Demonstrating Vibration Control of Lightweight Structures by Adjustable Impedance Elements

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Abstract

The mechanical interface properties between a lightweight structure and its surroundings are a key factor in vibration reduction. This paper demonstrates the principle of vibration reduction in lightweight structures by using Adjustable Impedance Elements (AIEs). AIEs enable the interface stiffness and damping to be adjusted over a wide range. Parameter studies are conducted to identify promising combinations of interface stiffness and damping using FEM. With optimal settings of the AIE a vibration reduction of 63% compared to a rigid interface is reached. However, the vibration response is effectively decreased only within a limited frequency range. Thus, the need for dynamically adaptive interface properties is deduced.

Keywords

Vibration Testing, Vibration Reduction, Adjustable Impedance Elements (AIE), Parameterized FEM, Lightweight Structures

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1. Introduction

Diverse operational scenarios of aircraft lead to a variety of static and dynamic load cases acting on aircraft structures. One example is the windmilling event, which arises from sustained engine imbalance (SEI) after the loss of an engine blade [1], [2]. The engine is shut down, but continues to rotate due to the airflow induced by the aircraft's flight speed [1], [2]. The vibrations introduced into the entire aircraft structure are of substantially large amplitude at low frequencies, typical around 17 Hz [2]. As a result, interior monuments located in the cabin are subjected to dynamic loads that can end in a major vibration amplification, if the excitation matches a resonance frequency of one of the interior monuments.

An approach for minimizing this risk is a vibration reduction of the interior monuments [3]. The dynamic behaviour of lightweight structures is highly susceptible to changes in the mechanical properties of their interfaces to the surrounding structures [4], [5], [6]. Finding a proper combination of interface stiffness and damping for each lightweight structure is consequently a promising approach towards a substantial vibration reduction [3]. This regularly demands a large number of tests, since empirical analysis is indispensable. Interface elements like *Adjustable Impedance Elements* (AIE), which allow interface stiffness and damping to be set over a wide range of values, can reduce experimental effort [7].

1.1. Aim of the Contribution

In this contribution, a demonstrator for the principle of vibration reduction of lightweight structures by optimizing the interface stiffness and damping through AIE is presented. Furthermore, the aim is to determine whether there is a need for non-stationary interface properties in the field of vibration reduction in lightweight structures.

Research Question: Are AIE sufficient for the experimental analysis of the vibration reduction of lightweight structures or is there a need for interface elements with dynamically adaptive impedance?

Hypothesis: If there is more than one resonance of a product relevant in the desired frequency range of its application, different values of the interface stiffness and damping are necessary for an optimal vibration reduction over the entire frequency range.

1.2. State of the Art

The need for vibration suppression is relevant in various fields of engineering. While a number of publications focus on vehicle dynamics [8], [9], [10], vibration isolation of machinery [11], [12] and civil engineering structures, such as bridges and buildings [13], [14], [15], [16], few publications address vibration reduction in lightweight structures.

