A NUMERICAL MODEL FOR VIBRATION ANALYSES OF AN AIRCRAFT PARTITION WITH PARAMETERIZED INTERFACE PROPERTIES

Philipp Hüttich[a], Emil Heyden[a], Dieter Krause[a]

a: Institute of Product Development and Mechanical Engineering Design (PKT), Hamburg University of Technology (TUHH), Germany – philipp.huettich@tuhh.de

Abstract: *Aircraft structures are subjected to environmental conditions that can cause critical damage. The vibrational behavior of cabin structures can be influenced by adjusting the mechanical properties at the connections. Many possible mechanical properties lead to a large number of necessary investigations. Therefore, a dynamic model of a partition with parameterized interface properties in frequency domain is presented. Physical tests are performed on a vibration test rig to determine the internal damping of the structure. The vibration amplification depending on different interface properties is determined by amplification functions over several resonance frequencies. The resulting behavior is analyzed and its vibration reduction over all interface properties is determined. With the model presented, the possibility of reducing vibrations through adjustable impedance elements can be predicted. Requirements for physical examinations and properties for vibration-optimized connections in aircrafts can be derived from this.*

Keywords: parameterized simulation; dynamic analyses; lightweight structure; environmental conditions; compliant elements

1. Motivation and Introduction

Products in the aviation industry are subjected to a number of environmental conditions, such as static and dynamic loads, pressure changes, temperature and humidity. Among others, vibration loads are of particular importance, as they introduce high amounts of energy into the structure and can cause critical damage when resonance is reached [1]. Cabin structures, such as the partitions of an aircraft, are sensitive to low-frequency vibrations because they consist of panel-like structures whose elasticity is comparatively low due to their small thickness compared to other dimensions. The vibration behavior can be influenced by adjusting the mechanical properties at the connections [2, 3]. The holistic modeling of the test object and its connections are necessary for the optimal selection of suitable mechanical properties. Many possible mechanical properties such as changing stiffness, damping and inertia at each connection, as well as a large number of possible product variants, lead to a large number of necessary investigations [4, 5]. A dynamic FEM model with parameterized interface properties is a suitable option to proceed efficiently in terms of time and costs. This paper presents a numerical model of an aircraft structure with parameterized interface properties. The dynamic model of the partition with detailed material properties and degrees of freedom is modeled in the frequency domain. A particular challenge is the determination of the inner damping of the test object [4], the modeling of the boundary conditions [6], and the systematic evaluation of the large quantities of simulation results.

 ©2022 Hüttich et al. doi:10.5075/epfl-298799_978-2-9701614-0-0 published under CC BY-NC 4.0 license

2. Methods and development of the model

Numerical models, especially finite element method (FEM) models, can be used to implement complicated geometries, combined load cases, changing boundary conditions, and different material laws [7, 8].

Abaqus brings, in addition to the powerful general-purpose program, advanced theories of nonlinear behavior and a programmable interface, which is why it is most commonly used in research [7]. Abaqus Scripting Interface offers a user-defined extension via the Python programming language with the help of which the possibility exists to create, modify and run models [9]. Due to these given possibilities, the process of creating a large number of models can be automated [9]. This brings a variety of advantages for the system investigation.

- The possibility of parameterization by modifying boundary conditions, materials, geometries, loads, etc. [10].
- Time-efficient creation of a large number of models [11].
- Automated documentation and accompanying evaluation [10], [11].
- Avoidance of failures when defining the respective parameters by the user.

3.1 Definition of the modelled system

Figure 1 a) shows the partition under investigation on the vibration test rig (details in [5]). The dimensions of the test object are defined in figure 1 d). Aircraft partitions are typical lightweight structures with a sandwich structure of GFRP face sheets and a honeycomb core. Due to their large dimensions compared to the thickness, these products are particularly sensitive to low-frequency vibrations. Table 1 shows the composite layers of the partition and each layer thickness. These layer properties can be assigned to the model with the Abaqus composite tool [10]. The material properties of each layer are specified by the manufacturer.

Table 1: Layer structure and thickness of the partition according to manufacturer's specifications.

