

Dynamic behavior of guideway switch beams

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ABSTRACT: The guideway switches for the Transrapid consist of a long bending beam supported at 15 m intervals as presented at the conference *MAGLEV'2004* [Fichtner&Pichlmeier2004]. The forces from the vehicle's support and guidance cause the bending beam to vibrate. Supported by the German Ministry BMVBS a switch with increased required lifespan for a Transrapid vehicle with increased payload (TR09) is developed. This demands a systematic analysis of the actual stress, taking vibration behavior into account. For this analysis two mechanical models are used. Firstly, analytical considerations and simulations using a Multibody System [Schiehlen1997] consisting of 37 elements are applied for parameter studies. Secondly, the stress situation is analyzed using a 3D-Finite Element model [Zienkiewicz1977] consisting of about 200 000 elements. Both models are verified by comparison with measurement results of the existing switches.

1 INTRODUCTION

The guideway switch has to withstand two kinds of loading: Firstly, the switch is statically loaded by the bending of its central beam in the turnout position. The bending displacement at the furthest end of the beam is 3.6 m, yielding tension and compression at the outer and inner side of the beam. Secondly, the switch is loaded dynamically by the passing vehicle. The main effect of the vehicle is the quasistatic load resulting from its mass; however, further loads arise from the vibration of the switch. The vibrations occur with several frequencies according to the Eigenvibrations of the switch and forced vibrations from the oscillating magnetic forces, as shown in Figure 1.

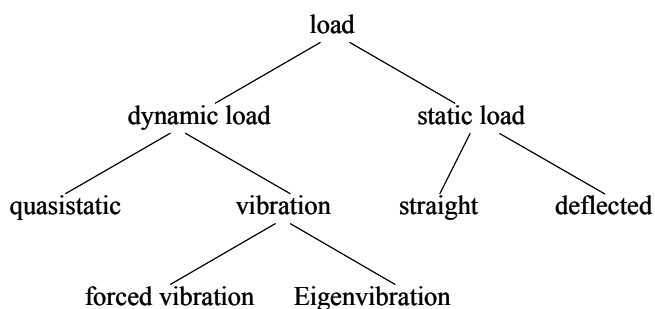


Figure 1: Classification of loads for the switch

With the current design of the bending beam and struts, the highest vibrational stress arises from the first torsional Eigenvibration at about 15 Hz, if no

measures for damping are used. While the stress in the beam itself is relatively moderate, it represents a considerable part of the net stress in the cantilever arms because the vibration amplitude increases with the distance from the rotation center. In the design of the switches in Shanghai this vibration was nearly completely eliminated by means of tuned mass dampers. However, to improve the design by increasing payload and lifetime, the vibration behavior is analyzed in finer detail.

2 METHOD OF ANALYSIS

In order to understand the vibration behavior in detail, at first the vibrations of the current design were analyzed by taking several measurements at the switches in Shanghai as well as on the test track in the Emsland (TVE). From these measurements the dominant vibration forms, frequencies and amplitudes could be extracted.

Secondly, a Finite Element Model (FEM) of the switch according to the technical drawings was set up. Using this model, the mechanical stress and strain in various parts can be determined for a given vibration amplitude and form. In particular the local stress peaks in certain parts can be calculated by this method, which is important as they can hardly be measured.

For an universal understanding of the vibration behavior, it is paramount to additionally determine the influence of the dominant design parameters on the Eigenvibrations. This includes, for example, the number and position of the struts and the stiffness of the support points, the height of the bending beam or the influence of gravel filling and other means for damping. Therefore thirdly, computations applying a Multibody System (MBS) model are carried out. The MBS model is greatly simplified with respect to the FEM model yielding much smaller computation times. In this way it is possible to analyze many different configurations. For some advantageous configurations determined using this model, a detailed stress analysis using the FEM model can be carried out.

3 MECHANICAL MODELS

Using the combined approach of a MBS model and a FEM model as described above the dynamic behavior can be investigated globally for many different configurations and, on the other hand, the local stress and strain situation can be analyzed in detail. Both models are verified by comparison with measurement results.

3.1 Multibody System

A Multibody System as shown in Figure 2 is used for investigating the global dynamic behavior. The model consists of 37 identical bodies, each of which has a length of 2064 mm. This length corresponds to the module length of the switch design.