SEEMANN ET AL. [6] have examined the current FEM modelling approaches for dynamically loaded aircraft interior monuments and have compared them to vibration test data. Among other findings, they have concluded that the dynamic behaviour of the FEM model is heavily influenced by the mechanical interface properties [6]. RASMUSSEN and KRAUSE [17] have investigated the influence of rubber mounts at the interfaces of aircraft interior monuments on their vibration response and have achieved a relevant vibration reduction. Furthermore, they have found that friction in additional elements attached to the monuments, such as literature pockets, is also a key factor in vibration reduction [17]. OLTMANN ET AL. [18], [19] have proposed a particle damper to be added to lightweight structures to decrease their amplification. The particle damper has been mounted to the top of a GFRP sandwich panel with an excitation at the bottom and has resulted in a decline of the resonance as well as a shift to lower frequencies [18], [19]. LIU ET AL. [20] have analysed the vibration reduction of an aluminium plate added to a frame made of aluminium sandwich by means of three different damping concepts: Interface damping, active constrained layer damping and particle damping. Interface damping has shown a broad effect on all modes of the plate and renders the lowest average amplitude, while active constrained layer damping and particle damping show mixed effectiveness on different modes and surpass interface damping only for some modes [20]. HÜTTICH ET AL. [21] have presented a parameterized FEM model of an aircraft partition, which allows for the mechanical interface properties to be systematically and efficiently varied. The aim of their model is to aid in the determination of proper interface stiffness and damping values in order to achieve a large vibration reduction [21]. HEYDEN ET AL. [22] have shown the vibration reduction of an aircraft partition by 65% experimentally and 62% by simulation compared to the original interfaces when using AIE at the upper interfaces. HEYDEN [3] has proposed an approach for the vibration reduction of lightweight structures using AIE as interface elements. The vibrating system of the lightweight structures and the interfaces to the environment are considered holistically for the optimization of the vibrating system [3]. The approach consists of an experimental analysis of the vibration response of the structure with rigid interfaces, the setup of an FEM model, a systematic parameter study of the interface properties using the FEM model to find the combination of stiffness and damping yielding the largest vibration reduction, the selection of a suitable AIE, followed by the experimental measurement of the resulting vibration reduction [3]. Except for LIU ET AL. [20], the mentioned publications that focus on an experimentally proven decrease in vibrations in lightweight structures consider means of vibration reduction with stationary properties during operation. In contrast, LIU ET AL. [20] do not consider GFRP sandwich material, which is primarily used in aircraft interior monuments.

2. Methods and Materials

The principle of vibration reduction of lightweight structures by AIE is to be demonstrated and its potential and limitations are to be investigated. To achieve this, experimental analyses as well as FEM simulations are to be conducted. Genuine aircraft interior monuments typically contain a variety of inserts, fillings or other local reinforcements. These complicate the modelling process because they have a strong effect on the dynamic behaviour of the monument [6] and can lead to unnecessary confounding variables, which prevent the vibration reduction by AIE from being investigated in isolation. To enable this, the real aircraft interior monuments are represented by a simplified demonstrator in this contribution.

2.1. Experimental Setup of the Demonstrator

The experimental setup of the demonstrator is shown in Figure 1. The lightweight structure is represented by a rectangular-shaped aviation-grade sandwich panel from Euro-Composites® S. A. featuring a Nomex® honeycomb core and GFRP face sheets (Figure 1e) and Table 1). Two variants of the demonstrator are analysed, one with an additional mass at the top of the sandwich panel ($m_{Add} = 0.1\ \mathrm{kg}$) and one without ($m_{Add} = 0\ \mathrm{kg}$). The sandwich panel is connected to the testing environment by two interfaces. The upper interface is designed as a pendulum support (similar to a tie rod, but for both tensile and compressive forces) to provide only a connection and transmission of force in one degree of freedom (DOF, z-direction in Figure 1a) without constraining the other five DOF. The lower interface comprises the AIE and a linear friction bearing, which is rigidly clamped to the sandwich panel. Relative motion of the lower edge of the sandwich panel is therefore only permitted in z-direction with respect to the surrounding testing environment. The AIE is shown in Figure 1b and allows for an independent setting of stiffness (S_k) and damping (S_d). For the specifications of the AIE used, please refer to [23] and for more information on the working principles, please refer to [24], [25]. As a reference measurement for the vibration reduction achieved by the AIE, the lower interface can be fixed rigidly to prevent a relative motion (Figure 1c). The demonstrator is equipped with one force sensor (S9M/10kN, Hottinger Brüel and Kjaer (HBK) GmbH, Darmstadt, Germany) to measure the interface force at the AIE and six accelerometers (3D 500g TLD356A02, PCB Piezotronics Inc., Depew, NY, USA). The sensor positions are depicted in Figure 1d and 1f. The mechanical model of the demonstrator is presented in Figure 1f and 2.