Ply Name	Material	Thickness [mm]
Face Sheet 1: PHG600-44-50-01	PHG600-44-50-01	0.09
Face Sheet 2: PHG600-68-50-01	PHG600-68-50-01	0.19
Core	WEB48N32	25
Face Sheet 3: PHG600-68-50-01	PHG600-68-50-01	0.19
Face Sheet 4: PHG600-44-50-01	PHG600-44-50-01	0.09

A plane stress state is suitable for a model that is mainly loaded in bending. As long as the sandwich structure is excited in the linear range, shell models offer a suitable computationally efficient alternative [12]. The mesh size in plane direction is chosen to be 10 mm, since at this size it has a negligible effect on the results.

Furthermore, the inserts for adding attachments must be considered in the model. While the real partition has a mass of 9.9 kg, the simulation model without the insert weight has a mass of 5.98 kg. This corresponds to a mass difference of 3.92 kg, this mass is distributed evenly over all 77 inserts on the partition. The partition is connected at four interfaces (shown in figure 1 a, b and c). The partition is rigidly connected to the excitation point in the area of the support and can rotate together centered around this point. Their masses must be taken into account in the model ($m_{C1} = 0.89\ \mathrm{kg}; m_{C2} = 0.87\ \mathrm{kg}; m_{C3} = 1.5\ \mathrm{kg}; m_{C4} = 1.35\ \mathrm{kg}$), they are evenly distributed among all connected nodes (highlighted in figure 1 d and e).

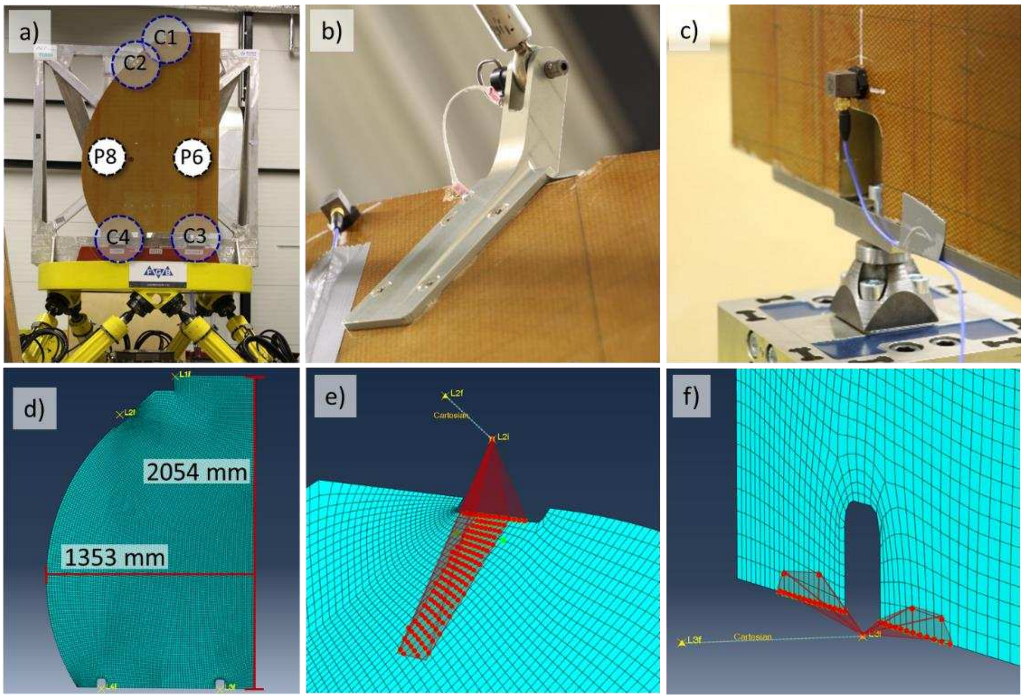

Figure 1. a-c) physical test setup of the partition; d-f) numerical implementation of the partition and its connections.

3.2 Definition of system characteristics

In reality, a wide variety of damping mechanisms occur, which cannot all be transferred separately to the simulation. Abaqus provides the value of the structural damping for this purpose, which must be determined individually for each case. To define the system properties such as the structural damping, a physical test must be carried out.

