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TOWARD IMPROVED TRANSPORT SOLUTIONS – EXPERIMENT, MODELING AND STRUCTURAL DESIGN

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Abstract

Transportation as an engineering discipline considers ways how to move objects easier, faster and less expensive. The posed challenges call for novel structural materials, modeling and design solutions and even novel structural features, such as adaptive behavior. The development in all these areas demands experiments, suitable modeling techniques, design optimization and innovative solutions in structural design. This paper gives a partial overview of the performed and ongoing work and developments done at the Department of Structural and Computational Mechanics of TU Berlin addressing some of the above mentioned aspects.

Key words*: structural design, dynamics, modelling & simulation, transport, adaptive structure, composite, experiment*

1 INTRODUCTION

Transportation as an engineering discipline calls for solutions that enable to move objects easier, faster and quieter, but at the same time less expensive. This request breaks down to a number of simpler requirements, such as: make the transport machines and load carrying structures lighter and quieter but stronger, more reliable and comfortable, safer and more robust, less expensive to produce and operate, etc. The innovative solutions are obtained by applying modern structural materials, optimal design and novel structural features. These further demands:

- **Experiments** – as the ultimate way for testing the structural behavior and properties, particularly those made of novel materials but also as an integral part of developed methods to monitor and validate their condition. Those methods should be non-invasive, if possible, and reliable. Experimental modal analysis is particularly in the focus as a procedure that provides valuable information about the overall mass and structural stiffness distribution, and therewith an insight about the options to influence the structural dynamics by deliberate change of structural properties or by means of active damping. We do not only perform the experimental modal analysis to extract the structural modal data, but we also use it in an innovative way to estimate the condition of the structures made of modern structural materials, composite laminates.
- **Modeling** – as one of the fundamental activities engineers are involved in. It aims at a good balance between the model complexity and the accuracy achievable in a simulation. To achieve such a compromise, one need to clearly define the model objectives, i.e. which aspects of the simulated phenomena are of interest and which can be neglected. This approach to modeling is well summarized in the well-known Einstein's sentence: "*Everything should be made as simple as possible, but not simpler*". We are very active in this field, and particularly in the field of structural dynamics. Our developments aim at highly efficient solutions ranging from solutions based on model reduction techniques to full fidelity FEM solutions. Special feature of our solutions is the coverage of geometrically nonlinear effects with great efficiency. The efficiency of some solutions allows even real-time simulations.
- **Structural design** – as another fundamental engineering activity. It implies optimization with respect to various objectives and which can be performed in various levels – from material, over geometry and even by providing structures with new features that resemble biological system, thus rendering their behavior highly robust. Our work includes structural optimization of transport means with the objective of minimizing noise radiation. Also, we have also embraced the idea of adaptive structures which has redefined the concept of structures from a conventional passive deformable system to an adaptive controllable system with inherent self-sensing, diagnosis, actuation and control capabilities. We have provided our modeling tools, particularly for thin-walled active structures made of composite laminates with piezoelectric patched involved.

