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Topology optimization to maximize the dynamic input stiffness of front axle coach structure

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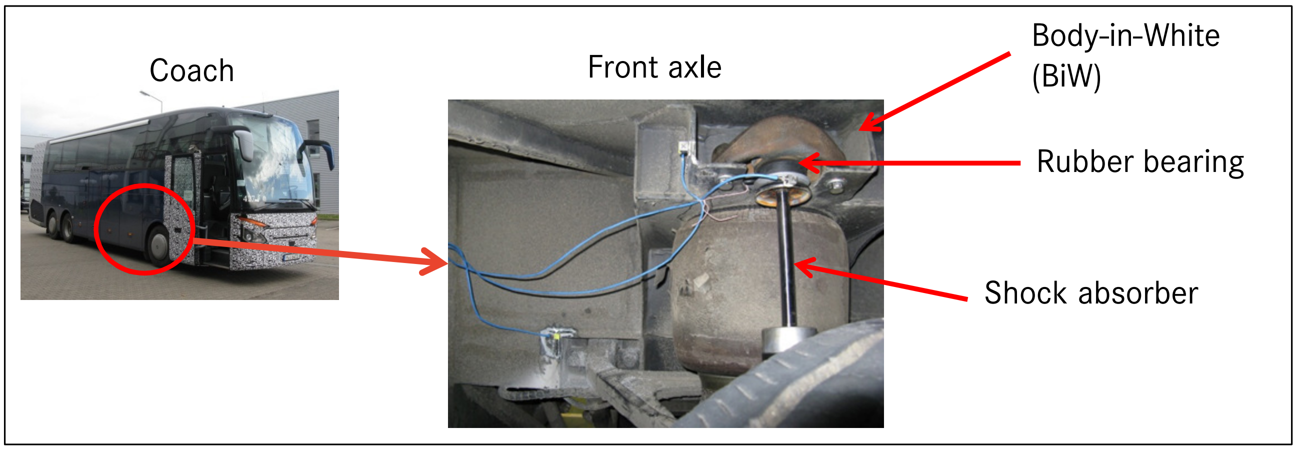
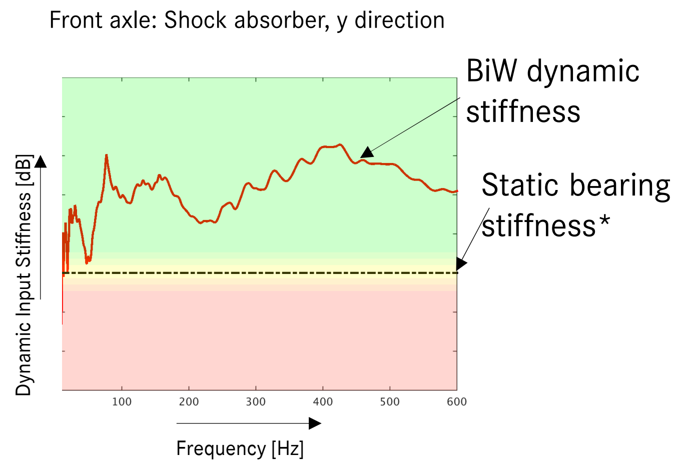
Introduction

In order to increase commonality of parts, it was decided to have a common front axle for all kinds of buses like city buses, inter-urban buses and coaches. Many factors like weight, dynamic stiffness, strength, durability, crash, roll-over and drive dynamics had to be investigated.

The work depicted in this presentation concentrates on maximizing the dynamic input stiffness of the front axle coach structure.

What is dynamic input stiffness?





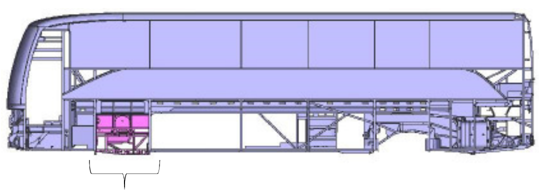
For isolation of vibrations between BiW and axle; $K_{BiW} > K_{Rubber}$, where K is the dynamic stiffness in the region of 0 Hz- 600 Hz.

* The dynamic stiffness values of rubber bearings over a large frequency range are always not available. Hence only the static stiffness of the rubber bearing is considered.

