Dynamic Vibration Absorbers and its Applications

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Abstract

Vibration absorbers are mainly used in civil engineering structures, e.g. high-rise structures and bridges. High-rise structures will experience along-wind vibration and across-wind vibration or earthquake loads whereas bridges are subjected to traveling loads such as moving crowds. However, the random load scenario leads to a different approach to the determination of optimal parameters and are not considered in this study. Here, we focus on deterministic harmonic loads. More recent studies try to extend the application of tuned mass dampers to the suppression of squeal noise.

Tuned mass dampers, consisting of a secondary mass, a viscous damper and an elastic spring are commonly attached to a vibrating primary system for suppressing undesirable vibrations. Closed form optimal design theories are only available for simple systems, i.e. two degree of freedom systems as pointed out by Den Hartog and Brock. However, in finite element analysis of flexible structures several modes must be damped. This requires a simultaneous sizing and positioning optimization of the vibration absorbers.

Several examples are used to demonstrate the procedure using the commercial finite element package PERMAS. Only one single finite element model is needed during the optimization loops. New positions of the tuned mass dampers are automatically updated. Herein, a modal frequency response analysis of this partially damped structural system with non-proportional viscous damping is conducted. Due to a low number of viscous dampers an explicit inversion of the system matrix in modal space according to Sherman-Morrison-Woodbury delivers a fast solution method for the frequency response analysis.

1. Introduction

The dynamic vibration absorber (DVA) was first proposed by Frahm about a century ago [8]. A standard tuned vibration absorber is a single degree-of-freedom system, which can be used to suppress a troublesome resonance or to attenuate the vibration of a structure at a particular forcing frequency. Presuming that the main structure is also represented by a SDOF system, there are three variables which can be selected by the vibration absorber designer to tune the performance of the absorber: the frequency ratio, the vibration absorber damping ratio and the mass of the absorber. The mass ratio between main system mass and DVA mass is usually assumed as a constant parameter defined in a pre-design stage and is restricted due to practical aspects. For this reason, most of the published works content themselves with two design variables. Two degree-freedom DVAs are considered in [10].

Closed form solutions for optimal parameters are limited to an undamped host structure. In that case, the vibration absorber is optimally tuned when the response of the system at the two fixed points is set to be equal. However, invariant points exist only when the main system is undamped [2, 6].

If the primary vibrating system is a MDOF or continuous system, we are faced with the dilemma of several resonance frequencies. Instead of a single vibration absorber, we now might want to apply multiple vibration absorbers [7]. Spatial structures with multiple tuned mass dampers are analyzed in [5]. Plate-like structures are considered in [9, 11, 21]. These vibrations absorbers interact with each other and the modal density is a further issue that unquestionable creates a problem in the design process. Mode crossing and mode veering, in which the modal content of eigenmodes rapidly varies, are well-known phenomena in this context. Besides that the optimal position of the vibration absorbers can be considered. Optimal locations of absorbers are discussed in [21]. The vibration responses of untargeted modes could be neglected, but their contribution might be significant if the exciting force has a wide frequency band such that many vibrations modes of the primary system are excited. Adaptive DVAs or hybrid vibration absorbers (HVAs) [4, 9], which have self-tuning capabilities, might be used to cope with this problem.

It is therefore obvious to solve the design problem by numerical optimization techniques. One of the most common performance indices is H_∞ optimization criterion which is to minimize the maximum amplitude magnification factor of the primary system. Another commonly used performance index is H_2 optimization criterion which aims at reducing the total vibration energy of the vibrating system under white noise excitation [4]. More optimization criterions are given in [22]. Most of the optimal design methods previously proposed are based on an SDOF model for the primary system [1, 3, 14]. Their application to MDOF or continuous systems is based on the hypothesis of a single vibration mode contributing to the response [19]. If the modal frequencies are well separated, the design approach based on equivalent SDOF systems works well [23]. If the DVA is used to suppress vibration over a frequency band in the vicinity of a targeted frequency, a good trade-off between the suppressed original peak and the two newly emerged coupled peaks induced by the insertion of the absorber is crucial to obtain a global vibration reduction within the current frequency band.