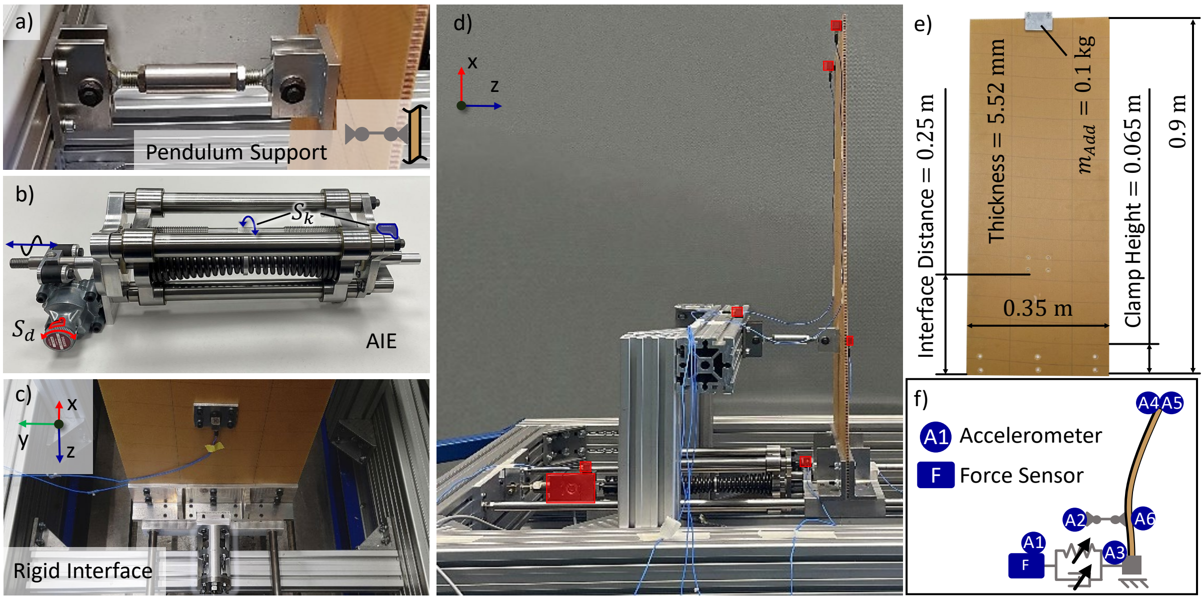

Figure 1: a) Pendulum support of the upper interface, b) AIE of the lower interface, c) rigid connection of the lower interface, d) and f) demonstrator with the sensors, e) specimen consisting of a sandwich panel and an additional mass, see also [3]

The vibration excitation of the testing environment is carried out as a translational sine sweep in z-direction (Figure 1d) at a constant acceleration amplitude of $\hat{a}_{A1} = 14\ \mathrm{m/s^2}$ and a frequency range from 3 to $23\ \mathrm{Hz}$.

Table 1: Layup of the sandwich panel

Component	Material	Nominal Thickness
Face sheet	1 layer GFRP prepreg (ABS5047-07)	$0.25\ \mathrm{mm}$
Core	Nomex honeycomb ECA 3.2-48 (ABS 5035 A4)	$5\ \mathrm{mm}$
Face sheet	1 layer GFRP prepreg (ABS5047-07)	$0.25\ \mathrm{mm}$

2.2. FEM Model of the Demonstrator

Prior to the experimental analysis a FEM model is set up. The aim is to determine promising combinations of interface stiffness and damping with high potential of vibration reduction by means of a parameter study [3]. This information is needed for deriving requirements for the selection of a suitable AIE and for focusing the experimental analysis on the relevant combinations of interface stiffness and damping [3]. The purpose of the FEM simulations is to support the execution of the experimental analysis and reduce the necessary number of tests, but not to replace the experimental analysis entirely. Accordingly, a suitable balance has to be found between modelling effort and computational cost, on the one hand and accuracy of the simulation results, on the other hand.

