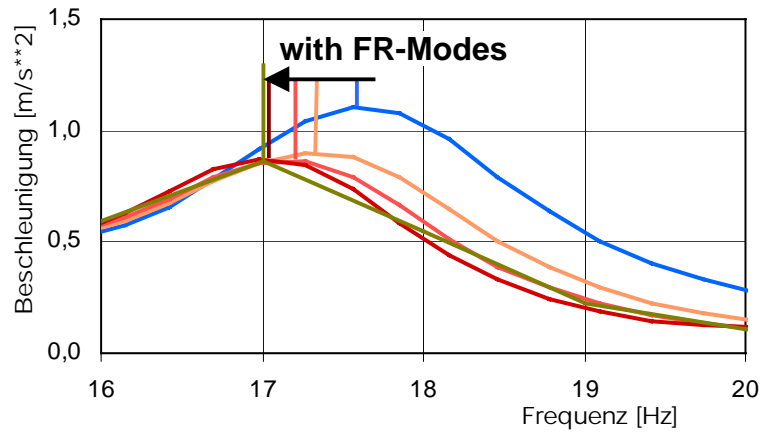
Influence of Frequency Response Modes



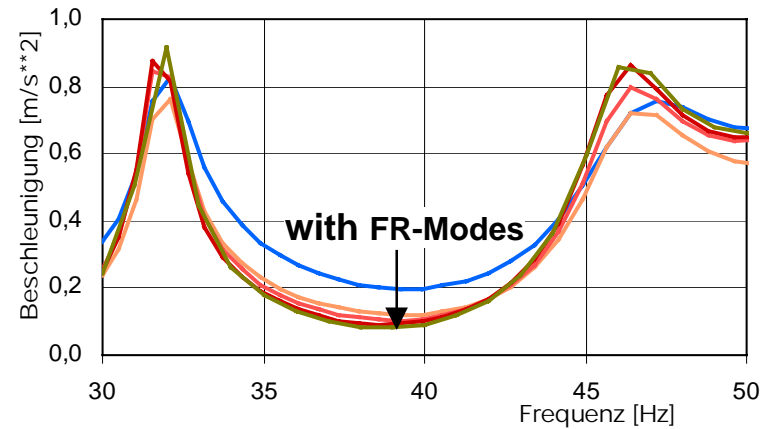
Linear Response Analysis

Vertical Acceleration at Rear Muffler Front Mount due to Force Excitation in Phase at Engine Mounts

- Without FR-Modes
- 5 FR-Modes up to 500Hz
- 11 FR-Modes up to 1000Hz
- 15 FR-Modes up to 1800Hz
- FEM