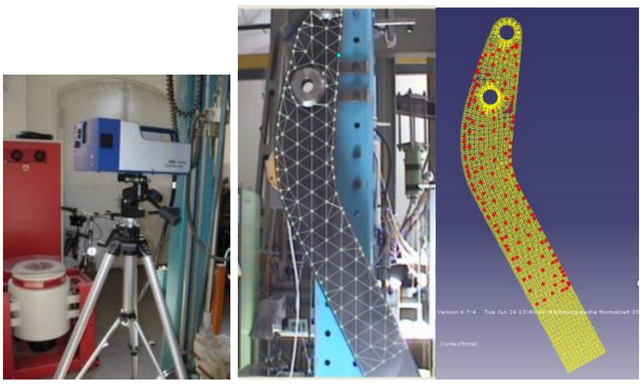
2 EXPERIMENTAL WORK

Since mass reduction is one of the basic demands in transportation, the use of composite materials in design has seen enormous progress over the past few decades,. Such novel materials have opened new frontiers in optimization of structural design. Composite materials enable the optimization that begins already on the material level – there are vast number of options to tailor material properties by choosing the constituent materials of a composite, their portions, form of reinforcements, thickness, number, angle and sequence of layers in a laminate, etc. Truck-mounted deployable concrete boom pumps can be seen as a rather illustrative example of transport machinery, at which application of composite laminates has enabled to redefine the operative range, Fig. 1). The advantage of composites lies in their high strength to weight ratio, good stiffness and functionality as well as non-corrosive features, among others. However, we are still in the relatively early days of application

Fig. 1 *Mobile concrete boom pump 60-Meter-Class (Putzmeister AG) and test of parts of the boom*

of composite materials, and every piece of reliable information that can be gained is very valuable. Experiments are the ultimate way of achieving this goal.

We perform experimental modal analysis of various structures on regular basis. We use a state of the art laser scanning vibrometer from Polytech (Fig. 2a). Typically, a structure is excited by means of a shaker within certain frequency range. The scanning-software defines a mesh of measurement points over the real structure. The results are obtained in the form of frequency response diagrams, from which eigenfrequencies and modal damping factors can be extracted. Structural vibration forms belonging to any of the excitation frequencies can also be obtained and therewith the eigenmodes. Figure 2b shows one of the first test objects and the generated scanning mesh (as generated and taken by the scanning vibrometer). It is an arm web segment of a concrete boom pump. Figure 2c shows the FE model, with a much finer FE mesh, where the marked points depict the FE mesh nodes that closely match the measurement points.



a) b) c)

Fig. 2 *a) Laser vibrometer and CFC arm: b) real structure with scanning meshc) FEM model with matched scan points*

This measurement can also be used to check the quality of the numerical (FE) model. For this purpose we use modal Assurance criteria like MAC, which compares calculated ith mode, φ_{ci}, with experimentally determined jth mode, φ_{ej} (Table 1):

$$MAC_{ij} = \frac{\left(\varphi_{ci}^{T}\,\varphi_{ej}\right)^{2}}{\left(\varphi_{ci}^{T}\,\varphi_{ci}\right)\left(\varphi_{ej}^{T}\,\varphi_{ej}\right)} \qquad (1)$$

Table 1 exemplifies the results of a MAC analysis. In this specific case, it proves high quality of the FE model (prepared in Abaqus – A-modes) through a very good agreement of the numerical and experimental (E-modes) results, despite numerous idealizations implemented in the numerical model.

Table 1 Modal Assurance Criterion (MAC)

		Measurement				
	MACs	E-Mode 1 7.9 Hz	E-Mode 2 35 Hz	E-Mode 3 97 Hz	E-Mode 4 182.9 Hz	E-Mode 5 213.1Hz
FEM calculation	A-Mode 1 7.25Hz	**0.997**	0.159	0.207	0.019	0.013
	A-Mode 2 35.5Hz	0.191	**0.98998**	0.045	0.219	0.042
	A-Mode 3 68.7Hz	0.058	0.034	0.689	0.009	0.08
	A-Mode 4 91.9Hz	0.18	0.05	**0.947**	0.008	0.092
	A-Mode 5 94.3Hz	0.076	0.001	0.0036	0.005	0.0095
	A-Mode 6 174.3Hz	0.016	0.25	0.051	**0.951**	0.023
	A-Mode 7 217.5Hz	0.014	0.0003	0.17	0.0029	**0.8999**

Furthermore, we use experimental modal analysis as an integral part of the novel method for non-destructive testing of composite materials. The approach is based on the assumption that the detection of potentially faulty areas can be obtained with combination of modal strains from FE analysis with measured nonlinear parameters. This is a particularly attractive idea, having in mind that modal data is relatively easy to obtain. The method requires both the real structure and its FE model.