The FEM modelling and simulation in this contribution has been performed analogously to [21], [22], [3]. It has been conducted with Abaqus Standard 2021 through the use of a parameterized Python script. This allows for an efficient variation of interface stiffness and damping values [21]. The sandwich has been modelled as orthotropic shell elements of the type S4R with the mechanical properties of the honeycomb core and the face sheets according to SEEMANN [26]. The excitation has been applied via a primary base motion [27] designed to simulate the surrounding testing environment. The principle of modal superposition has been applied to calculate a linearized response. Involving a modal analysis followed by a mode-based steady-state harmonic response analysis. In a direct comparison of the response at a specific point (Figure 3 and 5), the simulation has been evaluated at a sensor position corresponding to that of the experimental setup. However, to analyse the potential of vibration reduction during the parameter study and for determining suitable sensor positions, the point of maximum vibration amplitude has to be considered. The location of this point can change across the frequency range and for different interface properties [21]. Therefore, the node with the highest acceleration amplitude is automatically evaluated at every increment during the simulation, which is described in more detail by HÜTTICH ET AL. [21]. More information regarding the integration of the parameterized FEM simulation into the approach for vibration reduction of lightweight structures is given by HEYDEN [3].

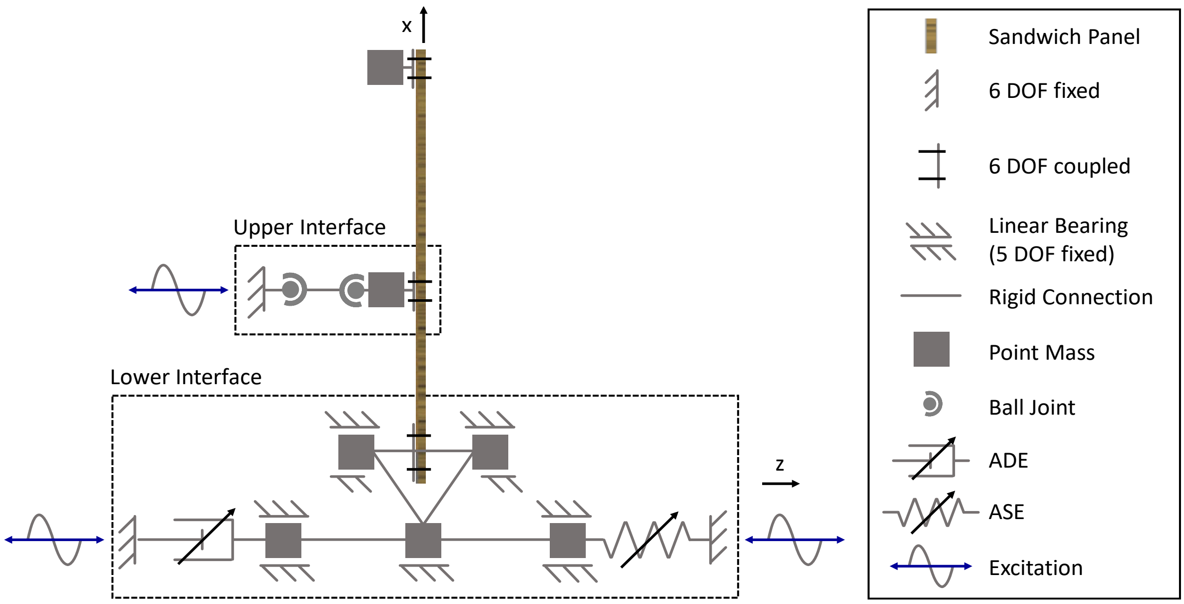

Figure 2: Mechanical model of the demonstrator as implemented in the FEM model

The mechanical model of the demonstrator, as it is implemented in the FEM model, is shown in Figure 2. The demonstrator is constrained to the environment in all six DOF, which acts as the base for the excitation by primary base motion. This corresponds to the testing environment in the experimental setup. The friction of the linear bearings is neglected. As suggested by SEEMANN ET AL. [6], the material damping of the sandwich panel as well as the damping at its attachments are modelled as structural damping. The structural damping values have been determined by parameter identification (0.105 for the variant with and 0.121 for the variant without the additional mass) from the experimental data with the rigid reference at the lower interface (Figure 1c and 3).