At the lower connections C3 and C4, the partition is attached to force transducers by a ball bearing (figure 1 c). The ball bearing is the original mounting form in the aircraft and prevents moment transmission in all rotational axes. The connections at the upper connections C1 and C2 to the partition are realized by pendulums with ball joints at each end. They are supported with locking bolts (figure 1 b). Original aluminium adapter plates are attached to the partition by means of screw connections for the two upper connections. To record test results accelerometers (3D 500g 356A02, PCB Piezotronics Inc., Depew NY, USA) are mounted at the test rig itself as well as on several positions of the partition.

A typical and in the aircraft certification process often used form of excitation in vibration analysis is the sweep excitation, in which the frequency of a sinusoidal oscillation increases over time [13, 14]. To be able to compare experimental and simulation data, sweep tests are performed at a constant acceleration amplitude of 3 m/s^2 at a constantly rising frequency from 3 to 23 Hz. From the recorded data, the excitation of the different measurement points in the frequency domain is determined using the Fast Fourier Transform and normalized by the reference signal, resulting in the amplification of each point.

In the frequency range from 3-23 Hz two resonance of the partition result. Among others, sensors are placed at points P6 and P8 (figure 1 a), since the movement of these points are representative for the two occurring resonances. The maximum amplification of the first resonance is 48 at point P6. Point P8 is amplified by a factor of 14 at the first resonance (see Figure 2a). To adapt the simulation model to the experimental data with respect to the amplification factor, the damping coefficient of the simulation model is approximated. The structural damping is set, so that the amplification level at the first resonance of the simulation and experimental data are the same (see Figure 2b below). Resulting, a structural damping coefficient of $s_d = 0.036$ is obtained and used for the further investigations of the partition.

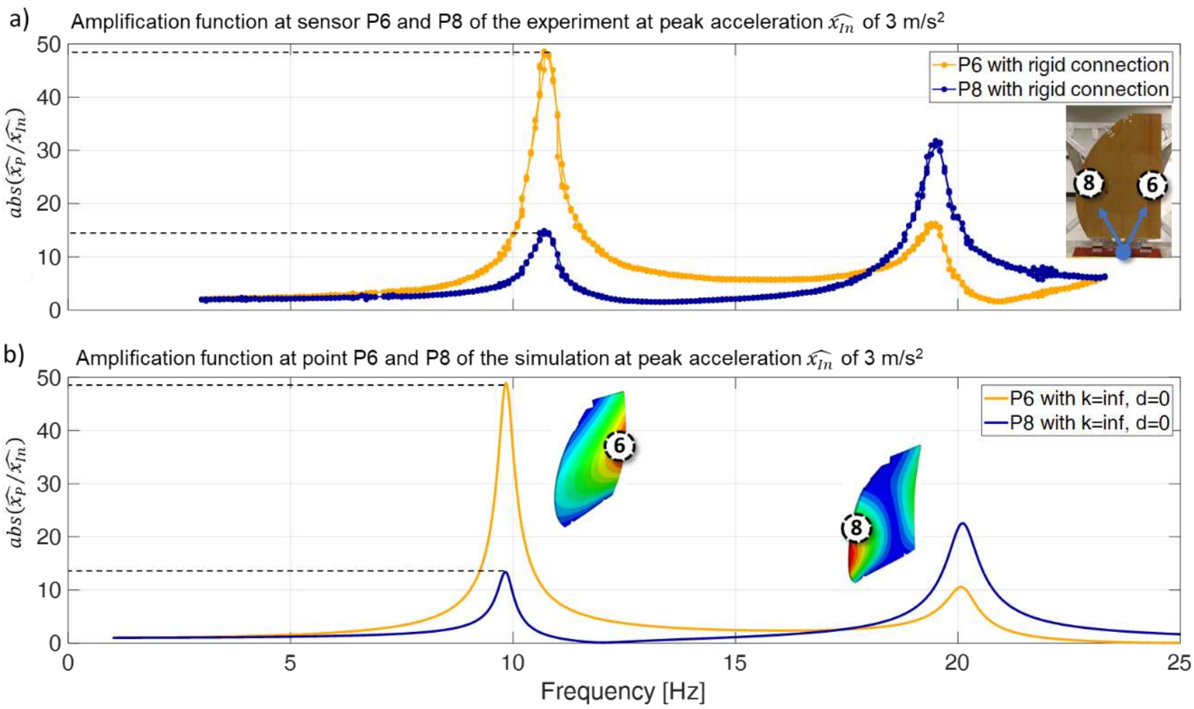

Figure 2. Comparison of the amplification function of the experiment and the simulation model to determine the structural damping.