Each module carries two guiding rails and four stator packs, as well as cover sheets on top of the guideway as shown in Figure 3 and Figure 6. Then, there is a small gap towards the adjacent modules in order to provide for the change of length of the outer and inner part of the switch during bending operation. The modules are mounted on a continuous bending beam which is the only connection between the modules. The cross-section of the beam is a welded, hollow rectangle with longer top and bottom flanges.

The corresponding bodies of the MBS each possess the mass and inertia of one of these modules as calculated by the FEM. The bodies are modeled using five degrees of freedom (d.o.f.) as the displacement along the track is a second order effect and not considered in the linear model. While four of the d.o.f. y_i , α_i , β_i and γ_i are free to move and influenced by the applied forces only, the fifth d.o.f. z_i is constrained by joints between the bodies. The applied forces are created by springs and dampers connecting the bodies with each other, as shown in Figure 2. The translational spring in y -direction is placed in the rotation centre for the static torsion of the beam.

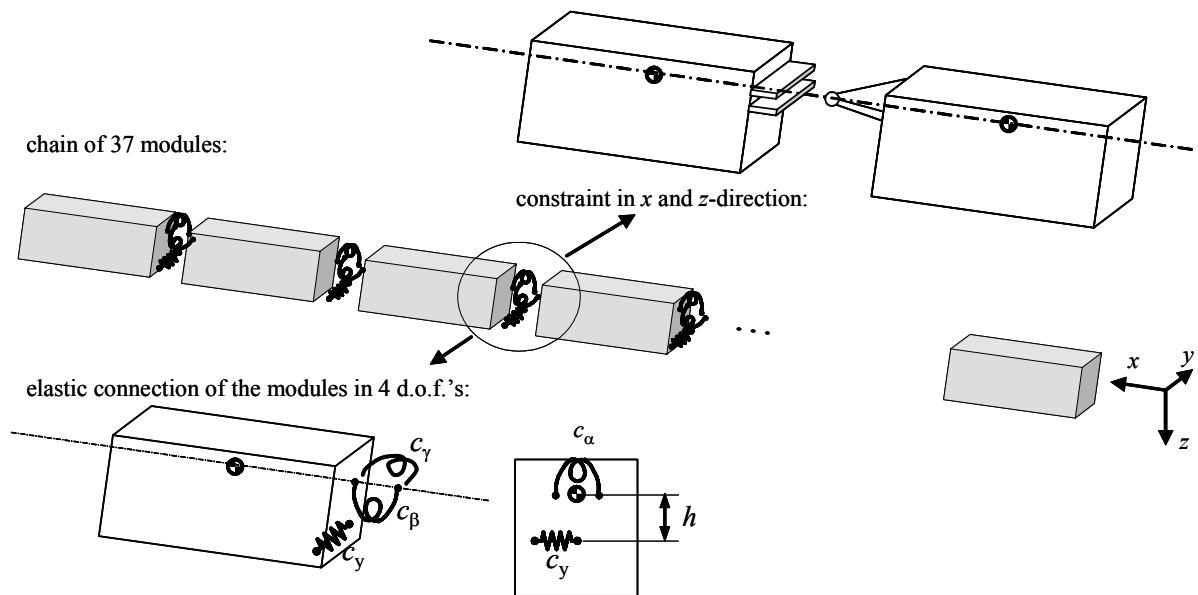


Figure 2: Multibody System model of guideway switch

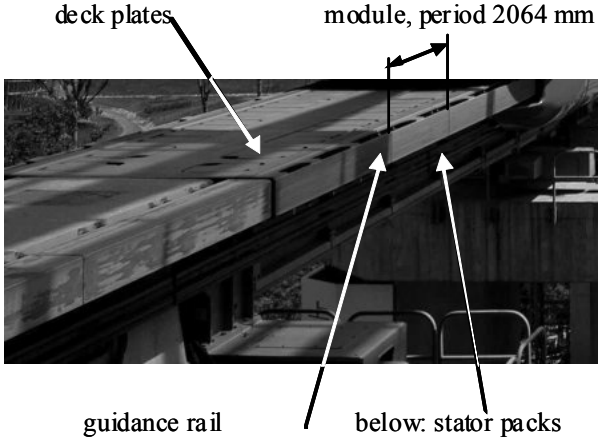


Figure 3: module of switch design

The stiffness parameters c_y , c_α , c_β , and c_γ are calculated from a comparison of the static response of the FEM with the MBS. For this comparison both models are reduced to a single module. The comparison consists of four steps.