Herein, the finite element method is used to discretize the continuous structure. The equations of motion are transformed into modal space and a modal frequency response analysis is conducted to obtain the response curves at selected positions. The size of the modal space is governed by the highest excitation frequency. An explicit inversion of the system matrix in modal space according to Sherman-Morrison-Woodbury delivers a fast solution method for the frequency response analysis [12,13]. Finally, the fully integrated optimization in the commercial FEA package PERMAS [24] is used to solve the combined sizing and position optimization problem. All optimization related definitions can be created by the sizing and shape wizard in VisPER (Visual PERMAS) [25]. The maximum displacement amplitude of a node set over a frequency band is minimized to obtain optimal parameters, i.e. stiffness and damping ratio of each vibration absorber and its optimal position. Initial values of the parameters are problem dependent and have to be determined beforehand. Strategies for a proper selection of DVA parameters are given in [15, 18]. In addition lower and upper bounds of the design variables can be defined. Moreover, additional constraints such as weight or frequency constraints can be used. The positions of dynamic vibration absorbers are updated during the optimization process, whereas the coupling with the main structure is achieved by incompatible multipoint constraints (e.g. $MPC ILINE, $MPC ISURFACE or $MPC IVOLUME). It is worth mentioning, that only one single finite element model is needed.

2. Examples

The first example is a simply supported beam (length L = 1 m, cross-sectional area $A = 0.025^2\ m^2$, material aluminium) subjected to a harmonic pressure load in y-direction. 2560 uniform hexahedral elements were used for discretizing the structure (Fig. 1). Two dynamic vibration absorbers are initially attached at the quarter points (x=250, 750 mm) and one dynamic vibration absorbers at the center point (x=500 mm) of the lower surface of the beam, respectively. The displacement amplitudes at the center point are minimized in a frequency range [0, 120] Hz (Fig. 2). The fundamental eigenfrequency of the primary structure is at 57 Hz. A total of 9 design variables is used, here. Due to the symmetry of the first mode shape dependencies between the design variables could be introduced to reduce the number of design variables by using $DSVLINK in PERMAS. The dynamic vibration absorbers at the quarter points (x_2, x_3) move towards the center point during the optimization (Fig. 3). The stiffness and damping coefficients of the dynamic vibration absorbers are illustrated in Fig. 4 and Fig. 5, respectively.

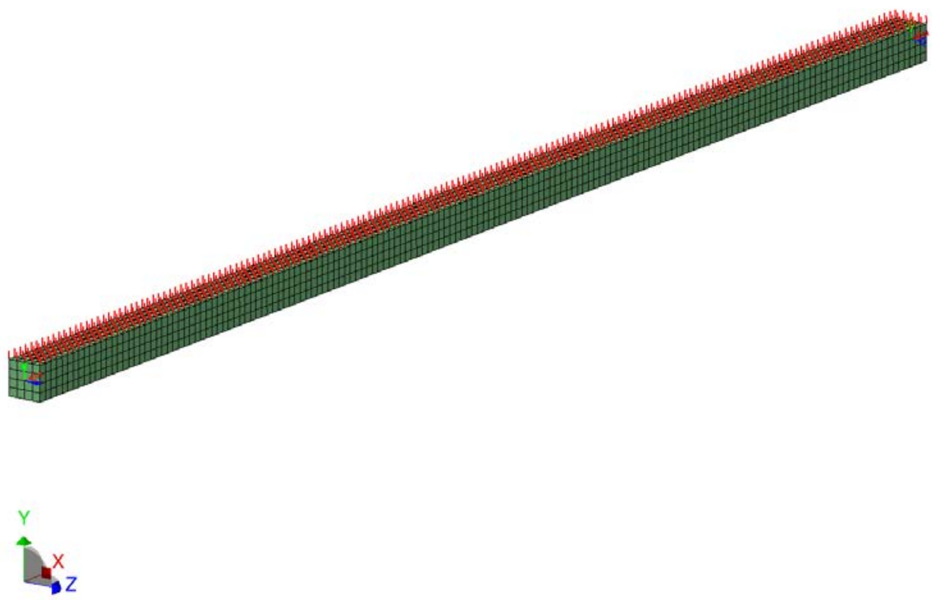

Figure 1: Simply supported beam subjected to a harmonic pressure load.