One of the major causes of composite material damages is related to delaminating. Delamination may be caused by impact loads, fatigue or poor fabrication. If it occurs and if the state of the damage has not progressed too much, the natural frequencies and mode shapes do not change remarkably and can, therefore, alone not be used to detect faulty areas of the structure. However, a measurable change in certain parameters of vibration motion (caused by nonlinearity of the damaged area) is expected to be observed.

Intuitively, damping appears to be quite an appealing parameter for the purpose. As a consequence of delamination, an increase of energy dissipation during structural vibrations occurs. However, damping is a rather complex effect influenced by numerous aspects and its measurement turns out to be rather sensitive and with significant noise. Therefore, we have turned our attention to the fact that the presence of damage in a composite laminate gives rise to certain nonlinear effects in vibration motion. Investigations of the nonlinear response seem to be promising for localization and to quantify the size of the damage on a global basis. Actually, vibration of delaminated composite laminates leads to a non-smooth dynamic system due to continuously developing impact-like contacts along the delamination. Hence, the nonlinearity arises from the local contact and friction phenomenon. The delaminated layer and the remaining part of the structure periodically strike against each other during the vibration and the typical resulting difference in the frequency response functions is seen in Fig. 3. The general idea of our method is to determine nonlinear factors and perform their superposition [1]. This is done both by means of the FE-model, which yields a reference result for an undamaged structure, and for the actual damaged structure by means of measurements. The comparison between the obtained distributions of the superposed modal clapping factors is supposed to spot damage/delamination in the structure. The development of this method is still work in progress.

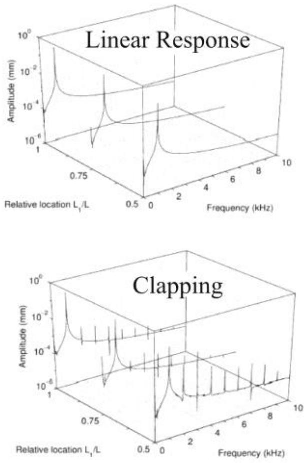


Fig. 3 *Difference in FRF of an undamaged and delaminated composite laminate*

3 MODELING

Structural dynamics, as a type of structural analysis that covers the behavior of structures subjected to dynamic loading, is a discipline used in many fields of engineering, involving automatic control, smart structures, automotive industry, multi-body systems, various types of interactive 'virtual-reality' simulators, to name but a few. The requirement for highly efficient simulation is intrinsic for all these fields and many researchers have devoted their work to this aspect. Typical engineering structures may easily have several 100,000 and even up to several million degrees of freedom (DOFs). For instance, the FE model of the rear car axle in Fig. 4a has 300,000 DOFs, while the FE model of the car model in Fig. 4b contains more than 5 million DOFs. Computing a dynamic response of such a model, even for a rather short time interval, is numerically very expensive.

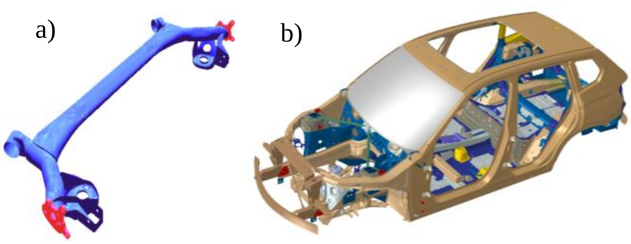


Fig. 4 *CAD models of: a) rear car axle; b) car chassis*

In order to keep the numerical effort in reasonable limits and having in mind that computation needs to be done for various loading scenarios of complex structures, such as those in Fig. 4, different model reduction techniques are used. The classical model reduction technique that is available in most commercially available FEM software packages is the modal superposition technique. It uses the natural frequencies and mode shapes to characterize the dynamic response of a structure. Besides some obvious advantages it offers, it also comes with certain drawbacks. The modal superposition technique is intrinsically applicable to linear analysis, as mode shapes, regardless how they have been determined, are the properties of the structure in its initial configuration. Hence, the technique covers only small deformations. In many cases, however, the actual deformation exceeds the limit of linearity and for adequate simulation accuracy, it would be necessary to cover those effects as well. In their nature, the nonlinearities could be geometric, material or related to variable boundary conditions (contact). We have developed techniques that enable extension of the modal superposition method into the realm of moderate geometric nonlinearities, which allows large computational saving. We also work on a novel approaches to modal reduction.