3. Results and Discussion

In Figure 3, the frequency response functions (FRF) of the reference configuration with the rigid connection at the lower interface are shown. The simulation and experimental data are compared for both variants of the demonstrator with an additional mass at the top of the sandwich panel of $m_{Add} = 0.1\ \mathrm{kg}$ and without an additional mass. The largest displacement of the structure during resonance occurs at the top of the sandwich panel. Hence, the complex amplification FRF

$$HI_{input \to output} = \frac{a_{output}}{a_{input}};\ HI_{A1 \to A4A5} = \frac{a_{A4A5}}{a_{A1}}, \tag{1}$$

is considered, relating the FOURIER-transformed input acceleration a_{A1} at sensor A1 and the output acceleration a_{A4A5} at the sensors A4 and A5. For both variants, the simulation and experimental data match well in the amplitude and the phase FRF. Therefore, the FEM model provides a suitable representation of the experimental data in the case of the reference configuration with the rigid lower interface. One pronounced resonance is formed within the analysed frequency range, which expectedly emerges at a lower frequency for the variant with the additional mass. The standard deviations of the largest amplitude in the experimental data are small for both variants (0.13 for the variant with the additional mass of $m_{Add} = 0.1\ \mathrm{kg}$ and 0.39 for the variant without the additional mass). Additionally, the evaluation of the other accelerometers at different locations of the demonstrator (Figure 1d and 1f) during the reference tests indicates effective transmission of excitation to the test object, without undesired amplification due to resonances of components of the testing environment.

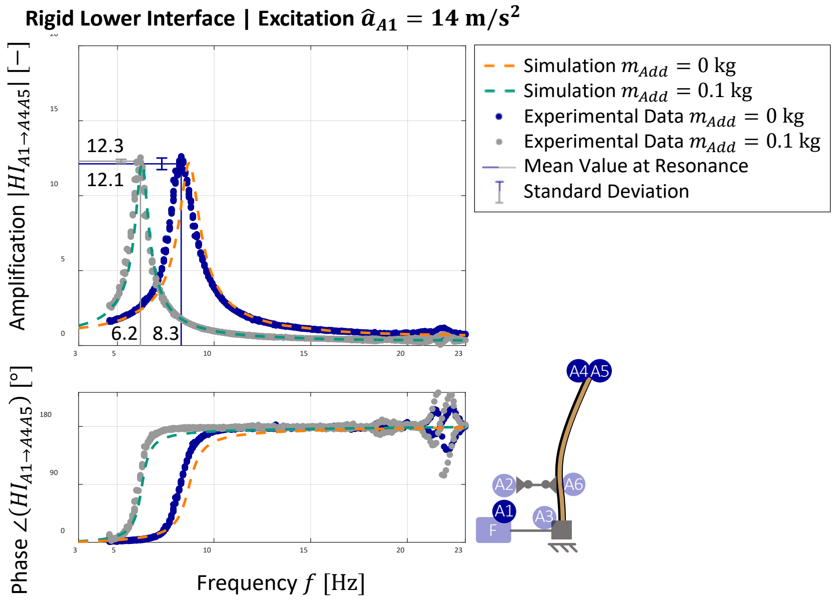

Figure 3: Experimental and simulated reference FRFs of the demonstrator with and without the additional mass, see also [3]