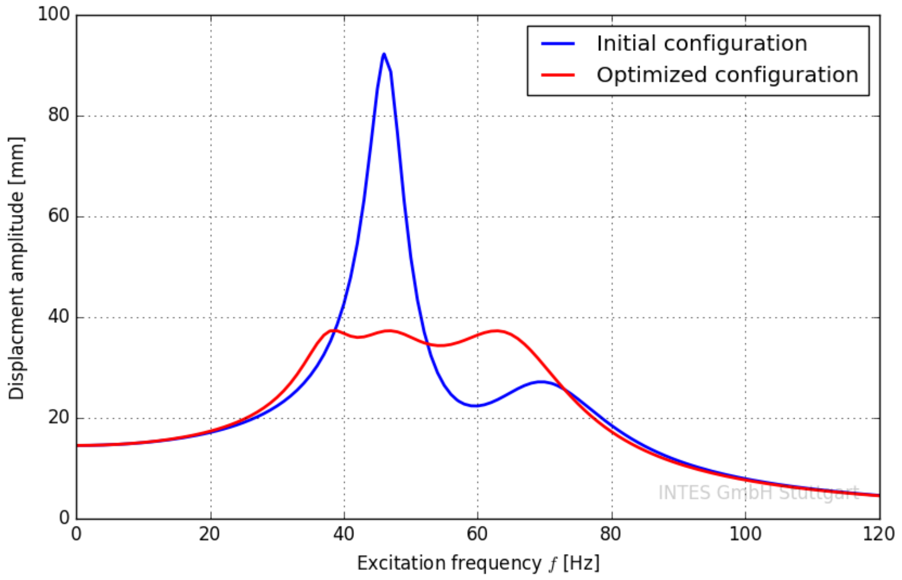

Figure 2: Frequency response curve of the central point

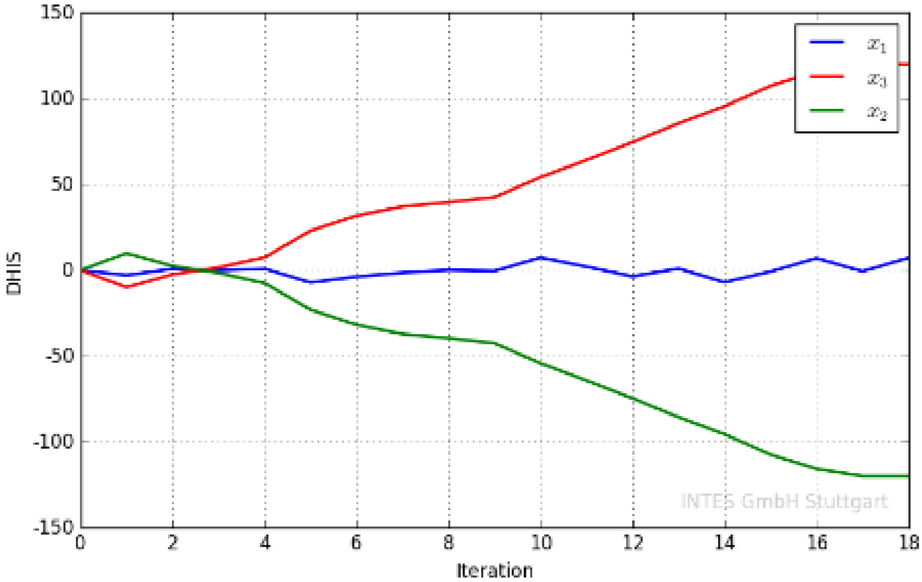

Figure 3: Positions of the dynamic vibration absorbers

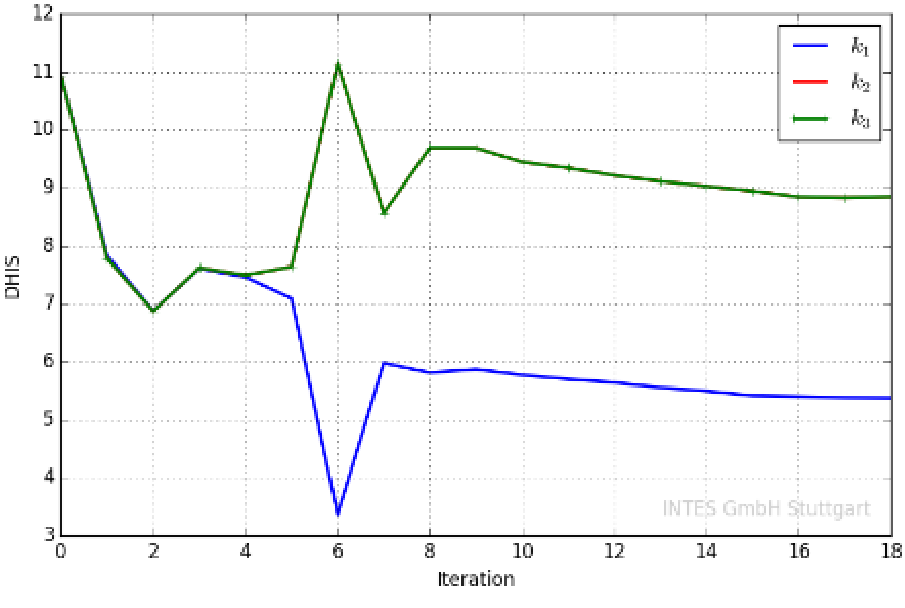

Figure 4: Stiffness parameters of dynamic vibration absorbers

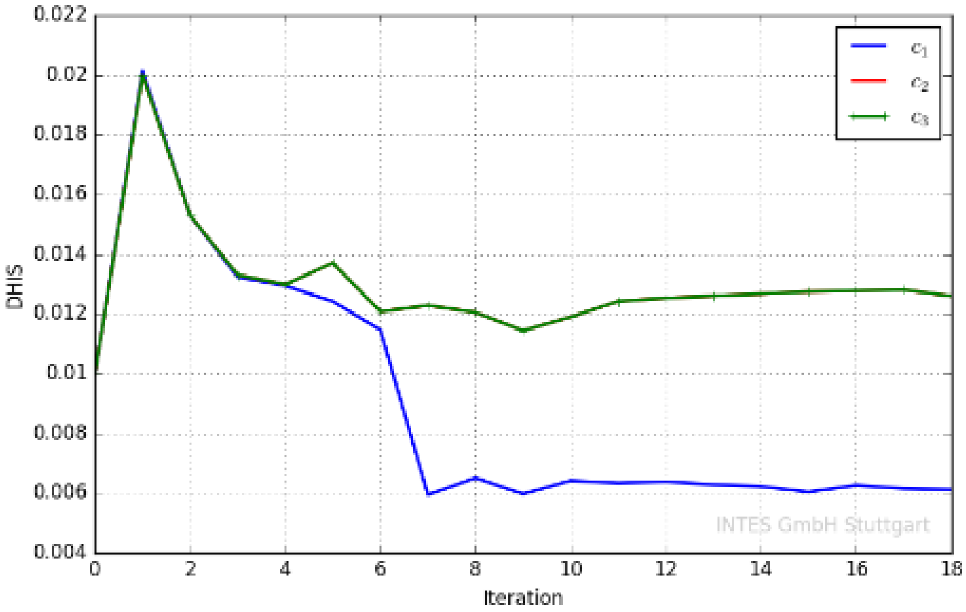

Figure 5: Damping parameter c_i of dynamic vibration absorbers

The second example is a clamped orthotropic plate (Fig. 4) taken from [17]. Two DVAs are initially attached in the antinodes of the second mode shape. Two stationary harmonic loads denoted by blue arrows are applied in the z-direction (Fig.6). Eight design variables, i.e. the positions of the absorbers with respect to x and y-direction, damping and stiffness of the absorbers are used in this study. The absorbers act solely in load direction. The vibration amplitudes in the frequency range [400, 1000] Hz are minimized. The initial positions of the dynamic vibration absorbers coincide with the position of the excitement. A total number of 5400 SHELL4, 2 MASS3, 2 SPRING6 and 2 DAMP6 elements are used here. The first resonance peak in the frequency response curve is almost not affected by the optimization, whereas multiple peaks in the frequency range of interest are significantly reduced (Fig. 7).

