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On the integration of tuned multi-mass dampers into a topology optimization method for machine tool structural dynamics

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1. Abstract
Topology optimization is becoming an integral part of the design process in various industrial fields in order to keep up with the continuous drive to increase productivity and efficiency. In the field of machine tools, the dynamic behavior of a machine tool's structure is largely responsible for its overall performance. Thus, topology optimization methods targeting the optimization of Eigen frequencies are often used in industrial practice.
A machine tool's structural frequency response (e.g. to external excitation during cutting processes) is also dependent on its damping properties. Therefore, the dynamic behavior of machine tools can be significantly influenced by utilizing one or more vibration suppression systems like tuned mass dampers (hereinafter called TMDs) to target specific vibrations. Although TMDs are often used to solve problems during operation, they are in some cases utilized during the engineering phase, becoming an integral part of the machine design.
By combining both optimal utilization of vibration suppression systems and topology optimization within a structural optimization framework, potential synergetic effects of both approaches can be utilized.
In this paper, the recently started development of such an optimization framework including the automatic optimal positioning and analytic tuning of multi-mass dampers (hereinafter called MMDs) is described. The advantages of MMDs include robustness and easy implementation, as demonstrated by initial simulative results presented in this paper. The described framework in development addresses issues like manufacturing constraints for the topology optimization and restrictions on the MMDs physical properties. The paper concludes with a brief outlook on the consideration of constraints for additive manufacturing and the volumetric distribution of multiple MMDs embedded inside those structures.
2. Keywords: Machine tools, dynamics, tuned multi-mass dampers, topology optimization

3. Introduction
The application of topology optimization methods for structural dynamics problems has been discussed for many years. The optimization of dynamic behavior is a very general term. From a mechanical engineering standpoint, one can distinguish between indirect and direct methods. Indirect methods include those trying to maximize certain Eigen frequencies [1], achieve target (or shift specific) Eigen frequencies [2] or create a so-called band-gap behavior by maximizing the distance between distinct Eigen frequencies [3]. Many indirect methods include Eigen frequencies as optimization constraints rather than objective functions, e.g. [4, 5], a practice often applied in commercial products as well (see [6]). These methods can be called indirect, because the frequency-tuning is essentially achieved by optimizing the stiffness-to-mass ratio of a structure for certain Eigen frequencies and associated Eigen modes. The resulting dynamic behavior is indirectly affected by this optimization.
Direct methods on the other hand aim at influencing the dynamic behavior more directly, e.g. by synthesizing specific mode shapes [7] or minimizing the (maximum) frequency response at specific points or regions within the structure [8]. The latter is especially interesting in the field of machine tools, where the dynamic behavior at the tool center point (TCP) is often the most important point of interest.
The dynamic behavior of machine tools, and whether specific Eigen modes are potentially critical during operation, is in no small part also dependent on the damping properties of the structural parts and coupling components connecting them [9]. However, the topology optimization of the damping behavior of structural, load-bearing components has not been the focus of research. Instead, problems like an optimal material distribution within damping layers [10], especially regarding acoustics important in the automotive industry [11], were investigated.
Besides structural optimization techniques, a classic approach to reduce an existing machine's (or any vibrating structure's) peak response amplitude in one of its Eigen frequencies is to attach a vibration suppression system to it. Strategically positioned and tuned to absorb one response peak, the kinetic energy is either dissipated by different means of damping behavior or counteracted by opposing forces. Expensive, high-maintenance active systems (actors) can adapt to varying frequencies, passive systems like TMDs are only effective within a narrow frequency range. [9]