The parameter studies on interface stiffness and damping are depicted in Figure 4 for both variants of the demonstrator with and without the additional mass at the top. For reference, the amplification factor at the resonance with the rigid lower interface (from Figure 3) is visualized as a grey plane. The simulated parameter studies form a distinct minimum at $k_{opt,R1,sim} = 70\ \mathrm{N/mm}$ and $d_{opt,R1,sim} = 0.05\ \mathrm{Ns/mm}$ for the variant with the additional mass of $m_{Add} = 0.1\ \mathrm{kg}$ and at $k_{opt,R1,sim} = 130\ \mathrm{N/mm}$ and $d_{opt,R1,sim} = 0.1\ \mathrm{Ns/mm}$ for the variant without the additional mass. These combinations of interface stiffness and damping are most promising in pursuing a substantial vibration reduction during the physical tests and thus have to be regarded as requirements for the selection of a suitable AIE [3]. For larger stiffness and damping values, an increase of the amplification factor is observed, but it remains under the reference plane and so still yields a vibration reduction. However, lower stiffness values may even lead to a rise of the amplification factor beyond the reference plane and result in a vibration enhancement rather than a reduction. The experimental data from the physical tests with the AIE are depicted in Figure 4 as well. For the variant with the additional mass, the entire grid of tested stiffness and damping values has been carried out with three repetitions, permitting calculation of standard deviations for each data point. In contrast, the variant without the additional mass has been tested mostly with single measurements across the grid. Only the stiffness value experimentally yielding the best vibration reduction has been tested with three repetitions (standard deviations only at $k_{opt,R1} = 117\ \mathrm{N/mm}$). The test data qualitatively match the simulation. The main goal of the simulation, to aid in the selection of a proper AIE for the physical tests, is thus fulfilled. Quantitatively, deviations in the size of the amplification factor and vibration reduction between experimental data and simulation as well as a slight shift of the optimal stiffness and damping values for the experimentally largest vibration reduction are observed. These are $k_{opt,R1,exp} = 62\ \mathrm{N/mm}$ and $d_{opt,R1,exp} = 0.07\ \mathrm{Ns/mm}$ for the variant with the additional mass of $m_{Add} = 0.1\ \mathrm{kg}$ and $k_{opt,R1,exp} = 117\ \mathrm{N/mm}$ and $d_{opt,R1,exp} = 0.07\ \mathrm{Ns/mm}$ for the variant without the additional mass.