- step 1)* application of torsional torque M_α in the FEM, measurement of torsion angle $\Delta\alpha$ and distance h of torsion axis from gravity axis. Stiffness $c_\alpha = M_\alpha/\Delta\alpha$
- step 2)* application of bending torque M_β , measurement of bending displacement Δz . Stiffness $c_\beta = M_\beta/(\Delta z/l)$
- step 3)* application of bending torque M_γ , measurement of bending displacement Δy . Stiffness $c_\gamma = M_\gamma/(\Delta y/l)$
- step 4)* application of force F_y about centre of rectangle cross-section, measurement of bending displacement Δy . Stiffness c_y calculated from $\Delta y = F_y/c_y - M_\gamma/c_\gamma l + M_\alpha/c_\alpha h$ using $M_\gamma = -F_y l$ and $M_\alpha = -F_y h_2$, where h_2 is the distance between the centre of rectangle and the static torsion axis.

The corresponding values are given in Table 1 in comparison to analytical stiffness properties of the cross section beam only. The values in Table 1 are normalized with respect to Young's modulus E , the shear modulus G , the corresponding area moments of inertia I_t , I_{yy} , I_{zz} , the area in y -direction A and the width of the beam b . For the analytical approximation the cross section is considered as a hollow rectangle with extended upper and lower flanges. The

values for the stiffnesses are then computed from a comparison of the MBS properties with the analytical solution of Bernoulli's beam, which yields –after using the formulae for the special finite series from [Bronstein et al. 2001]– the results shown in Figure 4. The results show convergence of the solution for torque loads with the solution for force loads for $n \rightarrow \infty$. As the switch is supported at intervals of about 15 m the solution for $n = 7$ and $L = 7 * 2.064 \text{ m} = 14.5 \text{ m}$ is given in Table 1. The difference between the FEM and the analytical solution is mainly due to the stiffness of the cantilever arms, partition plates and further attached parts that are neglected in the analytical solution. For the torsional stiffness, additionally, the effect of the extended upper and lower flanges was neglected in the analytical approximation. From the difference between the analytical and the numerical solution, the influence of the attached parts on the stiffness of the switch can be estimated. As Table 1 shows, the attached parts lead to increased torsional stiffness and increased stiffness for lateral bending but not for vertical bending. This is an important piece of information when considering design changes of these parts of the switch.

Table 1: Comparison of analytical approximation and numerical computation of stiffness properties

stiffness parameter	analytical approximation, bending beam only		FEM
	torque load	force load	full module
$c_\alpha / (G I_t / l)$	1.0	-	1.8
$c_\beta / (E I_{yy} / l)$	1.1	1.2	1.2
$c_\gamma / (E I_{zz} / l)$	1.1	1.2	2.4
$c_y / (E A / b)$	-	-	1.1

The support points at the struts are modeled as springs between the corresponding body and the ground at the given position of the struts. The stiffness parameters are calculated from analytical considerations using the technical drawings of the support design and by comparison with FE-results.

The complete model is set up in SIMULINK [The Mathworks Inc. 2006] using the recursive method described in the textbook [Schiehlen&Eberhard2004].