3.3 Define the excitation and adding parameterized interfaces

The partition is excited by a base motion through its connection elements. For the definition of the base motion in Abaqus it is valid that it acts on the respective blocked degrees of freedom in the given excitation direction of the motion [15]. If the excitations at the bearing points are not all the same, so-called secondary base motions can be defined [15]. The excitation is applied at the four connection points in accordance with the test setup. A sweep movement is applied orthogonal to the partition at the end of the four connections.

Parameterized numerical models are useful when analyzing a large number of factors with many levels to obtain an understanding of the overall system [8]. To investigate the effect of compliant connections on the vibration behavior of the partition, wire connections are added between the center point of the connector and the point of excitation. The mechanical properties stiffness and damping are added to these wire connections [10]. Depending on the simulation case, the properties can be systematically varied and adapted using the Python script.

In the case of rigid connections, the mass of the interface elements themselves is not crucial, since the entire interface follows the defined motion in the model. In the case of compliant interfaces, the movement before and after the connection can differ, therefore it is necessary to determine the weight of the connections themselves, as these have an influence on the vibration behavior.

4. Results and Discussion

The amplification resulting from the excitation vibration depends on different interface properties and is determined by amplification functions over several resonance frequencies. It is necessary to distinguish between two objectives, the reduction of the vibration of a given point on the partition and the reduction of the maximum vibrating point on the entire partition.

For the following investigations all four connections have the same properties. A combination among each other does not take place. For the given case of the aircraft partition, a range of 1-500 N/mm was selected for the stiffness and 0.1-5 Ns/mm for the damping.

4.1 Amplification at specific points

By varying the stiffness and damping parameters, the amplification function can be controlled and consequently the maximum amplification can be reduced. Figure 3 shows the resulting amplification function for different values of the stiffness k and damping d.

The vibration behavior of point P6 is shown in figure 3 a) and is particularly affected by the first resonance. At this point a reduction of the amplification from 48 to 19.9 (reduced by 58%) is possible by changing the stiffness from rigid to k = 100 N/mm and the damping to d=1Ns/mm. While the resonant frequency changes only slightly from 9.8 to 9.4. The amplification of point P8 is shown in figure 3 b). The second resonance is more sensitive to the interface adjustment than the behavior at the first resonance. The amplification can be reduced from 22.1 to 2.3 (reduced by 89 %) and the resonnance frequency from 20.1 to 18.35 by adjusting to k = 200 N/mm and d = 4 Ns/mm. An investigation of the vibration behavior with a few settings of the interface is not sufficient to provide an understanding of the overall vibration behavior.