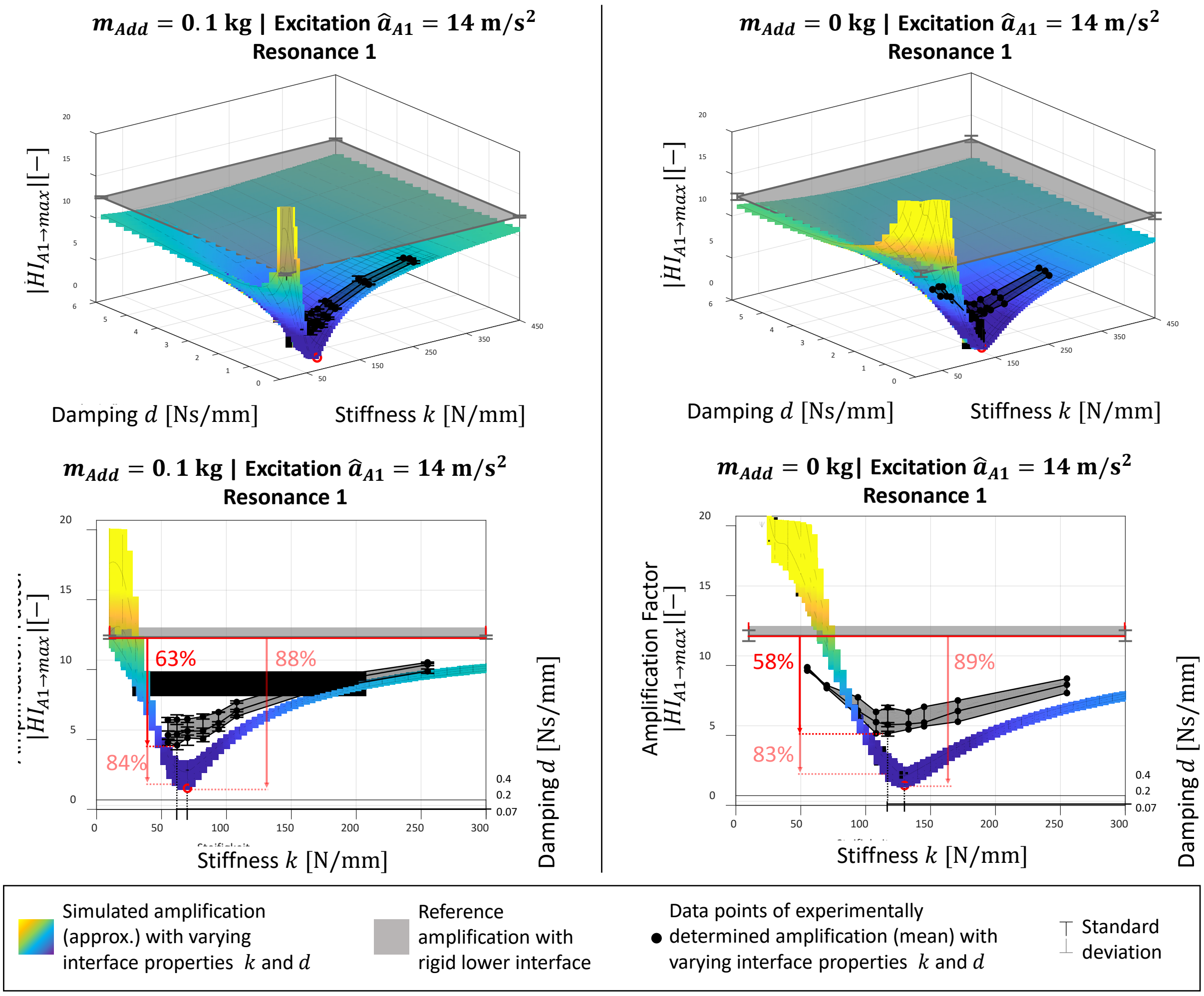

Figure 4: Parameter study of the vibration reduction at the first resonance for varying interface stiffness and damping values for the variant with (left) and without (right) the additional mass at the top of the sandwich panel, see also [3]

In Figure 5, the FRFs with the rigid reference at the lower interface are compared to the experimental data with the optimal settings of stiffness and damping of the AIE as well as the simulation obtained by using the same settings. These result in a vibration reduction of 63% for the variant with and 58% for the variant without the additional mass to be experimentally determined at the original resonance [3]. By means of the WELCH t-Test [28], the vibration reduction is additionally proven to be highly significant at a 99.9% confidence interval. The simulations show an even greater potential for vibration reduction of 84% and 83%. The discrepancy can be attributed to friction in the linear friction bearings at the lower interface, which has not been included in the FEM model. The friction of the linear bearings is not included in the structural damping, which has been determined through parameter identification from Figure 3. Since the lower interface has been fixed rigidly for that test, no relative motion and consequently no sliding friction has occurred at the lower interface. A separate measurement of the friction under realistic normal force conditions would have led to a major increase in the experimental effort. For a FEM model with the purpose of reducing the necessary number of tests this would have been neither reasonable nor required. To substantiate the hypothesis that the discrepancy is caused by neglected friction in the FEM model, the interface damping value is increased to $d_{frict,R1} = 0.85\ \mathrm{Ns/mm}$ for the variant with the additional mass and to $d_{frict,R1} = 1.11\ \mathrm{Ns/mm}$ for the variant without the additional mass. These curves match the experimental data in the lower frequency range around the original resonance well and therefore indicate the neglected friction of the linear bearings as the cause of the discrepancy between simulation and experimental data. As expected, the simulated curves with $d_{frict,R1}$ do not match the experimental data at higher frequencies, since the interface damping is of viscous and not of frictional type.