Adaptation of basic design using a conventional approach

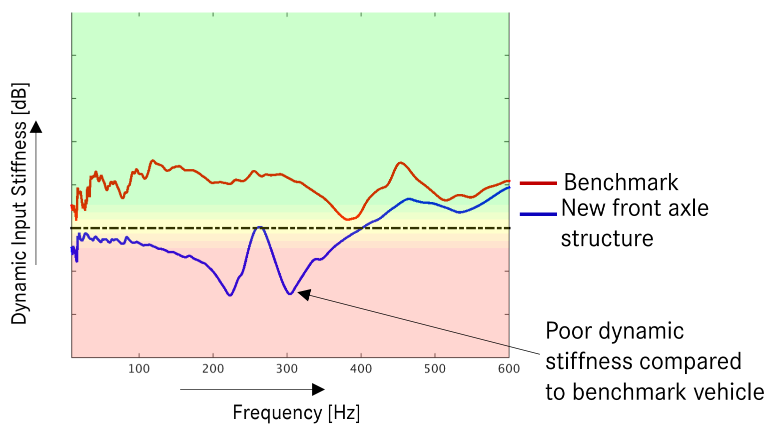
- Adaptation of the basic design with a conventional engineering approach: Based on the results of the dynamic input stiffness analyses, design changes were manually carried out. An optimum usage of design space cannot be guaranteed with this method.

Here the topology optimization was not considered.



Front axle body-in-white with design changes

Front axle: front lower control arm, z direction



Force direction	**X**	**Y**	**Z**
Air spring (left/right)	🟡	🟢	🔴
Shock absorber (left/right)	🟡	🟡	🟡
Front lower control arm (left/right)	🟢	🟢	🔴
Rear lower control arm (left/right)	🟡	🟢	🟡
Front upper control arm (left/right)	🟡	🟡	🟡
Rear upper control arm (left/right)	🟡	🟡	🟡
Stabilizer (left/right)	🟢	🟡	🟢
Steering gear box	🟢	🔴	🔴

🟢 Design criteria fulfilled
🟡 Test worthy
🔴 Design criteria not fulfilled

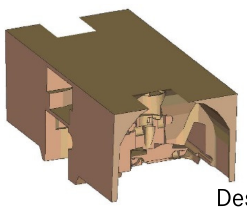
The conventional approach led to a complex design which was not fulfilling the required design criteria.

Hence a new design approach to solve this Problem was required!

New design approach using topology optimization in combination with dynamic input stiffness analysis

- Topology optimization: Process flow and key challenges

Definition of design space

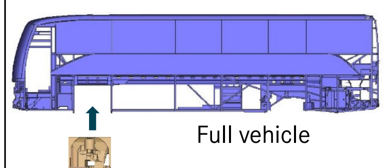


Design space for front axle structure

Key challenges:

- Axle kinematics
- Tools used during assembly
- Replacement of parts during maintenance
- Provision for flow of outgoing air from bus cabin

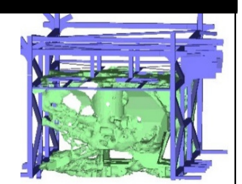
Integration of design space in vehicle model



Key challenges:

- Size of mesh
- Choice of element types
- Coupling nodes

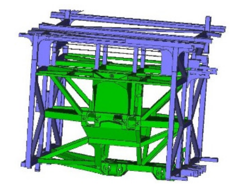
Topology optimization



Key challenges:

- 45 design constraints
- Realistic limits for the design constraints
- Optimization with frequency response
- Substructure technique and choice of static/ dynamic condensation
- Computation time

