dent that there is a clearly defined optimum, i.e. the set of material parameters that is best suited to describe the real mechanical behaviour of the correspondent test person's breast. Looking first at the variations in Poisson's ratio, there is a decrease towards higher values, meaning less compressibility. Thus the commonly used assumption of biological soft tissues to be incompressible or at least nearly incompressible can be confirmed by these findings. Since this is true for all tested models, in future work it seems no more necessary to deal with compressible material models at all, resulting in the reduction of unknown material parameters. Taking a further look at the material stiffness (figure 6), the Young's modulus, a clearly defined optimal position can be found. The model behaviour is described by a shallow slope when coming from high Young's moduli and a relatively steep increase when the material parameters become too soft. For all optimizations performed in the present study, defined global optima could be found. The individual optima for the eight test persons were found within the range of 0.494 kPa to 0.852 kPa for the Young's modulus. Hence, between the different test persons relatively high differences in soft tissue stiffness of 72.5% could be investigated, underlining the need for patient individual simulations.

**Discussion**

The advantage of the whole workflow presented here is the non-invasive character as a combination of volume imaging (MRI) and 3-D surface scanning (Laser triangulation) and the involvement of the computer for the actual simulation. No tissue samples of the patient's soft tissue have to be harvested what is especially a critical issue if the mechanical information derived from these specimens should be used in operation planning, because this would mean additional intervention for the patient. Furthermore, the expensive and cumbersome experimental testing can be circunvented.

![Diagram](image)

The high variation in stiffness of almost a factor of 2 between the softest and the hardest optimal material parameter set found in this study shows the distinct need for the patient individual assessment of soft tissue material parameters. Thus patient specific simulations seem inevitable. Hence, the advantages of this non-invasive and fully computerized approach become obvious.

**Outlook**

The workflow presented in this publication may in the future be used for the material parameter assessment of hyper-elastic parameters that are suited for patient individual modelling of the constitutive behaviour of the female breast soft tissue. These data may subsequently be utilized for numerical simulations and planning of complex surgical interventions in plastic surgery.

The presented approach is not limited to its application in plastic surgery of the female breast. Other uses of this procedure for different body parts, e.g. for abdominal surgery or the simulations of soft tissue compression caused by prostheses in orthopedic treatments, are also possible but need to be further investigated. Besides these medical utilizations, there are also applications beyond that scope in other fields of science, e.g. in the determination of material properties of polymer components.

However, more complex models as the ones that have been used in the study presented here may in future be necessary when it comes to the application in breast surgery planning. For instance, more different anatomical regions such as the muscular soft tissue as well as a distinction of soft tissue into an adipose and a glandular compartment may yield more accurate anatomical models. These models may contain more tissues with unknown material properties, thus the dimensionality of the design space increases and hence the optimization task becomes more complex. Furthermore the influence of the modeling of the skin may have a decisive role, especially when the anisotropic material properties are considered. For these models, as well as for the use of more complex theoretical models such as Mooney-Rivlin or Ogden with more parameters than just stiffness and compressibility that can be varied, the benefit of the optimization software OptiSlang becomes instantly more pronounced.
Numerical modelling

Description

The modal analysis of the BBN vehicle was performed using a three-dimensional finite element model developed in the ANSYS software. The use of a finite-element formulation allows considering the influence of the deformability of the car body, bogies and axles. Figure 1 presents a perspective of the numerical model. The car body was modelled by shell-finite elements while the bogies were modelled by beam-finite elements, with the exception of the suspensions, the connecting rods and the tilting system which were modelled by spring-damper assemblies. Additionally, the passenger-seat system was modelled in a simplified manner, by a one-DOF system composed of a mass over a spring-damper assembly. The masses of the equipment located in the under-floor of the car body and bogies were simulated through mass elements. The structure was discretised with 1082 shell elements, 1029 beam elements and 148 spring-damper assemblies. The total number of nodes is 1902, corresponding to 10,704 degrees of freedom.