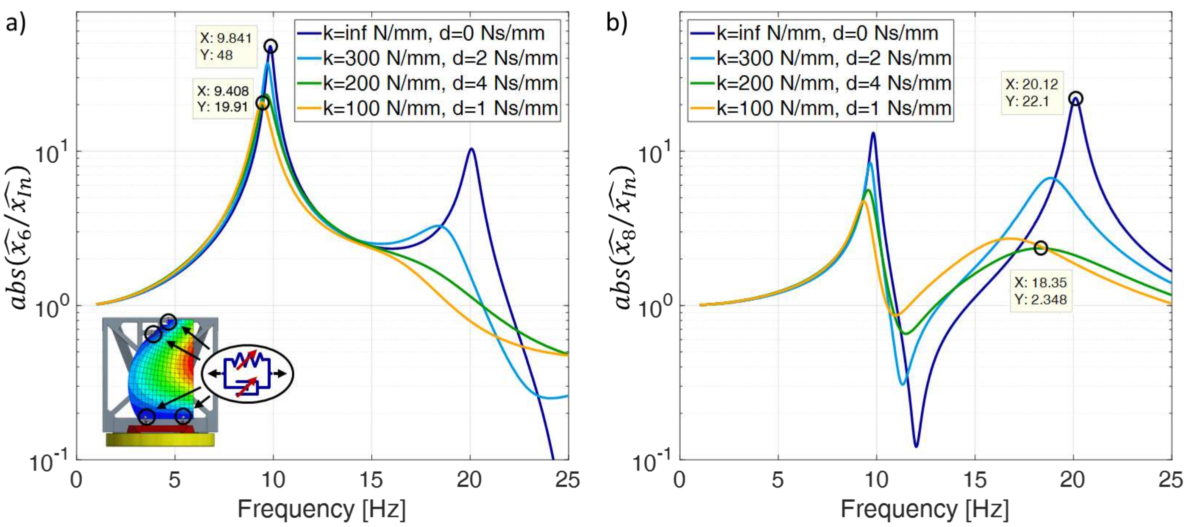

Figure 3. Amplification function of the simulation model with different interface stiffness and damping at Point 6 and 8.

4.2 The maximal amplification over the entire system

An evaluation of the vibration reduction at a specific point is deceptive, since the variation of the interface characteristics can cause the point of maximum gain to drift. To determine the maximum amplification of the system per interface properties, the movement at all points of the partition $\widehat{x_i}$ is to be determined, and the resulting maximum movement $\widehat{x_m}$ to evaluate. Figure 4 a) shows the resulting amplification at the maximal moving point for different stiffness and damping adjustments. At the kink of the function, the point of maximum amplification jumps from the right side of the partition to the left as shown by the path of the maximal moving point in red. Representable the setting from rigid to k = 200 N/mm and d = 4 Ns/mm leads to a reduction of the amplification from 50.7 to 20.8 (reduced by 59%).

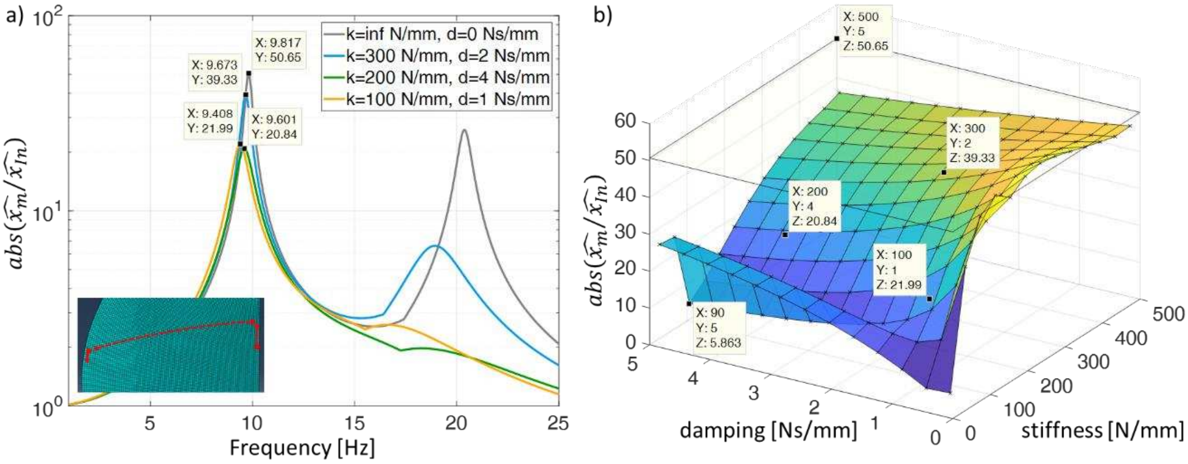

Figure 4. Amplification function of the model at the maximum point of vibration for different stiffness and damping parameters.

Figure 4 b) presents the maximum resonance of the point with the highest movement on the partition over the setting of damping and stiffness of the interfaces. For a stiffness of 90 N/mm and a damping of 5 Ns/mm, the minimum of the curve is obtained. This leads to a reduction of the resonance from the rigid connection at 50.65 to the compliant connection of up to 5.9, which corresponds to a reduction of up to 88%. However, it should be noted that this minimum is very sensitive to a change in its parameters, especially the stiffness. Therefore, this reduction will be challenging to reproduce in physical tests.

