Suppression of high frequent resonances by applying Tuned Mass Dampers

C.A.M. Verbaan[1], P.C.J.N. Rosielle[1], M. Steinbuch[1]
[1] *Control Systems Technology group, Department of Mechanical Engineering, Eindhoven University of Technology, The Netherlands*
c.a.m.verbaan@tue.nl

1 Introduction

In high tech industry, advanced motion systems are used to position components for processing purposes. Increasing performance requirements for fast positioning as well as improved accuracy are conflicting. Ideally, these requirements transform into lightweight mechanical designs with aim at high natural frequencies. This reduces the required actuator forces and increases the reachable control bandwidth. Damping is typically low in these types of structures and resonances with large magnitude peaks appear often in a broad frequency range. Tuned mass dampers (TMD) are commonly used in dynamic structures to dissipate energy on a specific resonance frequency[1,2]. In this research we look into lightweight motion platforms using passive damping absorbers. Firstly, the influence of TMD's on anti-resonances is studied and in addition the TMD high frequent damping behaviour is investigated.

2 Methodology

As model of a motion stage, an unconstrained square plate is analyzed with finite element method (FEM) using 30 shell elements on a plate edge. As material, aluminium is chosen by its isotropic material behaviour. An undamped modal analysis is performed and a number of mode shapes and corresponding natural frequencies, including rigid body modes are exported. A state space description of the plate in modal-1 form[3] is generated and modal damping of 10^{-3} [-] is assumed. A TMD state space model is created in nodal coordinates. A combined state space model of the plate with TMD's is calculated by Matlab. To study the damping behaviour, frequency response functions (FRF) are calculated and visualized in Bode diagrams and poles are calculated to visualize system properties. To verify the results, the FEM plate model is extended with TMD's and a full harmonic solution is calculated.

3 Damping of anti-resonances

Anti-resonances (AR) lead to a small-frequency-banded decrease of the system amplification. It decreases the efficiency to counteract disturbances if an AR is present below the open loop cross-over frequency. The appearance of AR's in the transfer function is not only a result of the mechanical design, but also a result of the actuator and sensor locations. Therefore, AR frequencies appear and/or change with actuator and/or sensor position. Four TMD's in z-direction are added to the plate model to influence the dynamics at an anti-resonance frequency (Figure 1a). The natural frequency of the TMD's is tuned approximately at the AR frequency. A unit force in z-direction on a plate corner is used as input and the z-displacement on this corner is taken as output. Figure 1b shows four collocated frequency response functions (z(t)/F(t)). The dashed line represents the free plate without TMD's. The dotted line and the continuous line show transfer functions with four TMD masses of 0,35% of the plate mass for different damping values. The dash-dotted line is generated with TMD's of 1.5% of the plate mass. The magnitude at anti-resonances can be increased in the same manner as the magnitude at resonances can be suppressed.

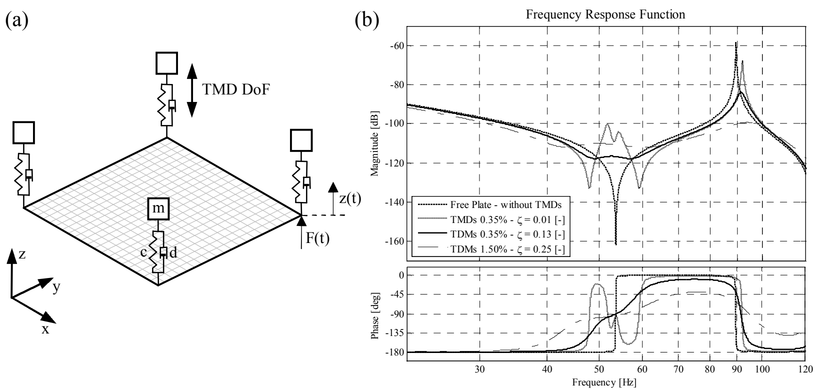

Figure 1a: The plate model with four TMD's added on the plate corners.
Figure 1b: Frequency response functions for different cases.

4 High frequent damping

Research is executed on the high frequent damping behaviour of TMD's with relatively large damping values.