Design

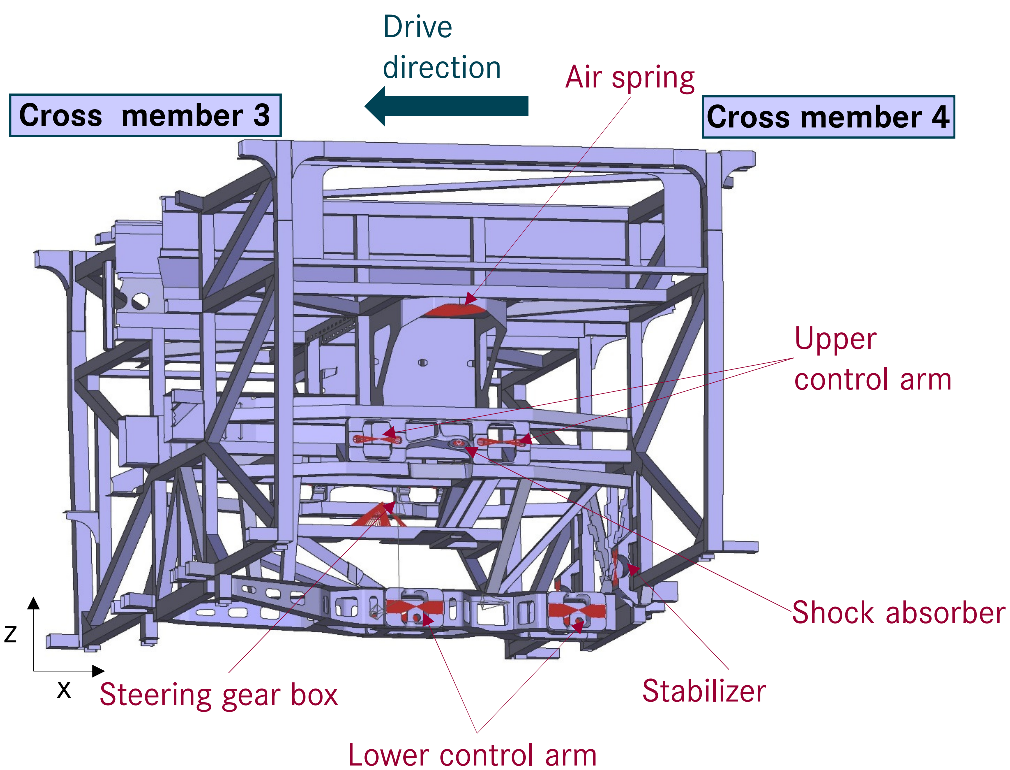


Key challenges:

- Interpretation of optimization results
- Derivation of a space frame structure corresponding to the common bus design philosophy.

Key challenges in topology optimization

- Displacement design constraints for optimization



45 displacement design constraints in a frequency range of 0 - 600 Hz:

- Air springs (left/right in x, y, z)
- Shock absorber (left/right in x, y, z)
- Front lower control arm (left/right in x, y, z)
- Rear lower control arm (left/right in x, y, z)
- Front upper control arm (left/right in x, y, z)
- Rear upper control arm (left/right in x, y, z)
- Stabilizer (left/right in x, y, z)
- Steering gear box (in x, y, z)

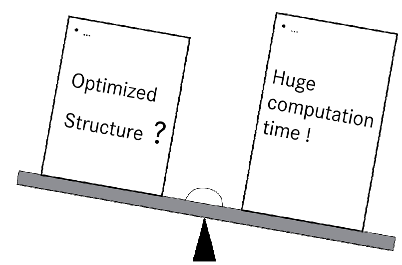
e.g.: with a sampling frequency of 5 Hz, each design constraint consists of 120 frequency points.
Hence we have 45 x 120 = 5400 conditions.

Key challenges in topology optimization

- The full vehicle model with the design space elements has more than 5 million nodes.
- Frequency response and modal analysis had to be carried out till 600 Hz.
- The design constraints (displacement) considered at some locations were too stringent to be fulfilled.
- The sampling frequency (1 Hz) considered was too small.

All the above led to huge computation time (> 7 days) and did not lead to convergence of results. The high performance machine used for the above analysis had the following configuration:

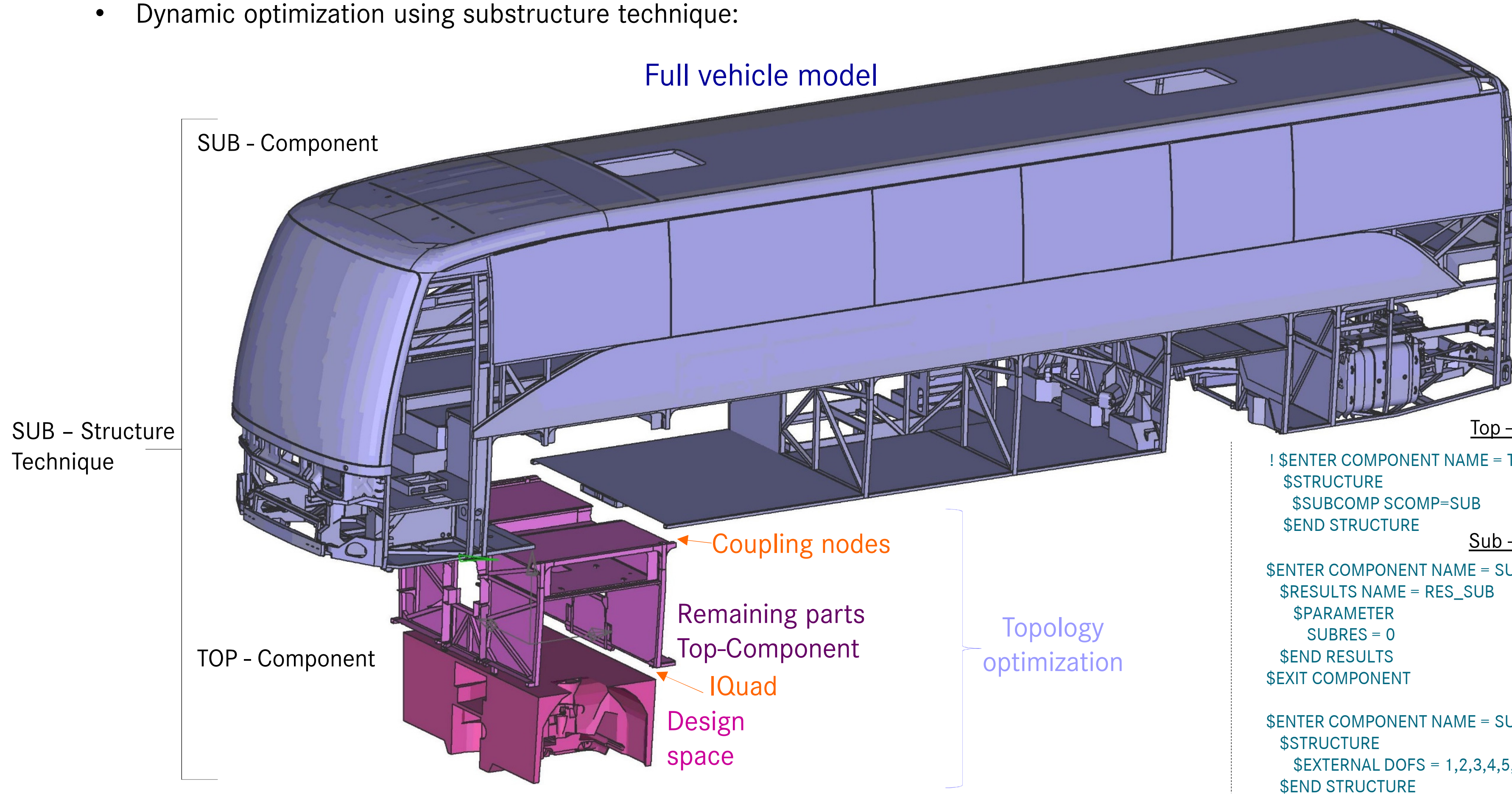
2 X HP DL380 Gen9 Intel Xeon E5-2667v3 (3.2 GHz/8- core/20 MB/135 W) Processor Kit
4X HP 1.6 TB HH/HL Value Endurance (VE) PCIe Workload Accelerator
NVIDIA Tesla K40C 12 GB Computational Accelerator