The first possibility to account for geometric nonlinearities would be the inclusion of the geometric stiffness matrix [2]. This approach takes into consideration the stress stiffening effect, which is appropriate for deformations characterized by relatively small displacements but large stresses induced. In this case, the tangential stiffness matrix is simplified as a sum of the linear stiffness matrix and the geometric stiffness matrix. Additionally, the stress state is computed in exactly the same manner as in the linear analysis, which results in the linear dependency of the geometric stiffness matrix on the deformation. Under these assumptions, the geometric stiffness matrix is obtained by superposition of the geometric stiffness matrices for single modes. In this manner, the basic advantage of the modal superposition technique, namely the high numerical efficiency, is kept.

The other possibility is referred to as warped displacement approach [3]. This technique aims at consideration of geometrically nonlinear behavior that is caused by relatively large rotation of single domains of the structure with respect to the structure as a whole. The approach consists in partitioning a complex structure into segments that perform relatively large rigid-body rotations with respect to the rest of the structure. Upon deformation, the average rigid-body rotation of the segments is determined and this amount of rotation is used to rotate the part of displacement field (determined in modal space) belonging to the segments. The technique can be particularly successful for beam-like structural segments.

Finally, it is also possible to divide a model into substructures and then use different approaches for different substructures e.g. Component Mode Synthesis of different level and "modes" depending on the need [4]. In this manner one may combine a full FEM model with the reduced one. The substructure modeled as a full FEM model provides the coverage of any type of nonlinearity. Such an approach does not offer a novelty in terms of model reduction methodology, but it represents a combination of existing techniques. Simple example is depicted in Fig. 5, where the tower crane is considered as a linear (reduced) model, while the ropes and suspended mass as a geometrically nonlinear part of the model.

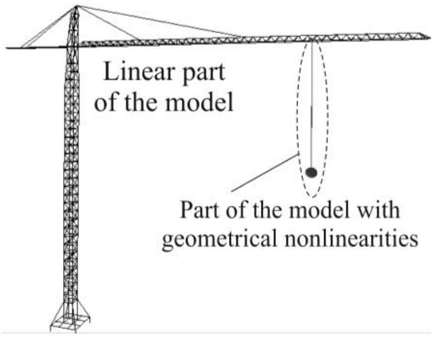


Fig. 5 *Linear – geometrically nonlinear model of a tower crane with suspended load*

We also work on new model reduction techniques such as those based on the Frequency Based Substructuring (FBST) [5]. The objective is to predict the dynamic behavior of a system of substructures based on the Frequency Response Functions (FRFs) of 'disassembled' substructures. For that purpose the structure needs to be adequately divided into substructures, the FRF of each substructure has to be determined as if it was a separate, stand-alone structure and, then, the equivalent reduced overall system is to be assembled (Fig. 6). The FRFs of the substructures can be obtained in different ways. One can use a FEM model to obtain the mass, damping and stiffness matrices needed to get the FRF. If not prohibitively expensive, experimental determination of FRFs could also be an option. In the frequency based substructuring method, the receptance matrix, which is the inverse of the dynamic stiffness matrix, is used. First, the dynamic equation is written in the frequency domain by means of Fourier transformation. The receptance matrix depends on the frequency and has to be determined for each frequency, which is computationally quite a demanding task for larger structures and a broader frequency range. Therefore, the FRFs of substructures are approximated in the vicinity of a certain frequency by means of the Krylov-based reduction method [5].