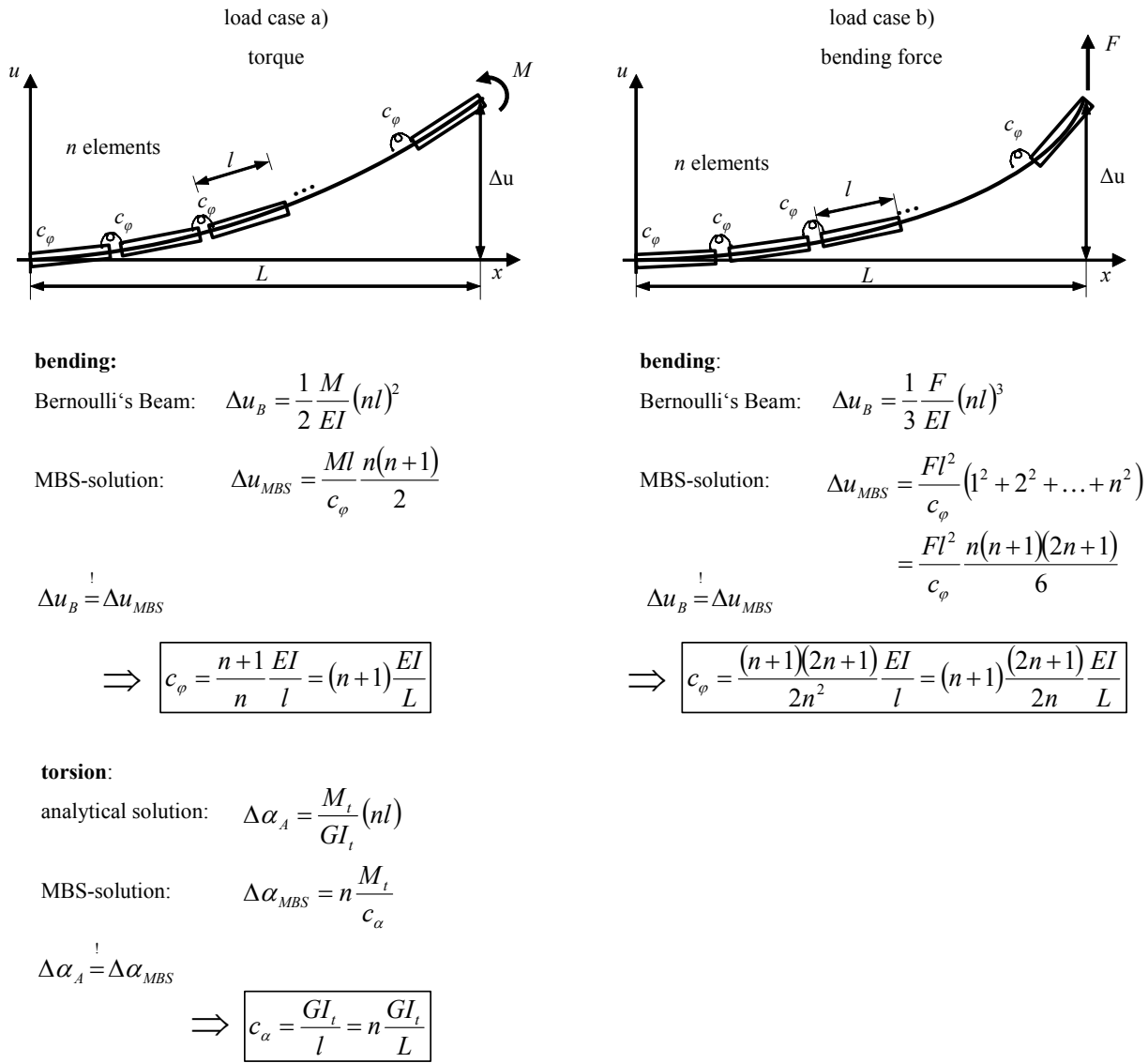


Figure 4: Analytical computation of stiffness parameters for discretization

3.2 Finite Element Model

For a few given configurations, the stress and strain behavior is computed in detail using the FEM model as shown in Figure 5. The model consists of 220 000 elements and possesses 1.1 Mio. degrees of freedom. Each of the 37 modules of the switch is modeled as shown in Figure 6 with slight alterations at the ends of the switch.

The struts are modeled as shown in Figure 7 with fixed boundary conditions at the traverse beams which in reality run on rails at each strut. The flexibility of the traverse beams is accounted for by the insertion of elastomer plates between traverse beams and the connecting sheets. This simplification is valid as the traverse beams are locked at each strut and much stiffer than the steel sheets connecting the traverse beams to the bending beam.

The FEM model is built and computed using the program ANSYS [Ansys Inc. 2006].

