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; $K_{\text{BiW}} > K_{\text{Rubber}}$, 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.

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**
- 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:
Simplification of the model for optimization

- Reduced model:

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

  **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.**

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Simplification of the model for optimization

• Static condensation:

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

Structure from topology optimization

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

Design space closed on the top

Top view

Structure from topology optimization

Cross member 3

Cross member 4

Space for mounting the steering gear box

Bottom view

Drive direction

Cross member 3

Cross member 4

Drive direction

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Dynamic topology optimization: Process flow

Design space

Final CAD structure

Smooth hull

Compliance

Max. Constraint

Design objective achieved

Constraints fulfilled

End of Topo

End of Topo

Main cycle of optimization

Dynamic relaxation

Main cycle of optimization

Dynamic relaxation

Dynamic relaxation

Dynamic relaxation

Iteration: 8
Elements: 232,496
Element filling ratio: 0.5

Iteration: 20
Elements: 663
Element filling ratio: 0.5

Iteration: 40
Elements: 53610
Element filling ratio: 0.5

Iteration: 77 => End of Topo
Elements: 99,258
Element filling ratio: 0.8

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Dynamic condensation, design constraints and computation time

<table>
<thead>
<tr>
<th>Condensation type in substructure:</th>
<th>static</th>
<th>dynamic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimization limit</td>
<td>250 Hz</td>
<td>600 Hz</td>
</tr>
<tr>
<td>Sampling frequency</td>
<td>10 Hz</td>
<td>5 Hz</td>
</tr>
<tr>
<td>Design constraint</td>
<td>stringent</td>
<td>relaxed (33% reduction in dynamic stiffness)</td>
</tr>
<tr>
<td>Computation time*</td>
<td>85 hrs.</td>
<td>106 hrs.</td>
</tr>
<tr>
<td>Structure after optimization</td>
<td><img src="image1.png" alt="Structure before optimization" /></td>
<td><img src="image2.png" alt="Structure after optimization" /></td>
</tr>
</tbody>
</table>

*same hardware as mentioned in page 6

Transfer of modal information from sub to top component for dynamic condensation

$\text{SEXTERNAL MODE DOFS = 9 DOFTYPE=DISP}$

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.
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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