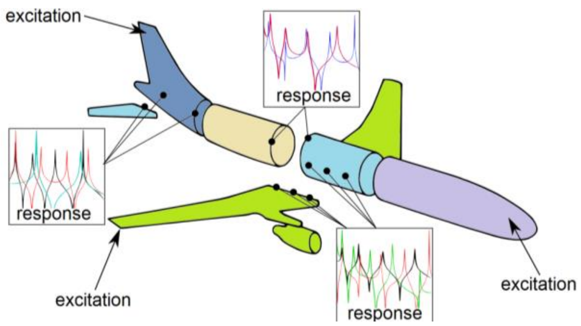


Fig. 6 *Schematic representation of the FBST*

In order to assemble the overall reduced system, the dynamic stiffness matrices of the substructures are determined by inverting the corresponding receptance matrices. The dynamic stiffness matrices are further used to assemble the overall dynamic stiffness matrix and, in the next step, it is inverted to obtain the receptance matrix of the complete system. Hence, a number of inversions are necessary, which is numerically not efficient. Instead, as proposed by Jetmunsen [6], the DOFs of each substructure are separated into the internal and interface DOFs, with the latter involved in the assembling procedure. The receptance matrix of the substructure is partitioned according to the two sets of DOFs. Thereafter, parts of the substructure receptance matrices related to the interface DOFs are assembled. Hence, only the submatrices related to the interface DOFs have to be inverted. For the assemblage, the Lagrange Multiplier Frequency Based Substructuring (LM FBS) method is applied, as proposed by Klerk and Rixen [7].

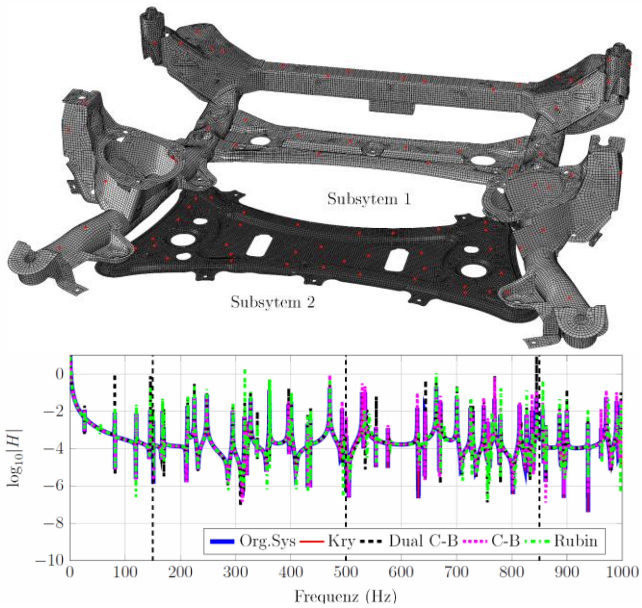


Fig. 7 *Engine mount divided into two substructures (Krylov-reduction from 486678 to 364 DOFs)*

We have also developed a FE formulation that keeps full fidelity FE models (no model reduction), cover geometrically nonlinear effects to a great extent and perform in real-time with several thousand elements with conventional hardware tools. The formulation is based on a simplified co-rotational FE formulation, in which the overall element motion is decomposed into the rigid-body motion and pure deformational motion [8]. This allows very high numerical efficiency as element behavior remains linear with respect to the element co-rotational frame that follows the element in its rigid-body motion.

4 STRUCTURAL DESIGN

Though the term 'design' quite often refers to aesthetic and ergonomic aspects of a product, it is used here in a more essential sense. It denotes the conceptual solution of a product that ensures the product performs, i.e. executes the aimed functions and, furthermore, that it is fit for its purpose, meaning that the performance is optimal or close to optimal. Hence, design implies optimization, and therewith optimization objectives and constraints.