An extensive amount of research has been conducted into different types of TMDs. Regular ones consist of a single absorber mass connected to the main system with a specific attachment stiffness and optimal damping, see Eq.(1), Eq.(2). Many different approaches have emerged in the literature, using multiple degrees of freedom (DOF) for attachment [12] or distributing the damper mass over multiple masses [13-15]. They have in common, that the optimal tuning is usually achieved by using numeric optimization methods and significantly simplifying the system they are attached to. Also, available research is dominated by its application to buildings exposed to ground vibrations due to seismic activity, e.g. [16].
There is noticeably less research on the automatic optimum placement of such auxiliary systems, although [17] proposed a genetic algorithm approach to position damper systems (again in the field of civil engineering) across different floors of high-rise buildings, for example. Industrial applications of those optimized TMD systems in general and applications to machine tools in particular are rare, e.g. [18].
One aspect which all passive vibration suppression systems have in common is that their effectiveness depends on their mass in relation to the mass of the system they are attached to. By reducing the main system's mass, e.g. by topology optimization, the auxiliary system can be smaller. Yet, no research on a combined approach of topology optimization and integrated optimal tuning of TMD systems is available.
This lead to the research project presented in this paper, where a concept of integrating optimally tuned systems of MMDs into a topology optimization framework targeting the direct optimization of structural dynamics of machine tools is proposed. The general concept is summarized in Fig. 1. Either due to problems during operation or requirements during the design phase, the goal of optimizing structural dynamics can either be solved by damping critical vibrations or by optimizing one or more structural components w.r.t. static and especially dynamic considerations. The challenges and potentials of combining both approaches into an optimization framework are described in the following paragraphs. The motivational application example on the right side of the figure will be described in detail in the outlook of this paper.

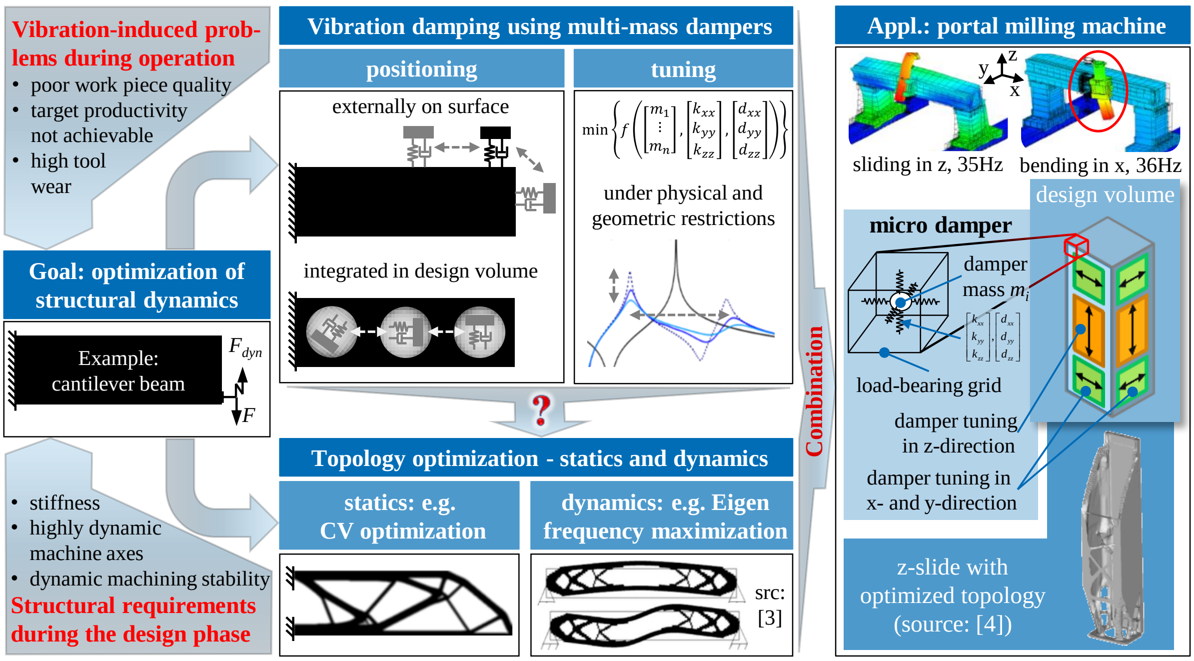

Figure 1: Schematic summary of the combined topology optimization and damping method concept

4. Analytically tuned MMDs

As mentioned above, the methods for optimal design of multi-mass or multi-DOF dampers presented in the literature focus on numerical optimization methods to optimize parameters like the number of masses, the mass fraction and the distribution of masses and attachment frequencies. Due to the computational effort of the combined optimization task, described in detail below, a fast analytic approach would be preferable.
Thus, the approach presented in this paper relies on analytic tuning methods of classical TMDs and applies them to a multi-stage design to broaden the effective frequency range and increase the robustness of the design.