Hence simplification of the model for optimization is necessary!

Simplification of the model for optimization

- Dynamic optimization using substructure technique:



Top - Component

```
! $ENTER COMPONENT NAME = TOP     DOFTYPE = DISP
  $STRUCTURE
    $SUBCOMP SCOMP=SUB
  $END STRUCTURE
```

Sub - Component

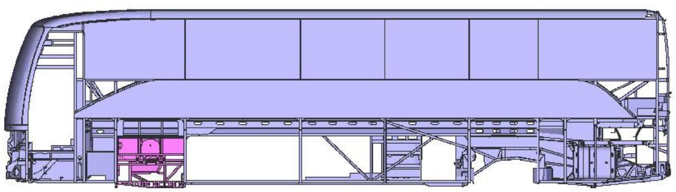
```
$ENTER COMPONENT NAME = SUB     DOFTYPE = DISP
  $RESULTS NAME = RES_SUB
    $PARAMETER
      SUBRES = 0
  $END RESULTS
$EXIT COMPONENT

$ENTER COMPONENT NAME = SUB     DOFTYPE = DISP
  $STRUCTURE
    $EXTERNAL DOFS = 1,2,3,4,5,6      ! Coupling nodes
  $END STRUCTURE
```

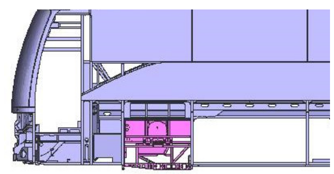
Simplification of the model for optimization

- Reduced model:

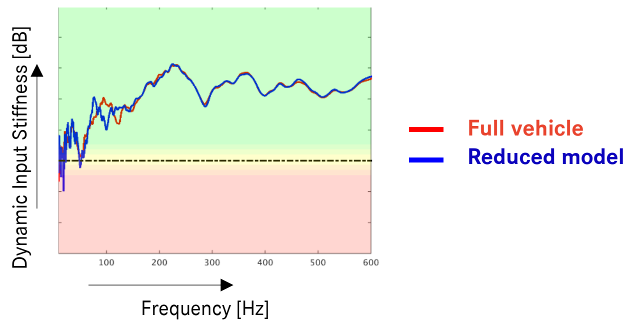
Full vehicle



Reduced model



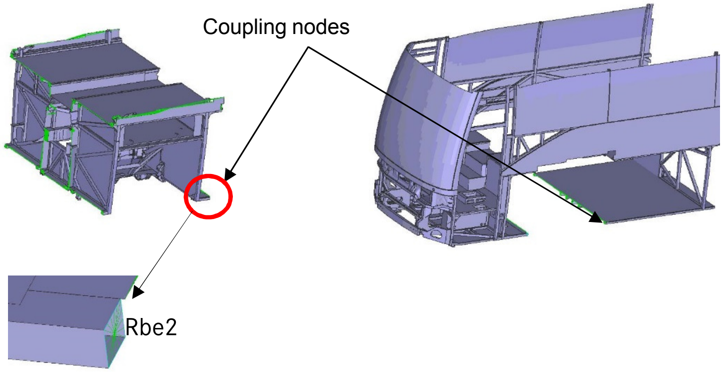
Front axle: left shock absorber, y direction **