Car body

Table 1 presents the main geometric and mechanical parameters of the car body's numerical modelling, including the designation, the selected value, the unit and the bibliographic references that were used. Additionally, the characteristics of the statistical distribution of some of the parameters, later used in the calibration phase of the model, are also shown. Figure 2 identifies the panels of finite elements considered in the numerical modelling of the car body in correspondence with the base, cover and side walls. In the finite-element panel is equal to the cross-sectional area of the real panel. The inertia correction of the panels, in directions x and z, was performed using the RMI parameter (Ratio of the bending Moment of Inertia):}

\[
RMI = \frac{I_{\text{real}}}{I_{\text{hand}}}
\]

where \(I_{\text{real}}\) is the real inertia of the panel and \(I_{\text{hand}}\) is the inertia calculated based on the thickness of the shell-finite element. The additional masses of the base, side walls and cover of the car body refer to the mass parcels of the items others under the component equipment and were uniformly distributed on the surface of the respective structural elements. The stiffness and damping parameters of the secondary suspension elements as well as their respective variation limits were estimated based on the values provided by the train's manufacturer.

Bogie

Figure 3 (see page 22) presents a perspective of the numerical model of the bogie. The chosen colours, combined with the legend, facilitate the identification of the different elements of the bogie. The beam elements connecting the wheel sets to the axle box have zero stiffness around their axle, so as to simulate the linkage with the axle box. The support conditions imposed on the bogie, particularly on the girders and on the tilting and load bolsters, allow translational vertical movements and rotations around the x and z axes, preventing any other movements. Table 2 shows the geometrical characteristics of the sections of the various elements. The geometric characteristics are expressed in terms of the area (A) and inertias (I). Table 3 (see page 22) describes the main mechanical and geometrical parameters of the numerical model and the characteristics of the statistical distribution of certain parameters, which will be used in the model's calibration phase.

The stiffness and damping parameters of primary suspension elements and their respective variation limits were estimated based on information from the manufacturers. The additional mass of the bogie, at the girders, crossbars and axles, is related to the mass of springs, dampers, connecting rods, links, reinforcement plates, axle boxes and others. These masses were linearly distributed in the different elements. In what concerns the girders the additional mass was further divided into two parcels according to their location: in the central zone, i.e. in the sections located between the crossbars and at the extremities. The wheel–rail connection was modelled by a spring element with unidirectional behaviour.
Table 4 shows the damped and undamped natural frequencies of the main vibration modes of the BBN vehicle. In what concerns the bogies, for modes 1B and 2B, there are different frequency values according to the movement of the two bogies in phase and in antiphase, respectively. In the 3B mode, the different values of the frequencies are related to the isolated movements of the left and right bogies, respectively. The modal results show differences between damped and undamped natural frequencies. These differences are more notorious for the rigid body modes of the car body and bogies since these modes involve significant movements of the suspensions. In case of the bogies the differences are even more important since this additional damping is provided simultaneously by the primary and secondary suspensions. Figure 4 illustrates the modal configurations associated with rigid body modes (1C, 2C and 3C) and structural modes of distortion (4C), bending (5C) and torsion (6C) of the car body. In these modes the movements of the bogie have very low amplitude.

![Modal parameters](image)

Table 4 shows the damped and undamped natural frequencies of the main vibration modes of the BBN vehicle. In what concerns the bogies, for modes 1B and 2B, there are different frequency values according to the movement of the two bogies in phase and in antiphase, respectively. In the 3B mode, the different values of the frequencies are related to the isolated movements of the left and right bogies, respectively. The modal results show differences between damped and undamped natural frequencies. These differences are more notorious for the rigid body modes of the car body and bogies since these modes involve significant movements of the suspensions. In case of the bogies the differences are even more important since this additional damping is provided simultaneously by the primary and secondary suspensions. Figure 4 illustrates the modal configurations associated with rigid body modes (1C, 2C and 3C) and structural modes of distortion (4C), bending (5C) and torsion (6C) of the car body. In these modes the movements of the bogie have very low amplitude. Figure 5 shows the modal configurations, in perspective and cross-section view, of a bogie of the vehicle. Mode 1B comprises the bouncing movement of the bogie. Modes 2B and 3B comprise the rolling and pitching movements of the bogie, respectively. In these modes the car body shows very limited movements.