4.1. Analytic tuning of multi-stage MMD units

When designing a regular TMD, the main system's dynamic behavior in the targeted Eigen mode is simplified to that of a single mass oscillator with an equivalent static stiffness and kinetic mass M. After defining the absorber mass m, the damper mass ratio $\mu = m/M$ is used throughout the optimal tuning. Using DEN HARTOGS [19] equations to derive the optimal frequency ratio f_{opt} between the absorber's and the system's Eigen frequency (optimal detuning), the necessary attachment stiffness $k_{TMD.opt}$ can be calculated:

$$f_{opt} = \frac{\omega_{TMD}}{\omega_{sys}} = \frac{1}{1+\mu} \Rightarrow k_{TMD,opt} = m\left(\frac{\omega_{sys}}{1+\mu}\right)^2 \tag{1}$$

A TMDs performance relies on its optimal damping, which can be determined using BROCKS [20] formula:

$$c_{TMD,opt} = 2m\omega_{sys}\sqrt{\frac{3\mu}{8(1+\mu)^3}} \tag{2}$$

Depending on the system's mass and the target frequency, it can be difficult to achieve the target damping value in practice, and deviations in $c_{TMD,opt}$ result in significantly reduced performance [16]. It is shown in several publications (e.g. [13,21]) that distributed mass dampers offer comparable performance to a single TMD with less damping, while being more robust to deviations in tuning (stiffness, damping or mass). Thus, distributing the single TMD mass represents the first design aspect of the MMD presented here as well (Fig. 2, left).

By attaching an absorber mass to a system, its resonance peak is split into two new resonance peaks. Neglecting any damping effects, the frequency spacing of those peaks can be calculated using Eq.(3) derived by [19]. The two solutions to this quadratic equation only depend on the mass fraction μ, where higher mass fractions increase the peak spreading effect of an absorber. A graphical representation of this formula is shown in Fig. 3 (top, middle).

$$\left(\frac{\omega}{\omega_{sys}}\right)^2 = \left(1+\frac{\mu}{2}\right) \pm \sqrt{\mu + \frac{\mu^2}{4}} \tag{3}$$

Obviously, the distribution of the single TMD mass in an MMD reduces the peak spreading due to the decreasing individual mass fractions, narrowing the effective frequency range of a MMD system. To increase this frequency range, a multi-stage design is proposed, which calculates the newly created resonance peaks from the previous stage of absorbers using Eq.(3), and tunes additional masses to the lower- and uppermost peaks $\omega_{i,l}$ and $\omega_{i,u}$, respectively. This multi-stage design (as illustrated in Fig. 2, right) iteratively increases the effective frequency range of the proposed MMD system.