The model and approach used are the same as in the previous study of anti-resonance damping (see Methodology section / fig 1a). However, in this case the damping is

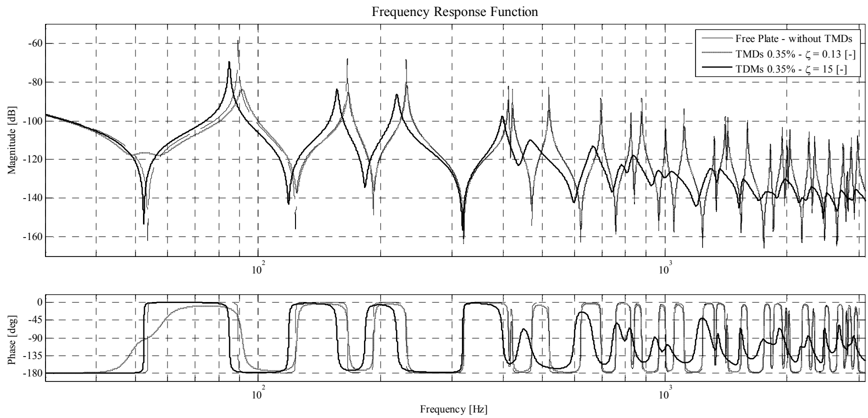

Figure 2: Frequency response functions for different cases

increased. FRF's are visualised and system poles are calculated.

In Figure 2, three frequency response functions are shown. The dashed line represents a free plate and the dotted line represents the anti-resonance damping as discussed in previous section. Notice the influence of the TMD on the damping of higher (anti)resonances. The solid line shows an FRF for high TMD damping (ζ=15). The low-frequent resonances peak up again, based on velocity coupling between plate corner and TMD. However, at higher frequencies, a magnitude reduction of 30 dB is obtained with respect to the free plate. The frequency band in which resonances can be suppressed depends, among other parameters, on the damping value. Figure 3 represents the poles of the plate system. Approximately half the number of poles is shifted more to the left half plane.

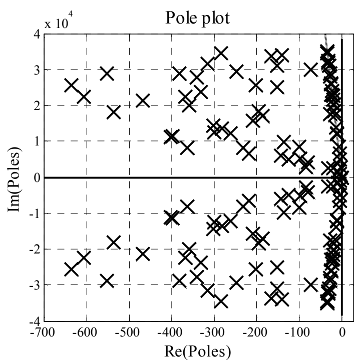

Figure 3: Poles of the plate with 4 TMD's added

The angles of the poles (damping ratios) are calculated to analyse the damping behaviour of the system and the result is plotted in a graph. In figure 4, the pole angles are represented as function of the corresponding pole numbers. On the x-axis the pole number is visualised and on the y-axis the damping angle is shown on a logarithmic axis. The first six poles correspond to rigid body modes and are

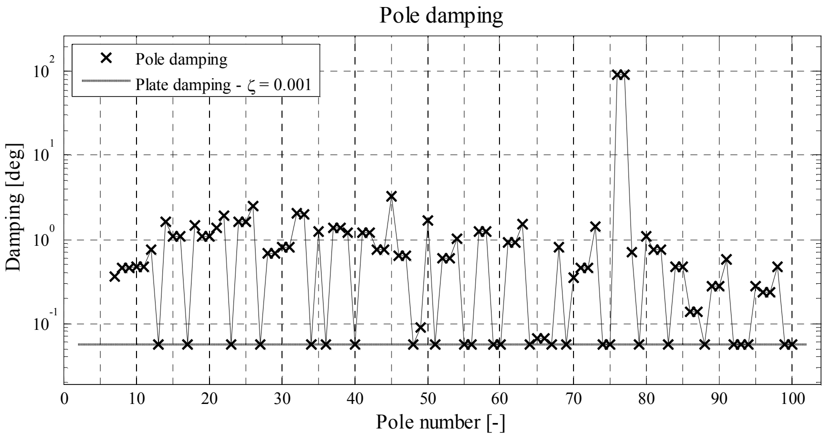

Figure 4: Pole damping in degrees as function of pole number

therefore not visualised.

The damping at approximately half the number of poles is influenced by the TMD's. The damping values are increased with respect to the low-damped plate poles, which have a damping of $5.7*10^{-2}$ [deg]. However, the damping values of the natural frequencies without displacements on the plate corners are not influenced.

4 Conclusions

It is possible to add damping to a major part of the system poles with a limited number of passive damper absorbers added to a square plate. This results in poles with a larger damping ratio. In this way it becomes possible to damp a number of high frequent resonances in a frequency band.

References:

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