4.3 Further findings of the vibration behavior of the partition

Requirements for vibration-optimized connections in aircrafts can be derived from this model. This makes it possible to design cabin structures in aviation in such a way that they themselves need to be less resistant to vibrations and therefore allows lighter structures to be developed.

Adjustable impedance elements are machine elements with separately adjustable stiffness and damping characteristics, which are used as interface elements in vibration testing [16]. The model can be used to derive requirements for these elements to demonstrate the resulting reduction in physical vibration testing. From the numerical model, necessary requirements for interface forces and displacement amplitudes can be derived in addition to the amplification function discussed in this publication.

Depending on the two resonances and corresponding mode shapes investigated, different interface properties can lead to improved vibration reduction. A representative example is provided by the amplification function at point P8 shown in figure 5 a). For the first resonance setting 1 lead to a gain of 5.6 and setting 2 to a gain of 4.7 (reduction from setting 1 to 2 by 16%). For the second resonance, setting 2 leads to a gain of 2.7 and setting 1 to a gain of 2.3 (reduction from 2 to 1 by 15%). Dynamically adaptive interface elements could adjust during operation and provide suitable interface characteristics depending on the excitation frequency. This would lead to further vibration reduction over several resonant frequencies.

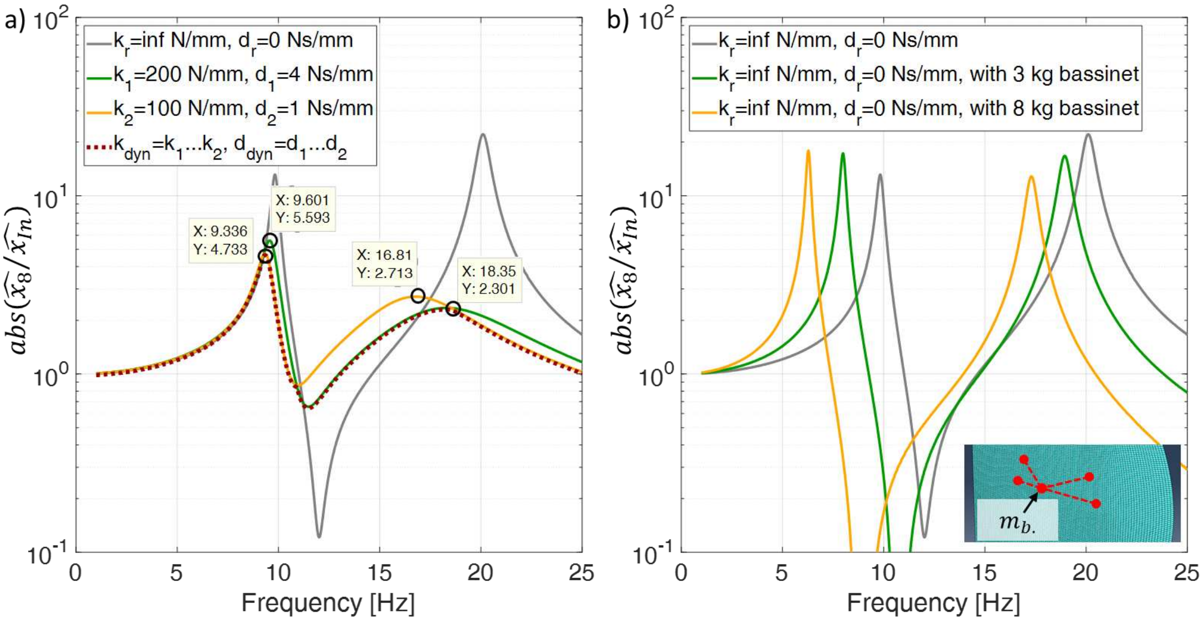

Figure 5. Amplification function at Point P8 a) for stationary adjustable interfaces and dynamical adaptive interfaces and b) for different product variants.

In addition, attachments can change the characteristics of the system during operation. As an example, Figure 5 b) shows the amplification function with an attached baby bassinet (simplified as point mass $m_{b.}$ and ignored change of internal damping). The increase in mass leads to an increase of the first resonance and a reduction of the second resonance. It is expected that dynamically adaptive interface properties lead to a better reduction of vibration per configuration, since they can be optimized for each resulting resonance separately.