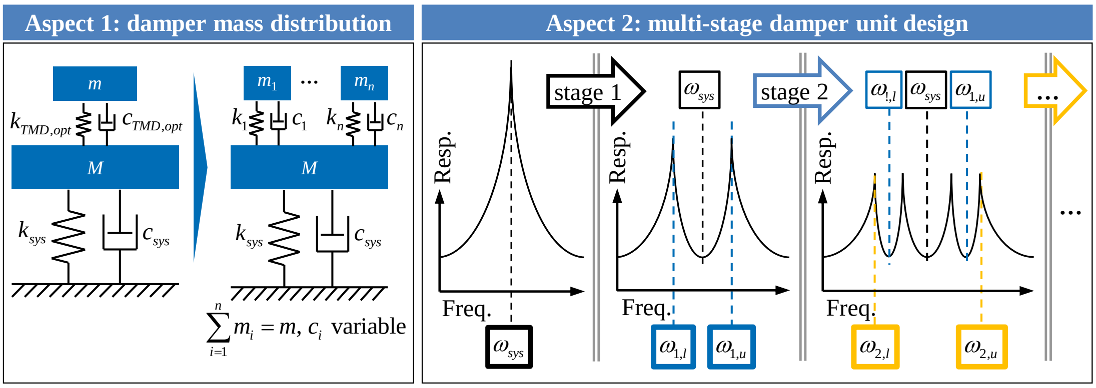

Figure 2: Design aspects of tuned multi-mass damper units

All MMD results presented in this paper are obtained from a FE simulation of a clamped, thick cantilever plate (steel, 20x100x500mm), where two MMDs are attached at different positions along the central axis and tuned to the 1st and 2nd bending mode (normal to the plate). The MMD attachment positions (as shown in Fig. 3, right) are chosen such that the mutual interference of both MMDs is low. The MMDs are idealized as point masses, individually connected via uniaxial springs/dashpots to the plate. A cantilever plate is chosen in order to validate the simulations on a test bench at a later stage. Also, the position of clamping can be moved along the longitudinal axis of the plate to detune the system and analyze the robustness of the MMD design.

An example of the multi-stage design and the spreading effect is demonstrated in Fig. 3 (bottom), where the overall damping of the MMD is kept constant and deliberately at a low level, such that the individual resonances can be clearly distinguished. Note that with an increase in stages, the maximum receptance is reduced, despite the constant damping.

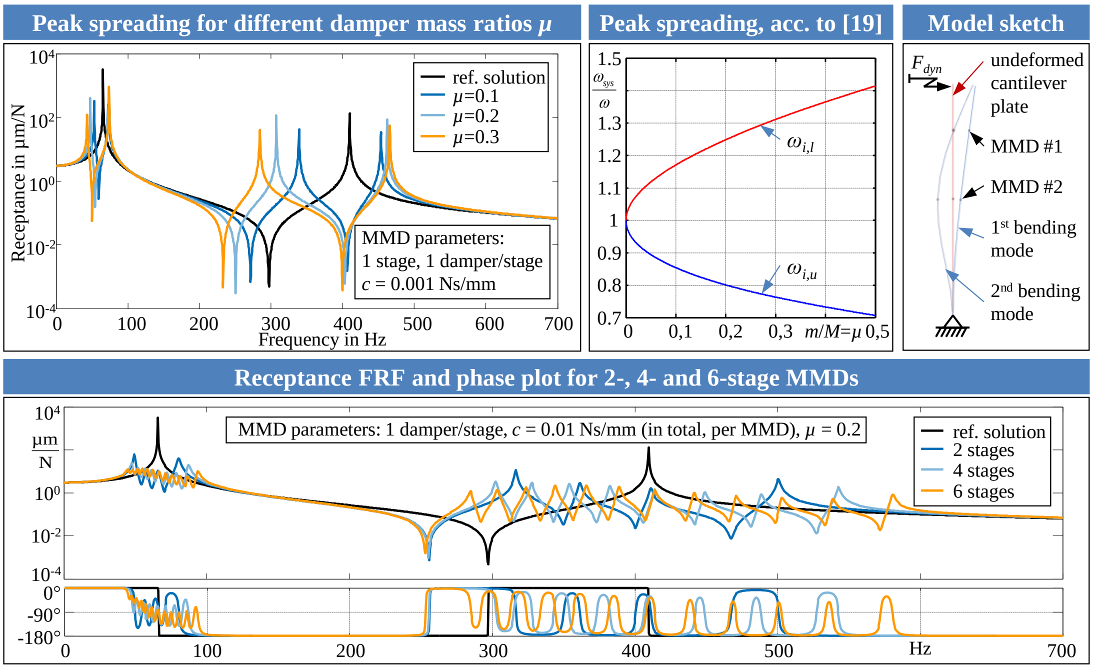

Figure 3: Utilizing the resonance peak spreading effect of adding absorbers to construct multi-stage MMDs

4.3. Robustness and manufacturing considerations

The premise of the presented MMD design is robustness both towards changing frequencies of the main system and towards variations and limitations in the stiffness and damping properties of available attachment devices for the individual dampers. Possible attachment devices include rubber attachment buffers in different configurations and shore hardness classes of rubber mixtures. An example of those buffers is shown in Fig. 4 (top, right).