- Minimizing the number of coupling nodes between sub and top component:

Top Component

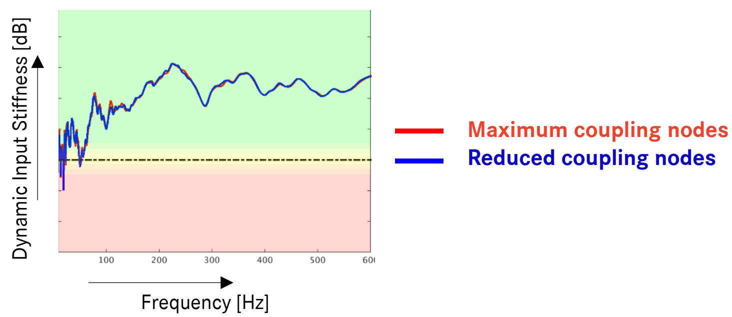
Sub Component



Coupling nodes belonging to major load carrying structure was considered.

Coupling nodes reduced by more than 90%!

Front axle: left shock absorber, y direction **

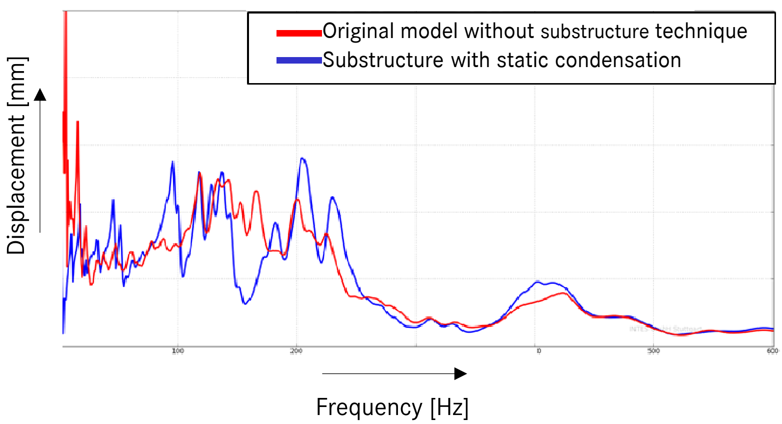


**The diagram of dynamic stiffness vs frequency of left shock absorber in y direction is shown as an example. All the other 44 interface points show a similar tendency.

Simplification of the model for optimization

- Static condensation:

Right air spring z direction



Super-positioning the static condensation results of the substructure on the top component shows a similar tendency as that of the original model without substructure technique.

Dynamic condensation was ignored in the initial calculations.
This means that the modes of the subcomponent are not computed and therefore do not contribute to the analysis

- Increase the sampling frequency to 10 Hz and carry out frequency response analysis only till 250 Hz.

It was decided to increase the sampling frequency rate from 1 Hz to 10 Hz. The error due to large spacing in the sampling frequencies were ignored.