Calibration methodology

The results of the conducted experimental tests of the BBN vehicle involving the dynamic tests of the car body, bogie and passenger-seat system are used to calibrate the numerical model of the vehicle. The calibration of the numerical model of the BBN vehicle was performed using an iterative method based on an optimisation technique. This method consists on the resolution of an optimisation problem, which consists of the minimisation of an objective function by varying a set of the preselected model parameters. The pre selection of the numerical parameters is carried out based on a global sensitivity analysis. Figure 6 (see page 24) presents a flow-
where $f_i^{exp}$ and $f_i^{num}$ are the experimental and numerical frequencies referring to mode $i$, $\Phi_i^{exp}$ and $\Phi_i^{num}$ are the vectors containing the experimental and numerical modal information related to mode $i$, and $a$ and $b$ are weighting factors of the objective function terms and $n$ is the total number of vibration modes.

**Calibration**

The experimental calibration of the numerical model of the BBN vehicle was performed based on modal parameters which were identified by the dynamic tests of the bogie and car body. The first phase focused on the calibration of the numerical model of the bogie under test conditions [1]. The second phase focused on the calibration of the complete numerical model of the vehicle. The numerical parameters of the bogie estimated in the first phase were assumed as deterministic parameters in the second phase.

**Calibration of the bogie**

*Numerical model under test conditions*

The calibration of the numerical model of the bogie forced the development of a model that would reproduce the specific conditions of the test. Changes to the original model involved the removal of springs and dampers from the secondary suspension's elastic blocks, with proportional damping matrix based on a classic energy criterion.

*Objective function*

\[ f(d) = a \sum_{i=1}^{n} \left| \frac{f_i^{exp}}{f_i^{num}} - 1 \right| + b \sum_{i=1}^{n} [MAC(\Phi_i^{exp}, \Phi_i^{num}) - 1] \]

where MAC is the MAC value, $f_i^{exp}$ and $f_i^{num}$ are the experimental and numerical frequencies referring to mode $i$, and $a$ and $b$ are weighting factors of the objective function terms and $n$ is the total number of vibration modes.

*Convergence criteria?*

- NO
  - YES

Fig. 6: Calibration methodology for the numerical model

---

**Table 5: Characterization of the parameters of the numerical model of the bogie under test conditions**

<table>
<thead>
<tr>
<th>Parameter Designation Type</th>
<th>Average value/standard deviation</th>
<th>Limits (lower/upper)</th>
<th>Adopted value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k_s$ Stiffness of the secondary suspension's elastic block</td>
<td>Dir $y^*$</td>
<td>Uniform</td>
<td>12,000/17,132</td>
<td>9,000/15,000</td>
</tr>
<tr>
<td>$f_{00}$ Frequency of the contact point of the actuation system</td>
<td>Dir $y^*$</td>
<td>Uniform</td>
<td>25,250/14,289</td>
<td>5,000/10,000</td>
</tr>
<tr>
<td>$\alpha_{min}$ Position of the contact point of the actuation system</td>
<td></td>
<td>Uniform</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\alpha_{max}$ Right side</td>
<td></td>
<td>Uniform</td>
<td>0.9</td>
<td>0</td>
</tr>
<tr>
<td>$\alpha_{max}$ Right side</td>
<td></td>
<td>Uniform</td>
<td>0.9</td>
<td>0</td>
</tr>
<tr>
<td>$c_{mod}$ Modulus of deformability of wood</td>
<td>Uniform</td>
<td>8,2/3</td>
<td>4/12</td>
<td>10</td>
</tr>
</tbody>
</table>