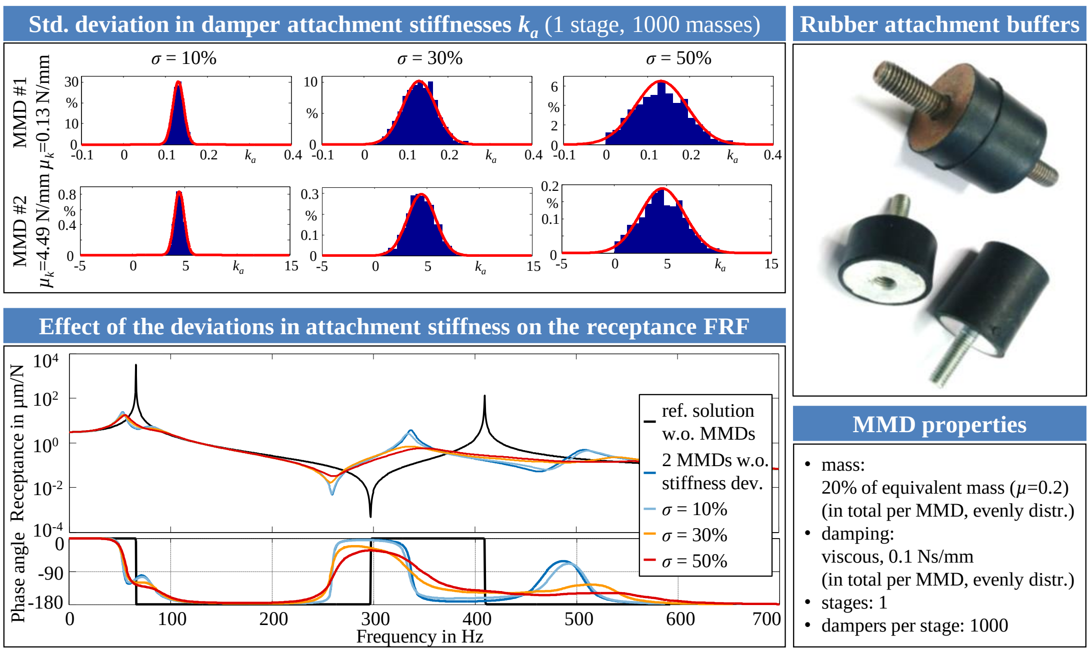

Figure 4: Effects of variations in the individual attachment stiffnesses on the MMD performance

Due to manufacturing and material property fluctuations, the stiffness of a single type of buffer and shore hardness can vary by more than 10%, the damping coefficients can vary even more. To consider those variations, the optimally tuned attachment stiffnesses, damping coefficients and masses of each individual damper are optionally subjected to a semi-truncated standard deviation to guarantee positive values, see Fig. 4 (top, left). Initial simulations for single- and multi-stage MMDs show a very beneficial effect of even large deviations, as can be seen in the example in Fig. 4 (bottom, left). Again, the sum total of viscous damping within each MMD

(0.1 Ns/mm) is very low compared to the optimum damping of a regular TMD with equal mass ($c_{TMD,opt}$ = 0.7262 Ns/mm for the 2nd bending mode), obtainable from Eq.(2).
In contrast to the analytic tuning of regular TMDs, whose effectiveness relies heavily on optimal damping, the optimal tuning procedure presented here allows for the consideration of damping coefficients of available attachment devices. Available attachment stiffnesses are taken into account by scaling the number of masses per stage, thus changing the individual damper masses to achieve the same tuning frequency with different stiffnesses. Detailed results of this MMD design and practical validations on a test stand will be presented at a later stage.