Dynamic topology optimization: Procedure

1.Step: Static

Statically determined model

Unit excitation at the interface points

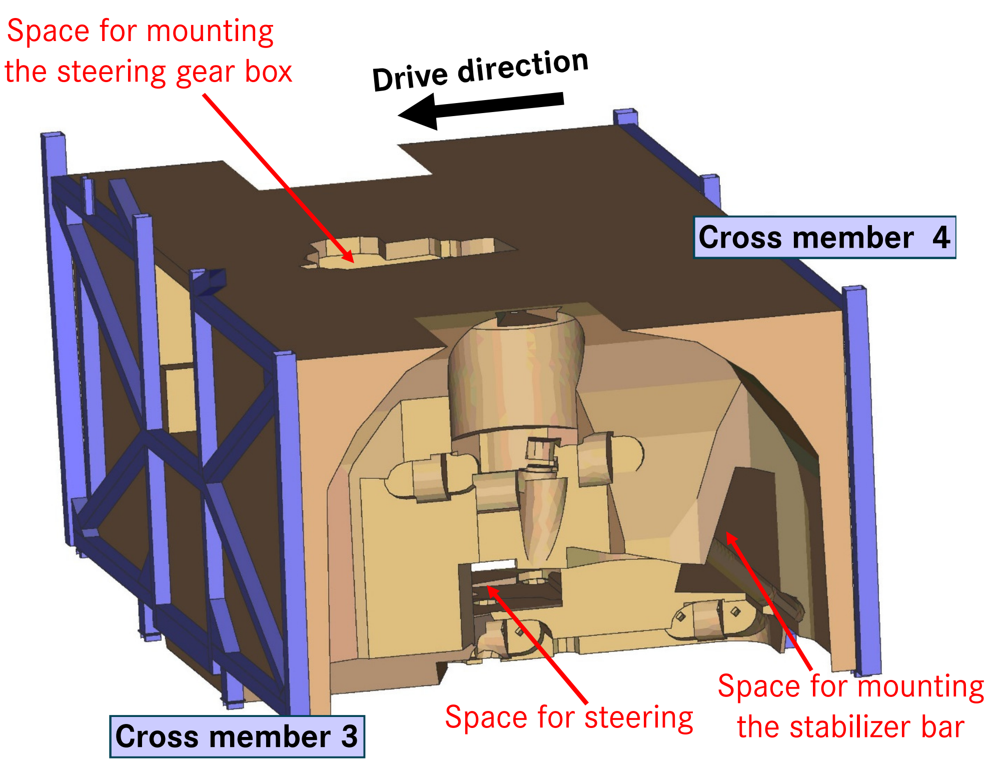
2. Step: Dynamic Stiffness Analysis

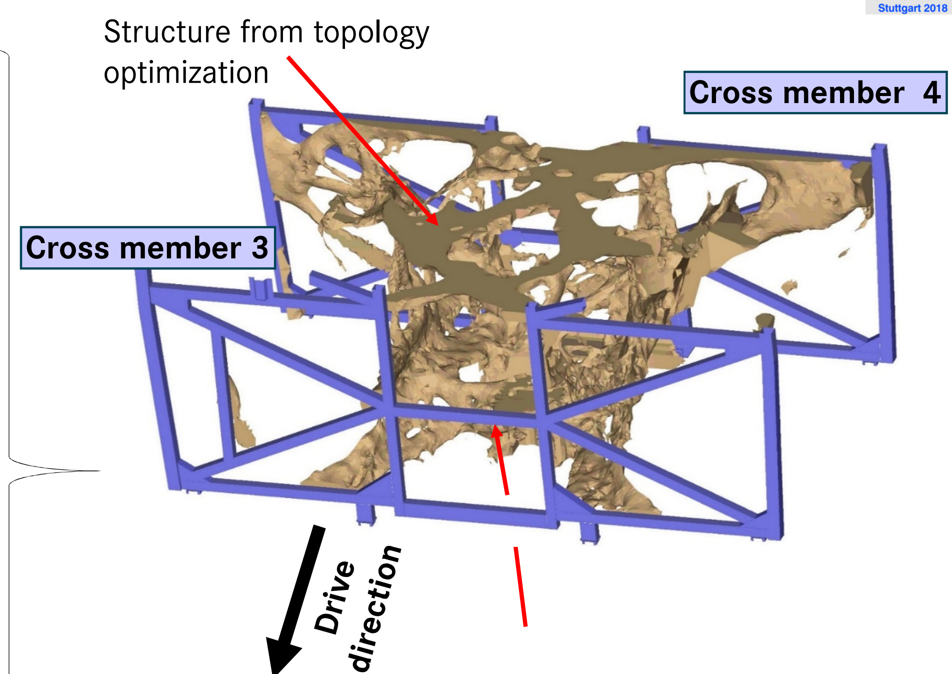
Eigen-frequency calculation till 250 Hz
Frequency response analysis till 250 Hz with sampling rate of 10 Hz

3. Step: Topology Optimization

Target:	maximum static stiffness
Design constraints:	Weight < 1 t
	Displacement limits at all excitation points for 10, 20, 30, … , 250 Hz