The correlation matrix shows that the stiffness of the primary suspensions, the additional mass of the giders (central area and extremities) and the stiffness of the lower traction rod of the axle box have significant influence over the vibration frequencies. In turn, the position of the actuators affects MAC values, particularly in modes 18T. The vertical stiffness of the secondary suspension blocks influences...
ences the vibration frequencies and also the MAC values in a significant way. The remaining analysed parameters do not have significant influence on the modal responses and were therefore excluded from the optimisation phase. The influence of the primary suspensions’ stiffness over the frequencies of modes 1BT and 1BT, for which the distance between the suspensions and the rotation axle of the bogie is larger, should be emphasised. In these modes the elastic block of the suspension has no influence over the responses due to its location near the rotation axle. It is not the case of the frequencies of modes 2BT and 3BT, which involve transverse translation and rotation of the bogie, respectively, and for which the stiffness of the suspension blocks, compared with the primary suspensions, is decisive for controlling the responses.

Optimisation

The optimisation of the model involved finally 10 numerical parameters and 16 modal results (8 vibration frequencies and 8 MAC values). The genetic algorithm was based on an initial population of 30 individuals considering 250 generations, in a total of 7500 individuals. The initial population was randomly generated by the Latin Hypercube method. A number of elites equal to 1 and a number of substitute individuals also equal to 1 have been defined in this algorithm. The crossover rate was assumed to be 50% and the mutation rate was considered equal to 15% with a standard deviation the optimisation problem still includes restrictions related to the parameters of additional mass of the bogie. Optimal values of the parameters were obtained from the results of four independent optimisation cases (GB1–GB4) based on different initial populations. Figure 8 shows the ratios of the values of each parameter of the model in relation to the limits indicated in Tables 1, 3 and 5. The limits of the distributions of some of the parameters were extended, such as the cases of the stiffness of the primary suspension (500/1000 kN/m), and the stiffness of the axle box’s traction rods (3/10 and 10/40 MN/m) due to the systematic tendency of the optimum solutions of these parameters to reach the limits indicated in Table 3. A 0% ratio means that the parameter coincides with the lower limit. A ratio of 100% means that it coincides with the upper limit. The stiffness and damping parameters of the primary suspension and increments of mass are presented in Figure 8a indicating, in brackets, the numerical parameters’ values.

The parameters related to the elastic elements (blocks and rod) and actuation system is shown in Figure 8b. The analysed parameters present a good stability with variations below 10%, except for the damping of the primary damper. This is one of the parameters that the sensitivity analysis has shown to have a smaller influence over the numerical responses. Figure 9 summarises the error values between the numerical and experimental vibration frequencies, taking as reference the average values of the experimental frequencies, and the values of the MAC parameter, before and after calibration. The results after calibration are related to optimisation case GB2, which was the one presenting the lowest residual of the objective function. The average error of the frequencies decreased from 10.6% before calibration to 0.8% after calibration. In turn, the average value of the MAC parameter increased from 0.894 before calibration to 0.95 after calibration. As visualised in Figure 10, the experimentally obtained and numerically derived optimised modal configurations of the bogie coincide almost perfectly.

Calibration of the complete model of the BBN vehicle

Sensitivity analysis

Figure 11 presents the results of the global sensitivity analysis using Spearman’s rank correlation coefficient. The sensitivity analysis was performed using a stochastic sampling technique based on 250 samples generated by the Latin Hypercube method. This analysis was based on the parameters intervals presented in Table 2. The random generation of samples, particularly for the parameters of the car body’s additional mass, was subject to the following restrictions:

\[- \varepsilon \leq \Delta \mathbf{M} = [\mathbf{L}_1 \mathbf{M}_2 \mathbf{L}_3 \mathbf{L}_4 \mathbf{M}_5] \leq \varepsilon\]

where \(\Delta \mathbf{M}\) and \(\varepsilon\) represent the additional mass on the base, side walls and cover, respectively, and \(\varepsilon\) is a tolerance equal to 10%. The mode pairing was performed by application of a technique based on the modal strain energy and on the EMAC parameter. The correlation matrix shows that the stiffness of secondary suspensions, from front (KS1) and rear (KS2) bogies, has significant influence over the frequencies and MAC values of the rigid body modes of the car body. In turn, the RMI parameters from the base (RMIB) and side walls (RMIP) essentially control the frequencies and MAC values of the structural modes of the car body. The parameters additional mass and stiffness of the connecting rod between the tilting and load bolsters (Kb) have significant influence over the vibration frequency of mode 1C. The remaining analysed parameters did not have significant influence with respect to the modal responses, and were consequently excluded from the optimisation phase.