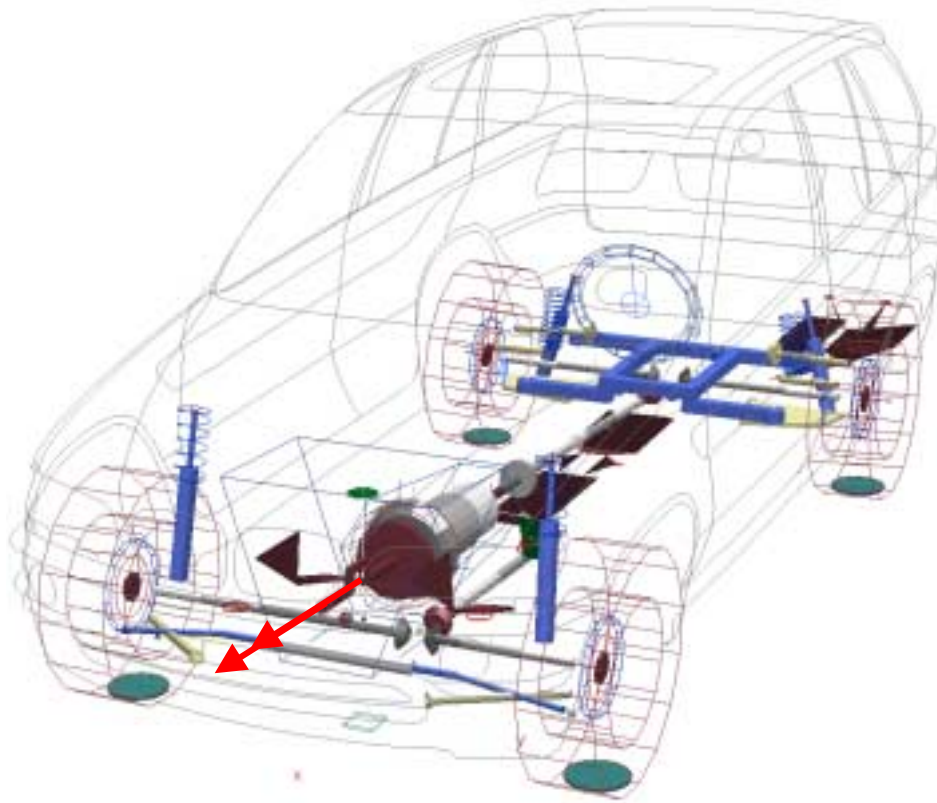
Improvement of Resonant Frequency



Improvement of Amplitude

MBS Full Vehicle Model

Principle Analysis of Engine Idling Comfort

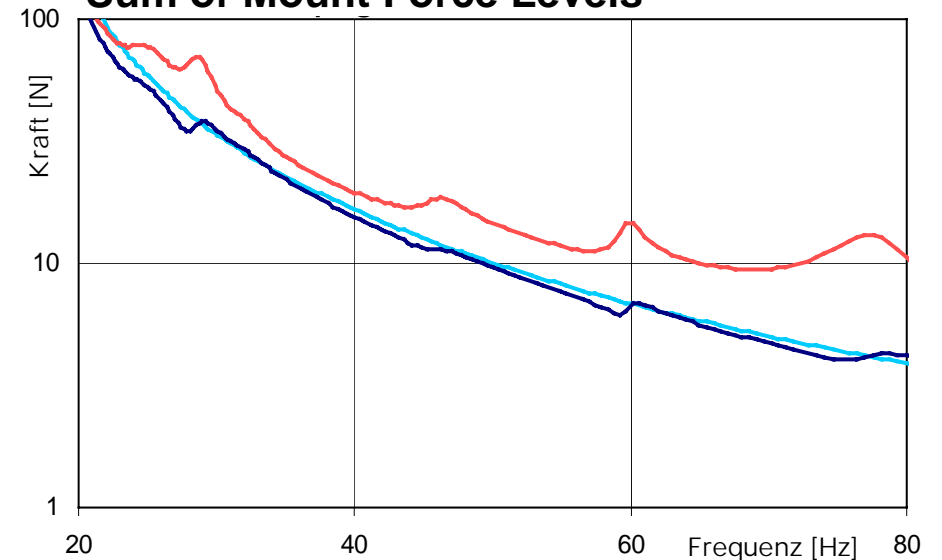


Linear Response Analysis

Mount Forces (Engine, Gearbox, Exhaust) due to **Excitation** by Alternating Torque 30Nm, 20-80Hz about the Engine Longitudinal Axis

- Engine-/Gearbox (Model without Exhaust)
- Engine-/Gearbox (Model with Exhaust)
- Engine-/Gearbox and Muffler Mounts

Sum of Mount Force Levels



Full Vehicle Model for Ride Analysis

- | | |
|--------------------------|-------------------|
| - Chassis | 84 Bodies |
| - Drive Train | 110 Joint DoF |
| - Engine Mounting System | 38 Constraints |
| - Exhaust System | 67 Additional ODE |

Summary

- The **Advantages** of the **Dynamic Reduction** are
 - **Simplification** of the FEM-reduction and
 - **Accurate Modelling** independent of the number of master nodes
- The generation of a flexible body with **‘too few Master Nodes’** results in incorrect frequency responses in the coupled MBS.

Possible cause

- FEM-model reduction in **Constrained Condition** leads to inaccurate **Mass Distribution**
- Possible improvements from INTEC
 - **Implement** the dynamic reduction in **FEMBS**
 - Investigate the problems of **‘too few Master Nodes’**