5. Conclusion

It can be demonstrated via the numerical model presented that significant vibration reduction of the aircraft partition can be achieved by using compliant boundary conditions. Moreover, the results show that the vibration is sensitive to a change in its interface parameters.

In vibration reduction, it is necessary to examine every point on the object, since the point of maximum motion drifts. This, together with the high number of simulations in a parameter study, leads to a high computational effort and memory consumption and the need for automated algorithms and analysis.

Requirements for adjustable impedance elements can be derived from this model. These are necessary to represent the vibration behavior of test objects with varying boundary conditions in physical tests. In order to achieve different vibration reductions for different vibration modes, dynamically adaptive impedance elements are necessary, that have the ability to adjust during operation.

This publication shows the influence of a parameterization of the interfaces, comparable studies with other environmental conditions are useful to better understand the behavior of the system under investigation.

Acknowledgements

Special thanks to Ibrahim Kassem, who supported the presented simulation model through many debates and by his student research.

The research results on which this publication is based are part of the project AIProVE that was funded by DFG (Deutsche Forschungsgemeinschaft) and the project CERTEV that is funded by the BMWK (Bundesministerium für Wirtschaft und Klimaschutz). The statements and information in this contribution do not necessarily represent the opinion of BMWK and DFG.

6. References

1. Rasmussen, O.; Krause, D.: Influence of Load Elements on the Dynamic Behaviour of Lightweight. Proceedings of the 6th International Workshop on Aircraft System Technologies, Shaker Verlag, Aachen, 2017.
2. Kim, J. Y. et al.: An Experimental Approach for Structural Dynamic Modification of Fixture in Vibration Test Control. JSME International Journal Series C, Vol. 442, 2001, pp. 334–340. https://doi.org/10.1299/jsmec.44.334
3. Heyden, E. et al.: Approach for Calibrated Measurement of the Frequency Response for Characterization of Compliant Interface Elements on Vibration Test Benches. Applied Sciences, 2021. https://doi.org/10.3390/app11209604
4. Seemann, R. et al.: FE-Modelling Guidelines for the Dimensioning of Aircraft Cabin Interior under Stationary Dynamic Loads. 29th Congress of the International Council of Aeronautical Sciences, 2014.
5. Lindenmann, A. et al.: Adjustable Impedance Elements for Testing and Validation of Aircraft Components and Hand-Held Power Tools. Stuttgarter Symposium für Produktentwicklung, Stuttgart, 2019.
6. Heyden, E. et al.: Transferability of Boundary Conditions in Testing and Validation of Lightweight Structures. Proceedings of the 30th Symposium Design for X, Hamburg, 2019. https://doi.org/10.35199/dfx2019.8
7. Klein, B.: FEM - Grundlagen und Anwendungen der Finite-Element-Methode im Maschinen- und Fahrzeugbau. Springer Vieweg, Lehrbuch, Wiesbaden, 10. Auflage, 2015.
8. Reddy, J. N.: Introduction to the finite element method. McGraw Hill Education, Fourth edition, New York, 2019.
9. Dassault Systèmes Hrsg.: Abaqus Skripting Guide. 2014.
10. Dassault Systèmes Hrsg.: Abaqus Analysis User's Guide. 2014.
11. Puri, G.: Python scripts for Abaqus - Learn by example. Charleston, SC. 1. ed., 2011.
12. Salerno, G. et al.: Reduced Order Modeling of Composite Laminates Through Solid-Shell Coupling. Journal of Aerospace Technology and Management, 2017, 9; pp. 397–403. http://dx.doi.org/10.5028/jatm.v9i3.742
13. Ewins, D. J.: Modal Testing: Theory and Practice. Volume 2, Letchworth, 1986.
14. RTCA, Inc.: RTCA/DO160G – Environmental Conditions and Test Procedures for Airborne Equipment. Washington, 2010.
15. Dassault Systèmes Hrsg.: Abaqus Theory Guide. 2016.
16. Heyden et al.: Adjustable Impedance Elements for Testing and Validation of System Components. Symposium Lightweight Design in Product Development, Zürich, 2018, pp. 45-47. 10.3929/ethz-b-000283432