One of the revolutionary ways of optimizing, or improving the structural behavior is by rendering structures adaptive. The concept implies that structures are supplied with active elements (sensors and actuators) coupled with a controller. Sensors provide signals that typically contain information about the state of the structure. The sensor signals are transmitted to a controller that implements the control law, i.e. the desired structural behavior. In other words, the controller processes the sensor signals and determines what action should be performed by actuators in order to produce a desired structural behavior. The corresponding signal is then sent to actuators. In this manner the structure can actively react to external excitations in order to adapt its response. Obviously, the adaptive systems mimic the behavior of natural systems. An example of such structural behavior would be the adaptive car roof depicted in Fig. 1. In this case, the adaptive behavior of the roof is used to suppress its vibrations with the final objective of diminishing noise in the car and thus improving driving comfort of passengers.

Fig. 8 *Adaptive car roof for cabin noise reduction*

We have developed several finite elements that can model most common type of adaptive structures [9]. Those are thin-walled structures made of composite laminates with embedded or bonded piezoelectric sensors and actuators. Modeling of such structures requires consideration of coupled-field effects. In this specific case, it is the electro-mechanical field.

Whereas the example in Fig. 8 shows that the noise reduction can be achieved by active vibration suppression, it is also possible to achieve this objective (with more or less success) by relatively simple structural modifications. The change of transport technology in Europe has seen many railroad freight cars returning empty after unloading. Some of those wagons have been in service unchanged for many years now and the bodies of the empty wagons in operation remarkably contribute to total radiation of noise. For the necessary acceptance of more rail traffic for environmental reasons it is essential to develop innovative freight wagons or change existing ones with lower noise emission. Since panels and other areas with thin walled parts of the freight cars are much more prone to noise radiation than other components, it is advisable to improve their acoustic behavior. For that purpose, we have developed and employed a new structural optimization methodology, see Fig. 9. In dealing with the problem of noise radiated by freight car panels, we have limited our approach to attaching curvilinear stiffeners and point masses to the surface of the panels. In this manner, the solution is kept rather inexpensive. The CMA-ES (Covariance Matrix Adaptation Evolution Strategy) optimization strategy is adopted to find out the optimal shape of the stiffener and its position together with the position of masses within a FE model with automatic adaptation of the FE mesh during optimization. Our efforts were also focused on design solutions for rail dampers (see Figs. 9 and 10) with exactly the same objective – reduction of radiated noise.

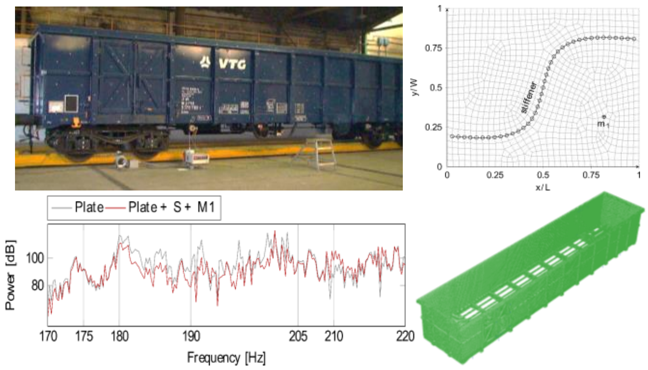


Fig. 9 *Noise radiation reduction of an empty fright wagon using Evolutionary Optimisation Strategy*