Optimisation

The optimisation of the model finally involved 7 numerical parameters and 10 modal results (5 vibration frequencies and 5 MAC values). The control parameters of the genetic algorithm and the objective function are identical to those in the optimisation of the bogie. The optimisation problem also included constraints involving the car body’s additional mass parameters. Optimal values of the parameters were obtained from the results of four independent optimisation cases (GC1–GC4) based on different initial populations. Figure 12 (see page 28) shows the values’ ratios of each parameter of the model in relation to the limits given in Table 1. The lower and upper stiffness limits of the secondary suspension were extended from 242 and 272.9 kN/m to 200 and 400 kN/m, respectively. Parameters related to the characteristics of the secondary suspension, connecting rod and geometrical properties of the car body are presented in Figure 12a indicating, in brackets, the estimated values for the stiffness of the secondary suspension. The parameters referring to mass distribution are presented in Figure 12b.
parameters of the base and side walls. The stiffness values of the front bogie’s secondary suspension are higher than those estimated for the rear bogie. Regarding the additional masses of the side walls and cover, the estimates show higher variations, close to 25%. This should be related to the fact that these parameters contribute in a similar way to the participant mass on vibration mode 1C. Therefore, there may be different combinations of these parameters leading to the same solution, in terms of optimisation of the problem. Figure 13 summarises the error values of the numerical and experimental vibration frequencies taking as reference the average values of the experimental frequencies, and of the MAC parameter, before and after calibration. The results after calibration are related to the GC1 optimisation case, which was the one with the lowest final residual of the objective function. The frequencies’ average error dropped from 20.3%, before calibration, to 2.9%, after calibration. This error decrease is mainly due to the reduction of the error associated with the frequencies of structural modes 4C and 5C. The average value of the MAC parameter did not change significantly, increasing from 0.927, before calibration, to 0.937, after calibration. The excellent agreement between the car body’s experimentally obtained and numerically derived optimised modal configurations can be verified in Figure 14.

Final results
The combination between the numerical parameters, obtained for the optimisation case of the bogie GB2, and the parameters obtained in optimisation case of the complete vehicle GC1, were the basis for the establishment of the vehicle’s calibrated numerical model. Table 7 presents the values of the damped vibration frequencies of the main vibration modes of the BBN vehicle obtained from the calibrated numerical model. Comparing the values of the frequencies with the values given in Table 4, concerning the initial numerical model, there is a visible tendency towards the frequency increase on the rigid body modes of the car body and bogies, being that, in the bogies’ case, this increase ranged from 10% to 15%. This tendency is due to the significant increase of the stiffness of the primary and secondary suspension springs. In turn, the structural modes of the car body, particularly modes 4C and 5C showed a decreased tendency of approximately 20%, mainly due to the reduction of the RMI parameter of the car body’s side walls.

Conclusions
This article described the calibration of the numerical model of a BBN vehicle of the Alfa Pendular train based on modal parameters. The calibration of the numerical model was conducted through an iterative methodology based on an optimisation algorithm and was performed using a multistep approach involving two phases: the first phase focused on the calibration of the model of the bogie under test conditions and the second focused on the calibration of the complete model of the vehicle. Global sensitivity analysis allowed the identification of numerical parameters to be considered in the calibration. The parameters that have shown the highest sensitivities in relation to the modal responses were, for the bogie, the vertical stiffness of the secondary suspension block and the vertical stiffness of the primary suspensions. As for the car body, the RMI parameters of the base and side walls and the vertical stiffness of the secondary suspension were the parameters with highest sensitivity in relation to the modal responses.