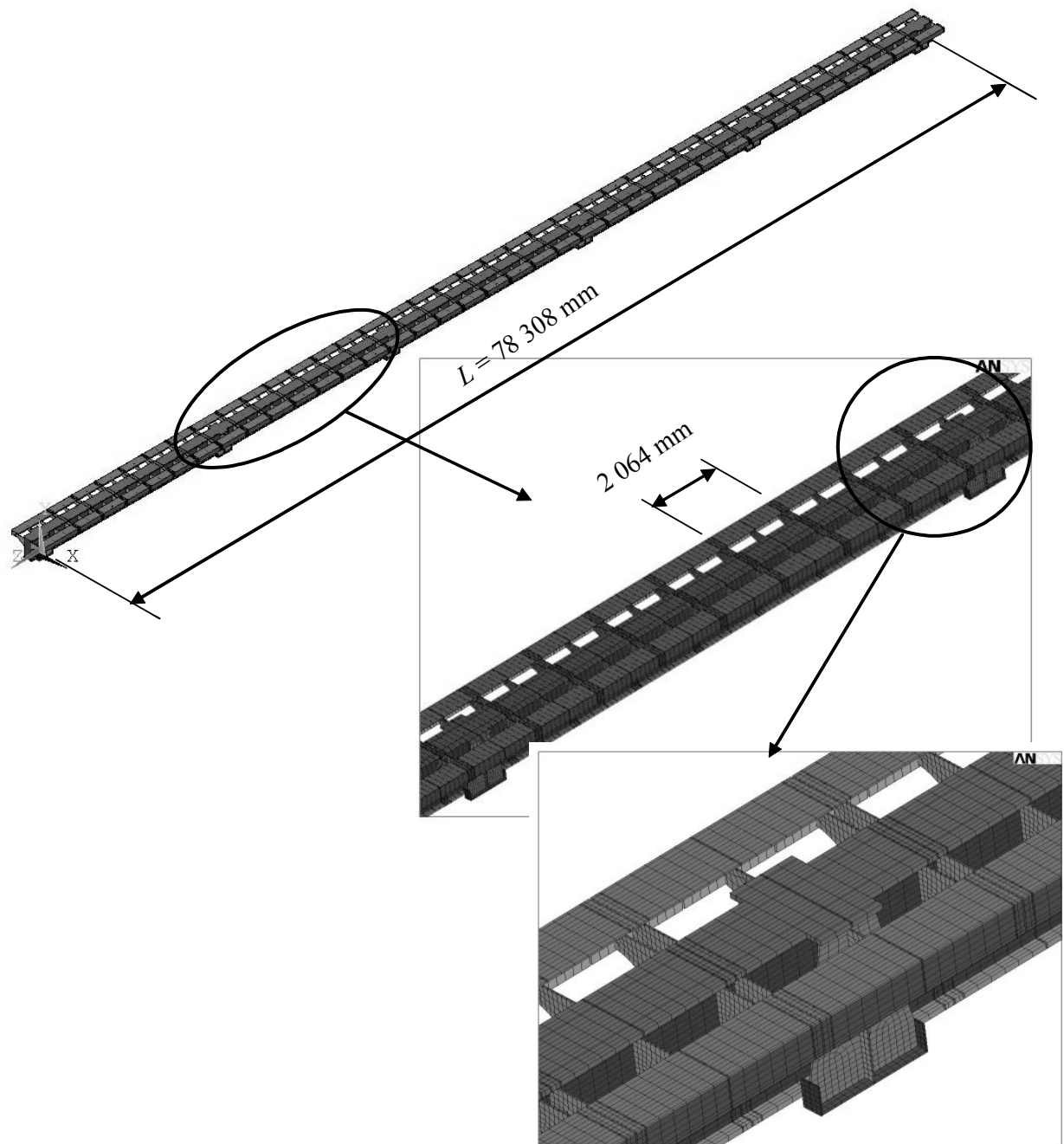


Figure 5: Finite Element Model of guideway switch

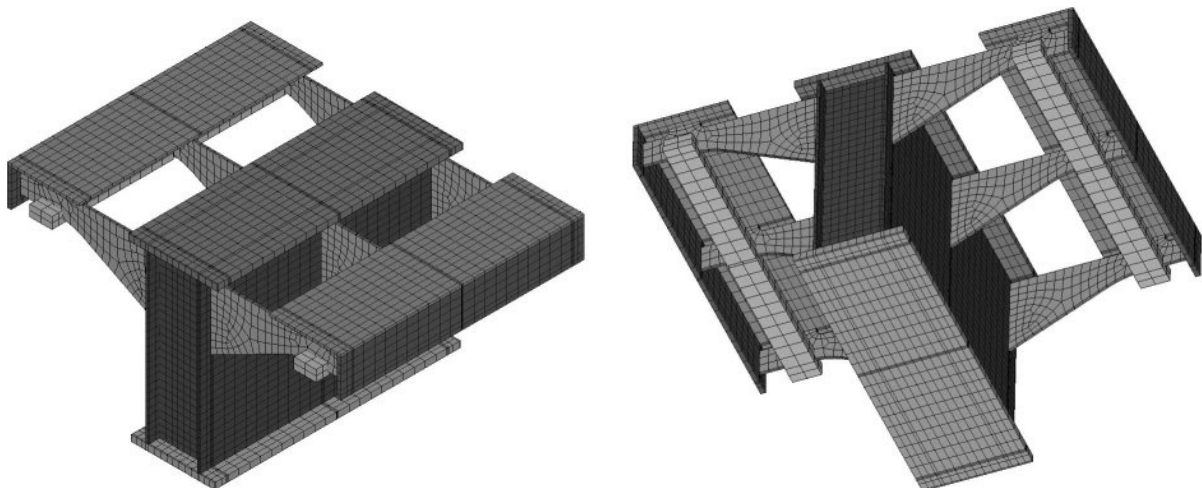


Figure 6: Detail of FEM: one out of 37 (nearly) identical modules

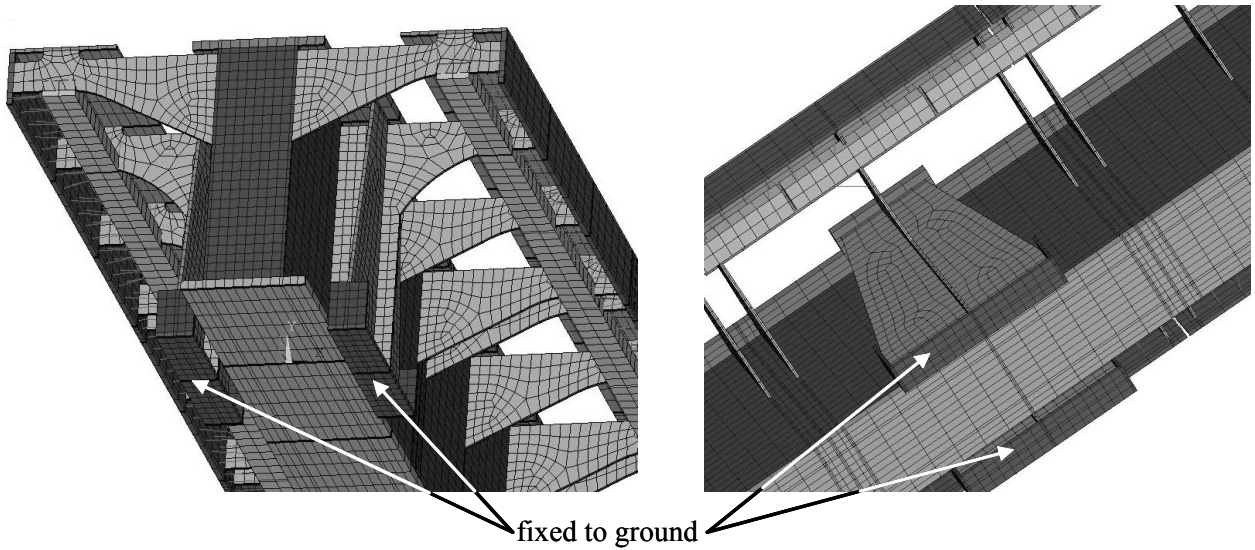


Figure 7: Detail of FEM: boundary condition at struts

4 VERIFICATION BY MEASUREMENTS

The results of both models in the original configuration are verified by comparison with measurement results for the switch in Shanghai. As an example, the 1st torsional Eigenvibration is shown as a result of the FEM computation in comparison to the measurement results from Shanghai in Figure 8.

The switch in Shanghai was investigated with and without a tuned mass damper mounted in the middle of the bending beam. Both configurations can therefore be compared to the numerical results and the validity of both the structural model and the consideration of the tuned mass damper can be confirmed.