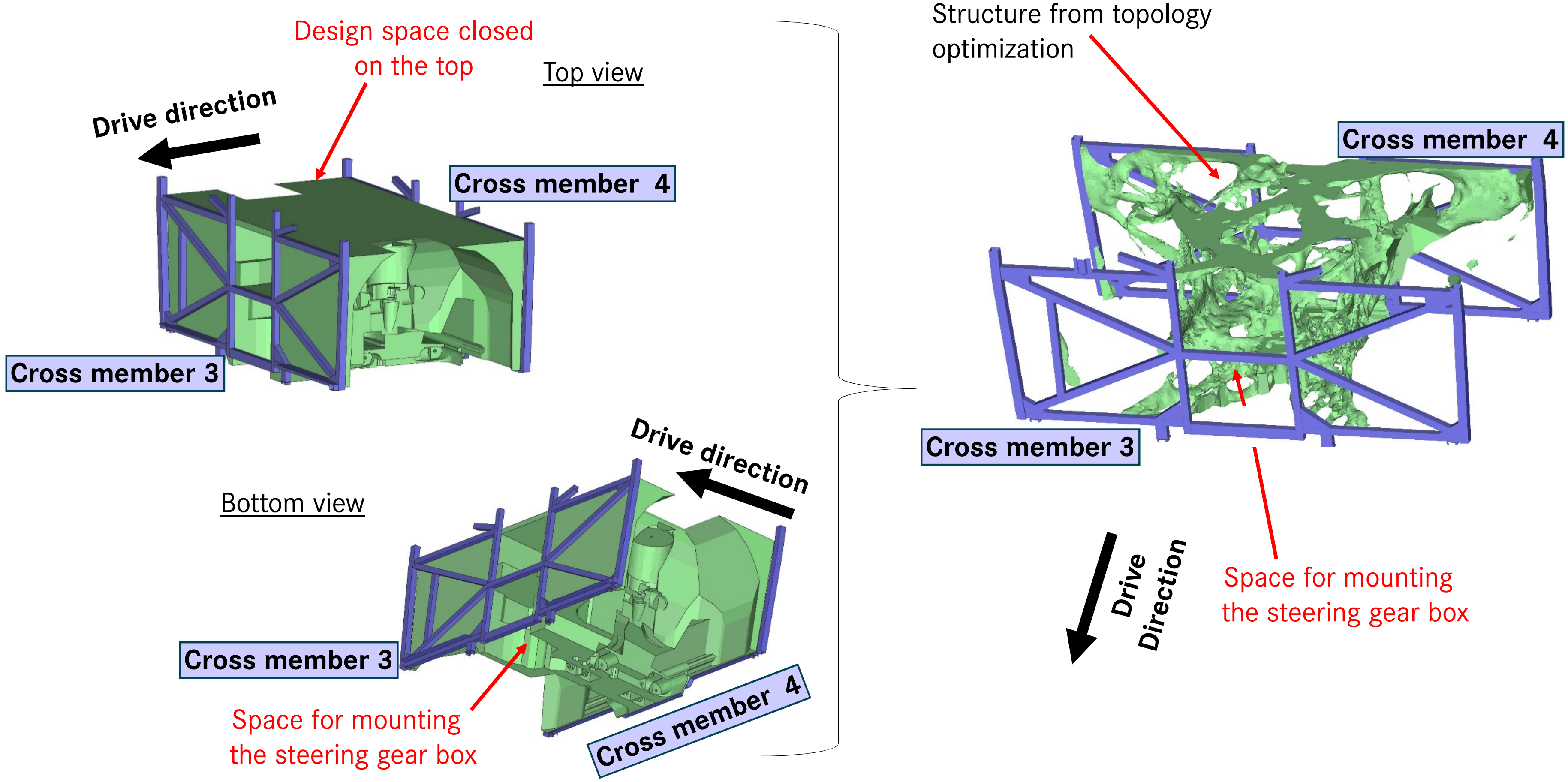
Dynamic topology optimization from initial design space