The optimisation was conducted using a genetic algorithm involving a total of 17 numerical parameters and 26 modal responses. The results of the optimisation cases of the bogie and vehicle, based on different initial populations, led to the same solution, in terms of optimisation of the problem. Figure 13 summarises the error values of the vibration modes of the complete vehicle, the average error of frequencies went from 20.3%, before calibration, to 2.9% after calibration. Significant improvements were also observed in MAC values, particularly in the vibration modes of the bogie. This result demonstrates the robustness and efficiency of genetic algorithms on the estimation of the vehicle’s modal responses. The combination of numerical parameters obtained for the GB2 bogie optimisation case with the parameters obtained for the GC1 case of vehicle optimisation provided the basis for developing the calibrated numerical model of BBN vehicle. Compared with the initial numerical model, the calibrated numerical models show higher frequency values of the rigid body modes of the car body and bogies, essentially due to the increased stiffness of the primary and secondary suspension springs.

On the other hand, most of the car body’s structural modes tended to decrease, largely due to a reduction of the RMI parameter of the side walls of the vehicle’s car body. In future studies, the calibrated numerical model of the vehicle will be used to access the dynamic behaviour of the train-track coupled system, in terms of passengers comfort and wheel–rail contact stability, on plain track, on bridges or on transition zones.

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![Fig. 12: Values of numerical parameters for optimisation cases GC1–GC4: (a) characteristics of the suspension, connecting rods and geometrical properties of the car body, (b) masses of the side walls and cover, the estimates show higher variations, close to 25%. This should be related to the fact that these parameters contribute in a similar way to the participant mass on vibration mode 1C. Therefore, there may be different combinations of these parameters leading to the same solution, in terms of optimisation of the problem. Figure 13 summarises the error values of the numerical and experimental vibration frequencies taking as reference the average values of the experimental frequencies, and of the MAC parameter, before and after calibration.](image1)

![Tab. 7: Natural frequencies of the BBN vehicle obtained from the calibrated numerical model](image2)

<table>
<thead>
<tr>
<th>Element</th>
<th>Mode</th>
<th>Damped frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbody</td>
<td>1C</td>
<td>1.01</td>
</tr>
<tr>
<td></td>
<td>2C</td>
<td>2.34</td>
</tr>
<tr>
<td></td>
<td>3C</td>
<td>2.70</td>
</tr>
<tr>
<td></td>
<td>4C</td>
<td>8.19</td>
</tr>
<tr>
<td></td>
<td>5C</td>
<td>12.16</td>
</tr>
<tr>
<td></td>
<td>6C</td>
<td>17.73</td>
</tr>
<tr>
<td></td>
<td>1B</td>
<td>9.21/9.24</td>
</tr>
<tr>
<td>Bogies</td>
<td>2B</td>
<td>7.70/8.12</td>
</tr>
<tr>
<td></td>
<td>3B</td>
<td>14.16/14.09</td>
</tr>
</tbody>
</table>

The average error of vibration frequencies of the modes of the bogie under test conditions went from 10.6%, before calibration, to 0.8%, after calibration. Concerning the vibration modes of the complete model of the vehicle, the average error of frequencies went from 20.3%, before calibration, to 2.9% after calibration. Significant improvements were also observed in MAC values, particularly in the vibration modes of the bogie. This result demonstrates the robustness and efficiency of genetic algorithms on the estimation of the vehicle’s modal responses. The combination of numerical parameters obtained for the GB2 bogie optimisation case with the parameters obtained for the GC1 case of vehicle optimisation provided the basis for developing the calibrated numerical model of BBN vehicle. Compared with the initial numerical model, the calibrated numerical models show higher frequency values of the rigid body modes of the car body and bogies, essentially due to the increased stiffness of the primary and secondary suspension springs. On the other hand, most of the car body’s structural modes tended to decrease, largely due to a reduction of the RMI parameter of the side walls of the vehicle’s car body. In future studies, the calibrated numerical model of the vehicle will be used to access the dynamic behaviour of the train-track coupled system, in terms of passengers comfort and wheel–rail contact stability, on plain track, on bridges or on transition zones.

![Fig. 14: Comparison between the vibration modes of the carbody, experimentally and numerically obtained, after calibration](image3)