Finite Element computation:

$$f_{\text{FEM}} = 14.99 \text{ Hz}$$

measurement:

$$f_{\text{meas}} = 14.9 \text{ Hz}$$

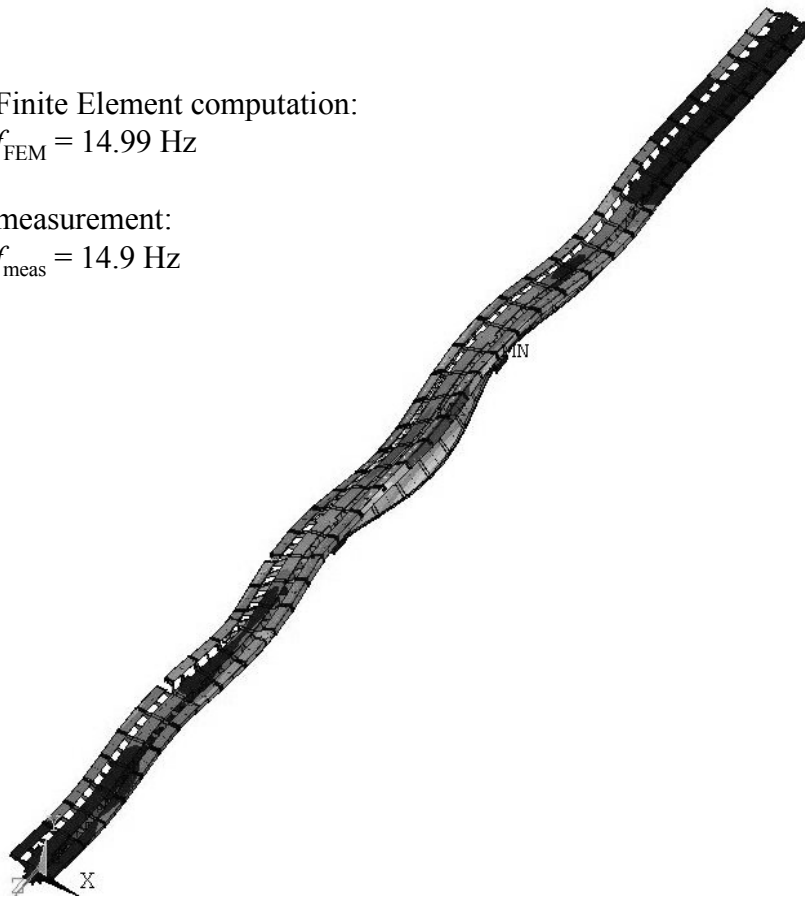


Figure 8: Comparison of FEM with measurement results for the 1st torsional Eigenvibration

5 SUMMARY

A combined approach using two models, one very detailed and the other much simplified, is presented for the investigation of the dynamic behavior of the guideway switches. Both models are verified by measurements at the existing switches.

Using these models, the dynamic behavior can be analyzed universally. In particular, following aspects can be investigated:

- Shape of the Eigenvibrations and their sensitivity with respect to the design variables of the switch
- Stress and strain in various parts of the switch for a given deflection and loading
- Effect of damping measures, e.g. gravel filling or tuned mass dampers

By combining the methods described, an optimization cycle for improving the dynamic behavior in the future can be set up as shown in Figure 9.

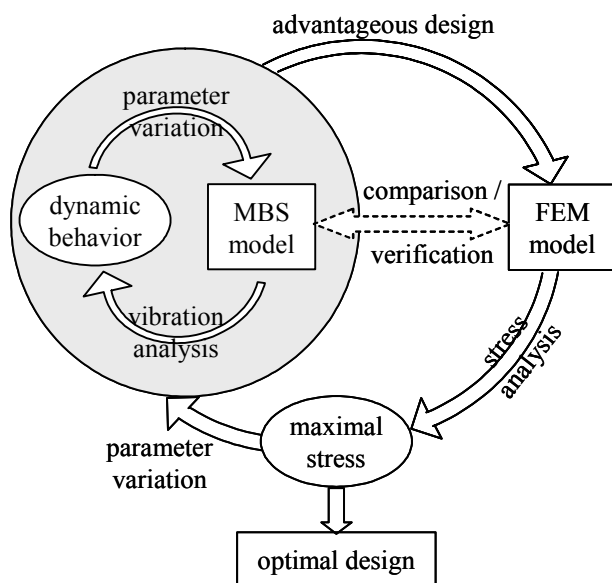


Figure 9: Optimization procedure with combined MBS/FEM approach

6 ACKNOWLEDGEMENTS

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