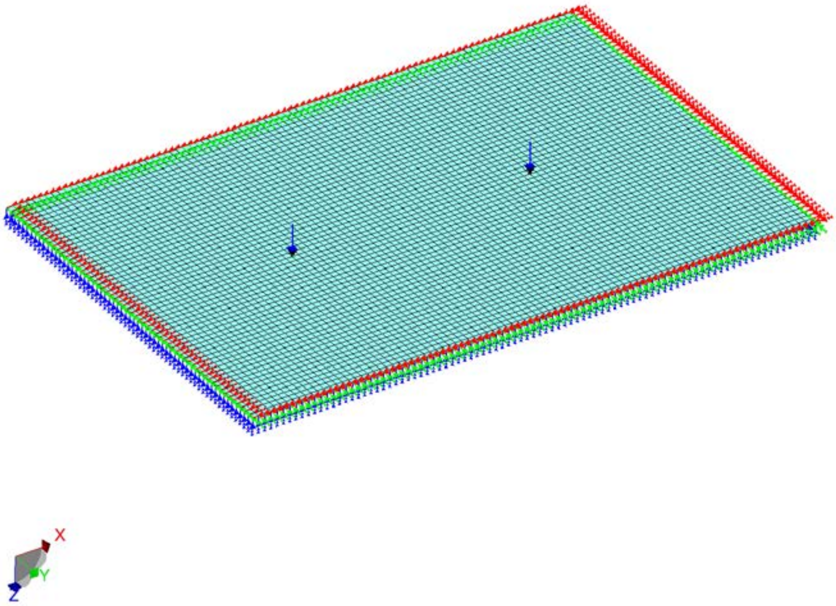

Figure 6: Finite element model of the clamped plate subjected to a harmonic point force

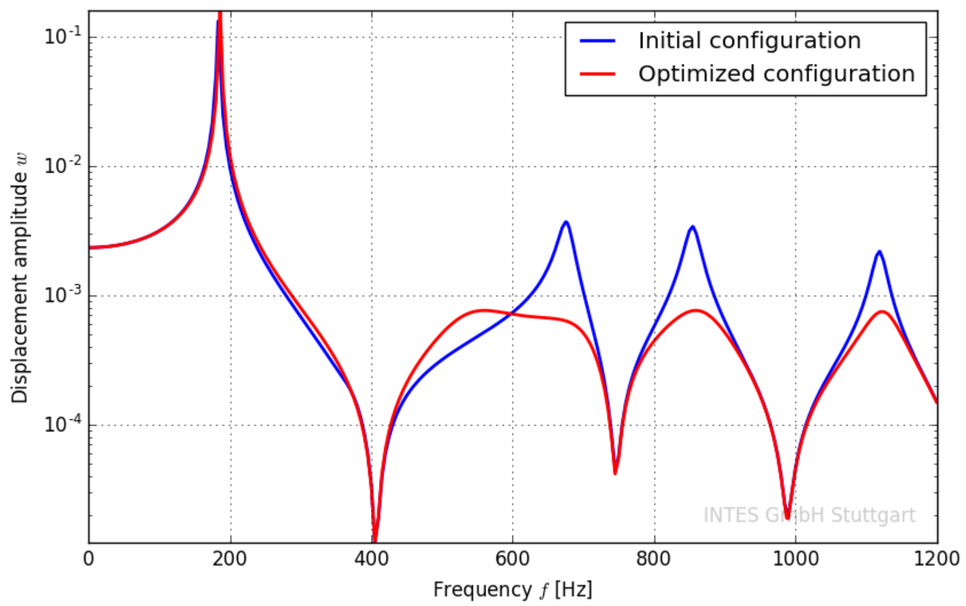

Figure 7: Frequency response curve of the central point

3. Conclusions

Finding closed solutions for optimal parameters of dynamic vibration absorbers is usually limited to simple academic two-degree-of-freedom systems. A fully integrated optimization procedure enables the user to find optimal parameters including positions of multiple dynamic vibration absorbers in the frequency domain. Different objective functions will be investigated in the near future. Although the mass ratio is limited to a few percent of the primary structure, the effect of this design variable seem to be worth investigating.

4. References

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