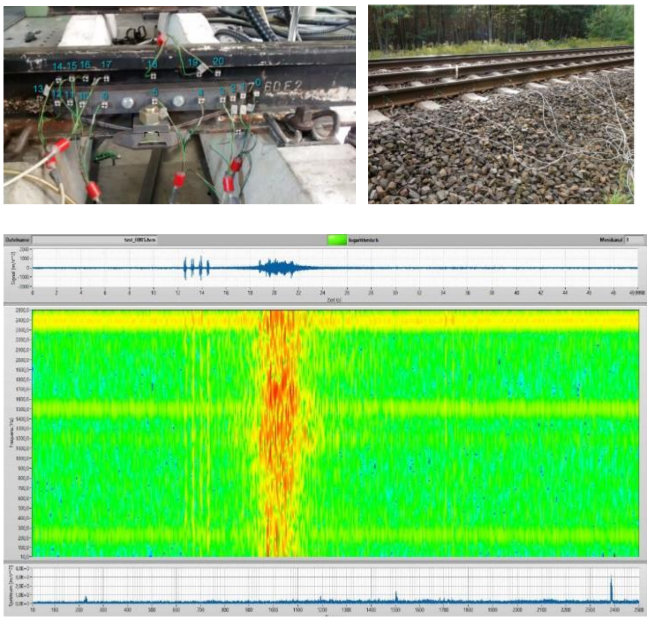


Fig. 10 *Measurements of excitation on a rail track*

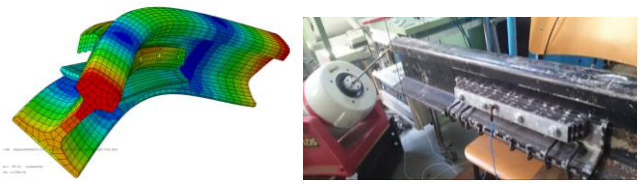


Fig. 11 *Rail damper*

5 CONCLUSIONS

Improvements in transport solutions demand a multilevel approach. Improvements are possible by: application of novel structural materials; experiments to obtain reliable structural properties and as a support to modern methods to monitor their condition; appropriate modeling that enables reliable and highly efficient simulations; and optimization of existing structural designs or innovative designs with novel features, such as adaptive behavior. An overview of the work in all these fields conducted at the Department of Structural and Computational Mechanics of TU Berlin has been briefly addressed in this paper. The overview is certainly not exhaustive, but rather representative as there is an ongoing research in all these fields and new ideas are brought to life practically on daily basis.

REFERENCES

1. Rademacher, T., Zehn, M., 2016, *Modal triggered nonlinearities for damage localization in thin walled FRC structures – a numerical study*, Facta Universitatis series Mechanical Engineering, 14(1), pp. 21-36
2. Marinković, D., Milić, P., Marinković, Z., 2013, *Combined linear – geometrically nonlinear FEM modelling for highly efficient dynamical simulations*, Facta Universitatis series Automatic Control and Robotics, 12(1), 9-18.
3. Marinković, D., Zehn, M., 2012, *Modal space based solutions including geometric nonlinearities for flexible MBS*, Proceedings of the 8th International Conference on Engineering Computational Technology, ECT2012, Dubrovnik, Croatia.
4. Marinković, D., Zehn, M., 2012, *Finite element formulations for effective computations of geometrically nonlinear deformations*, Advances in Engineering Software, 50, pp. 3-11.
5. F. Wesolowski, M. Zehn, 2015, *Application of Krylov subspace reduction methods in frequency based substructuring and an extension of the Lagrange multiplier assembling procedure*, Proceedings of NAFEMS Seminar: "Practical Aspects of Structural Dynamics, Vibration, Damping", Wiesbaden, Germany.
6. Jetmundsen, B., 1986, *On frequency domain methodologies for structural modification and subsystem synthesis*, PhD thesis, Rensselaer Polytechnic Institute, Troy, NY.
7. de Klerk, J. d. J. D., Rixen, D.J., 2006, *The frequency based substructuring (FBS) method reformulated according to the dual domain decomposition method*, Technical report, TU Delft.
8. Marinković, D., Zehn, M., 2012, *Finite element formulations for effective computations of geometrically nonlinear deformations*, Advances in Engineering Software, 50, pp. 3-11.
9. Rama, G., Marinković, D., Zehn, M., 2017, *Efficient 3-node finite shell element for linear and geometrically nonlinear analysis of piezoelectric laminated structures*, Journal of Intelligent Material Systems and Structures, accepted for publishing

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