4.3. Optimal positioning criteria

Any TMD needs to be in a position undergoing significant amplitudes within the targeted mode shape to be able to dissipate kinetic energy. Finding an optimal MMD position within a structure for a single targeted mode shape is straightforward. The nodal displacements of the considered mode shape, extracted from a FE modal analysis, yield optimal positions with high amplitudes. As soon as multiple MMDs are to be used to target multiple frequencies and mode shapes, the optimal positioning turns into a minimization problem, where optimal positions have high amplitudes at positions with low amplitudes in target modes of other MMDs. Even for a simple cantilever plate like the one used for demonstration in this paper, finding two optimal positions for both targeted Eigen modes cannot be realized without any mutual interference, see Fig.3 (top, right). The optimal positioning subproblem is further constrained by the necessity to have enough surface area at certain positions within the structure to attach MMDs to or to have enough local volume to integrate the MMDs into (see volumetric distribution in Ch. 6). Those constraints are imposed onto the positioning problem by the changing topology of the structure.

5. Topology optimization framework

5.1 Computational effort and technical challenges

The optimization framework under development is to be used for the optimization of real structural components of entire machine tools, which are often very large and complex structures containing a high number of DOFs. The FE dynamics analyses needed to calculate the system responses require a lot of computational effort. A modal analysis and subsequent frequency response calculation is necessary to determine the mode shapes and Eigen frequencies for tuning the MMDs and the frequency response behavior for the topology optimization module. Additionally, the tracking of Eigen modes (by using the MAC criterion, see e.g. [22]) will be considered, since mode shifting is likely to occur during the optimization process. To cope with the huge computational effort, a hybrid topology optimization scheme will be implemented to reduce the overall size of the system of equations by fully deleting Finite Elements in regular intervals (and adding some in critical areas, BESO-type approach). In between those reduction intervals, a localized topology optimization problem will be solved.
By adding, moving and retuning MMDs to and in the structure during the optimization process, the dynamic behavior of the structure is changed. The topology optimization in turn changes the dynamic behavior of the system, detuning the MMDs. A suitable coupling scheme has to be developed, while the achievable stability and convergence of the combined process and the topology optimization procedure in particular are critical but unknown. Due to this added complexity, the optimization core is kept modular to investigate the performance and suitability of different topology optimization approaches and coupling schemes (both implicit and explicit).

5.2 Applicability to real structures and manufacturing considerations

Structural components of machine tools are usually designed as welded steel or cast iron constructions. Welded steel constructions are comprised of mostly planar plates with relatively low wall thicknesses and are inherently ill-suited to be target designs of topology optimization methods. Thus, manufacturing constraints focus on improving the castability of the design proposals. A simple casting constraint would be to forbid the formation of undercuts in the designs w.r.t. a single cast removal direction. The framework in development operates on FE models discretized with a uniform Cartesian grid of hexahedra, allowing the definition of a casting direction along one of three principal axes of said grid. By allowing only the removal of surface material on the top and bottom projection of the structure w.r.t. the casting direction, undercuts are avoided, as proposed in [23], for example.

6. Outlook on additive manufacturing and the volumetric distribution of MMDs

The full potential of the combined optimization method can be exploited using additive manufacturing and distributing a large number of micro-dampers volumetrically within a load-bearing lightweight structure. A motivational, practical application is shown in Fig. 1 (top, right). Portal milling machines exhibit a dynamically challenging design component, namely the z-slide providing vertical movement of the main spindle. Typical mode shapes of this type of machine design include horizontal bending and sliding of said z-slide in the z-axis guides. Depending on the z-slide’s vertical position, the Eigen frequencies change significantly, and surface area for the attachment of external TMDs is very limited. Volumetrically distributed MMDs which are tuned towards those typical Eigen modes and provide broadband performance due to their multi-stage design do not suffer from the same spatial and frequency-related restrictions.

As mentioned above, the lighter the structure, the less damper mass is needed for vibration suppression. Also, the necessary individual damping associated with each mass increases with the number of dampers. By using additive manufacturing, it is conceivable that hundreds or thousands of masses can be integrated into a load-bearing truss structure derived by topology optimization and connected to it by parametrizable beams of material. Damping can be introduced into the masses for example by designing them as hollow spheres and retaining unmolten material within. These aspects will be the focus of research at a later stage of the ongoing framework development.

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