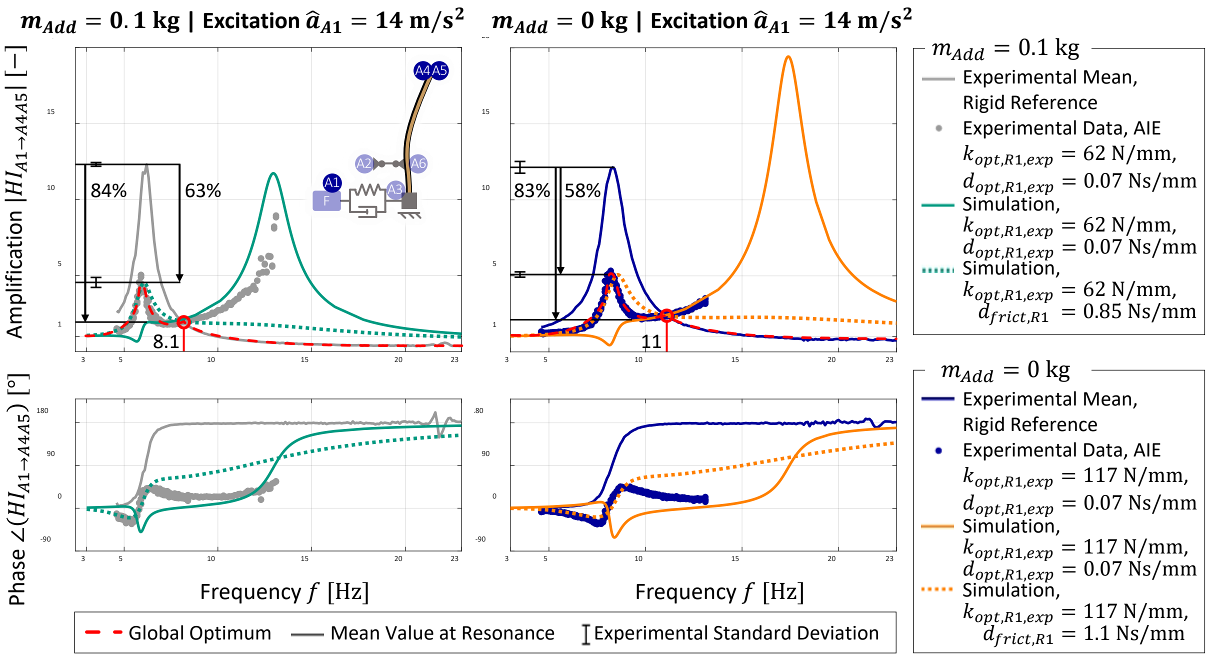

Figure 5: Experimental and simulated FRFs of the demonstrator with (left) and without (right) the additional mass at the top of the sandwich panel

Apart from the effective vibration reduction in the frequency range of the original first resonance of each of the two variants, the interface stiffness and damping settings of $k_{opt,R1,exp}$ and $d_{opt,R1,exp}$ give rise to a newly formed second resonance at higher frequency (Figure 5). There, $k_{opt,R1,exp}$ and $d_{opt,R1,exp}$ result in a vibration enhancement rather than a vibration reduction compared to the rigid reference configuration of the lower interface. This has caused the rods of the linear bearings at the lower interface to oscillate violently and has consequently led to the abortion of the sweep to higher frequencies for safety reasons.

The global vibration optimum over the entire analysed frequency range cannot be achieved by stationary interface properties. A shift of the interface properties at $8.1\ \mathrm{Hz}$ for the variant with and at $11\ \mathrm{Hz}$ for the variant without the additional mass from $k_{opt,R1,exp}$ and $d_{opt,R1,exp}$ to $k \rightarrow$ rigid would render the best vibration reduction.

4. Conclusion and Outlook

In this contribution, the potential for significant vibration reduction of lightweight structures by means of adjustable interface properties using AIE is shown with a simplified demonstrator both experimentally and numerically. By determining the proper stiffness and damping values of the interface between the lightweight structure and the testing environment, a vibration reduction of up to 63% has been experimentally proven compared to the resonance of the rigid reference interface. The advantage of a parameterized FEM model is displayed by its ability to efficiently identify promising combinations of interface stiffness and damping values with a high potential for vibration reduction while minimizing the necessary number of tests [21].

However, the limitations in the use of AIE are also apparent from the results. Although manually adjustable, the interface properties of AIE remain stationary during operation. For some products, including this demonstrator, multi-stationary interface properties are needed when optimal vibration reduction over a wide frequency range is required in the specific application. For other products, an AIE with stationary interface properties might be sufficient when only a narrow frequency range of operation is relevant or when a compromise of the optimal interface properties of different resonances is possible and yields an ample vibration reduction.

Two aspects should be addressed in further research: First, *Dynamically Adaptive Impedance Elements (*DAIE*)* with semi-active properties should be developed and applied to the field of vibration testing of lightweight structures. The capability to adjust the interface properties during operation would enable a vibration reduction of lightweight structures across wide frequency ranges with multiple resonances or under changing load conditions, which require multi-stationary interface properties. Second, to fully analyse the potential of DAIE, the testing environment of demonstrators should be designed even stiffer as the current one, thereby allowing resonances with larger amplification factors to be endured without safety concerns.

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