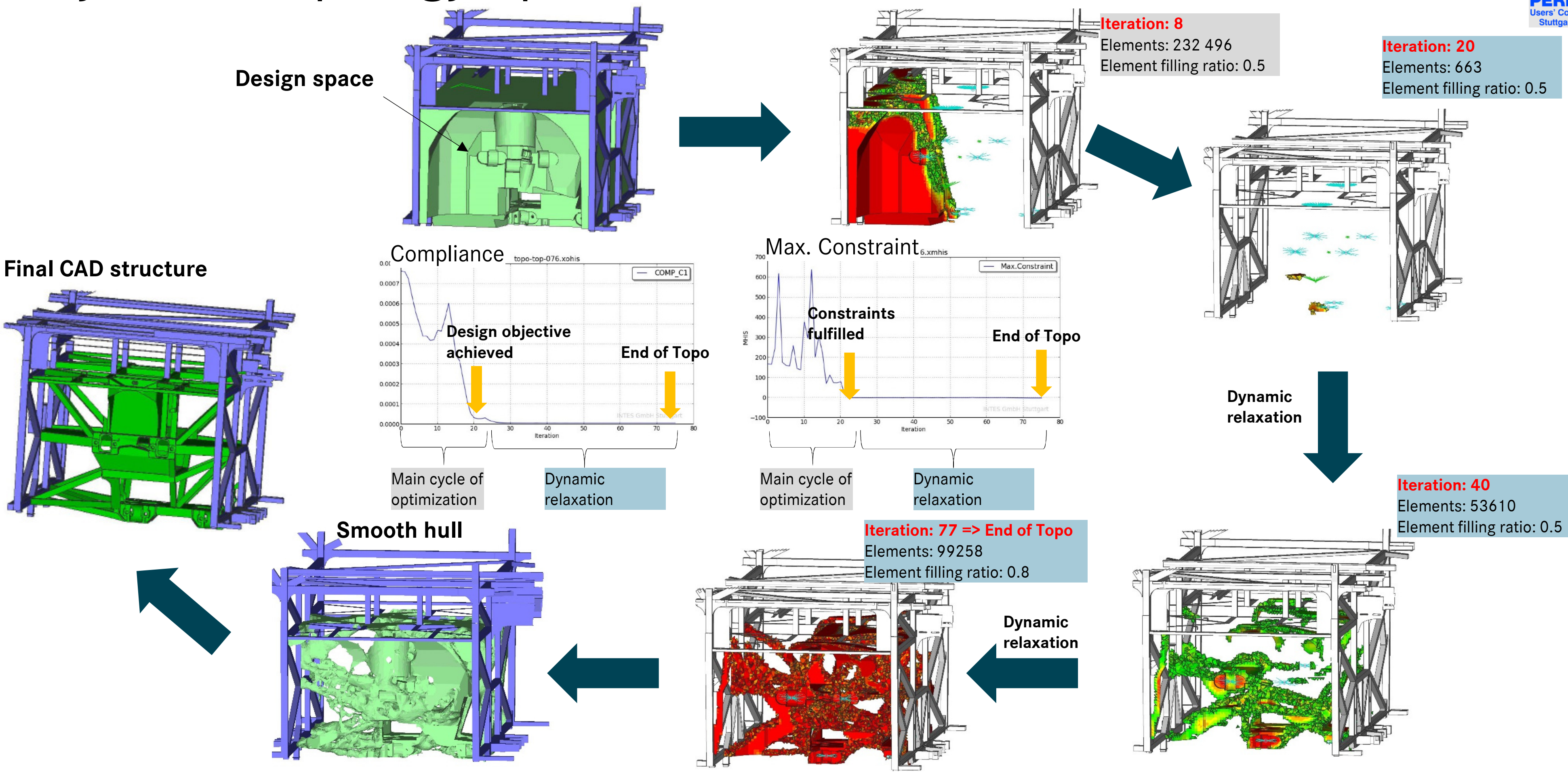


New idea for mounting the steering gear box from underneath the axle structure ensuring easy handling and reduction of mounting time in assembly

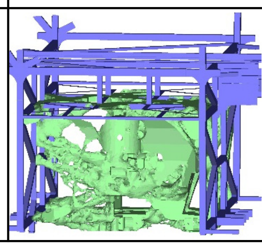
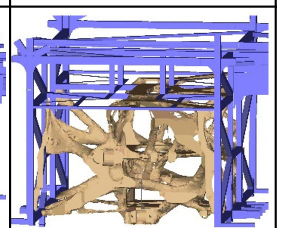
Dynamic topology optimization from new design space



Dynamic topology optimization: Process flow



Dynamic condensation, design constraints and computation time

Condensation type in substructure:	static	dynamic
Optimization limit	250 Hz	600 Hz
Sampling frequency	10 Hz	5 Hz
Design constraint	stringent	relaxed (33% reduction in dynamic stiffness)
Computation time* *same hardware as mentioned in page 6	85 hrs.	106 hrs.
Structure after optimization		

<u>Transfer of modal information from sub to top component for dynamic condensation</u>

$EXTERNAL MODE DOFS = 9 DOFTYPE=DISP

If the dynamic condensation is taken into account, the stringent design constraints have to be relaxed to get the results in the desired time leading to convergence.

Final result and future work

Initial design

Force direction	X	Y	Z
Air spring (left/right)	🟡	🟢	🔴
Shock absorber (left/right)	🟡	🟡	🟡
Front lower control arm (left/right)	🟢	🟢	🔴
Rear lower control arm (left/right)	🟡	🟢	🟡
Front upper control arm (left/right)	🟡	🟡	🟡
Rear upper control arm (left/right)	🟡	🟡	🟡
Stabilizer (left/right)	🟢	🟡	🟢
Steering gear box	🟢	🔴	🔴

- 🟢 Design criteria fulfilled
- 🟡 Test worthy
- 🔴 Design criteria not fulfilled

Design after topology optimization

Force direction	X	Y	Z
Air spring (left/right)	🟡	🟢	🟡
Shock absorber (left/right)	🟡	🟢	🟡
Front lower control arm (left/right)	🟢	🟢	🟡
Rear lower control arm (left/right)	🟢	🟢	🟢
Front upper control arm (left/right)	🟢	🟢	🟢
Rear upper control arm (left/right)	🟢	🟢	🟢
Stabilizer (left/right)	🟢	🟢	🟢
Steering gear box	🟢	🟢	🟢

- For the first time at Daimler Buses the dynamic input stiffness analysis has been coupled with topology optimization resulting in a successful design of front axle coach structure with maximum possible dynamic input stiffness.
- Simplifications carried out in this work are problem specific, requires cross checking of the results and depends on the judgement of the user.

It is desired to include more design constraints related to other disciplines as well. This may include simplified static load cases from a crash, rough road or other NVH analyses.

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