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MULTI-BODY DAMPING OF A VANE CLUSTER

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ABSTRACT

Spring-damper systems are standard for reducing blade vibration amplitude at vane clusters. Spring-dampers can only be used with an altered geometry of the inner shrouds. In most cases a separation of the inner shrouds is inevitable.

In this paper an alternative damping system without changes of the outer inner shroud geometry is developed and analyzed. Two analytical models - a simplified Rigid Body Model and a 3D Finite Element Model show, based on similar results, a good comparison. The analytical results were validated by shaker tests.

A high level of agreement between simulation and test was achieved.

INTRODUCTION

Motivated by increasing HCF-loads of vane clusters additional damping systems are necessary. The usual damping system is a damper-spring positioned on the inner shroud. Such a damping system with separate inner shroud and additional seal carrier (Fig. 1) was described analytically and experimentally in [1].

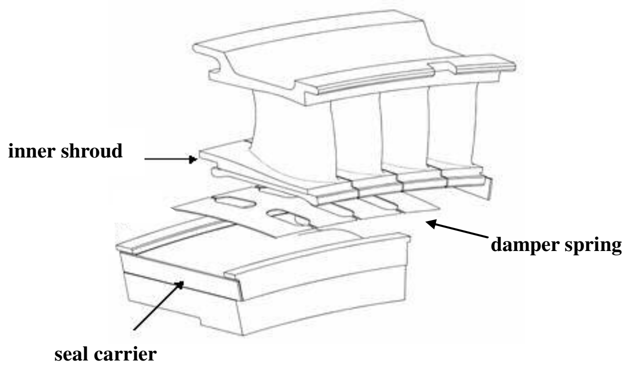

Fig. 1: Damping concept MTU Aero Engines.

Further damping systems for vane clusters were analyzed in [2] and [3]. In both cases, significant geometric changes of the inner shrouds were needed. For this reason, the result were modifications of the natural frequencies, mode shapes and aerodynamic properties of the clusters. To avoid this geometric modifications an alternative damping concept will be presented based on internal damping bodies. The use of internal cavities to place mechanical dampers has been investigated many times. Free damping bodies as a damping system have been analyzed and patented. The basic patented options for placement of the damping bodies are as follows:

- a self-tuning impact damper [4] as a single body damper:

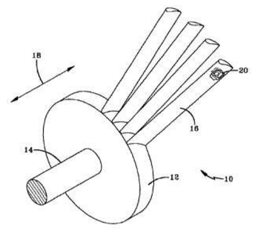

Fig. 2: NASA, 2004, US 6,827,551;

- a damping system consisting of some dampers placed in many different cavities [5]:

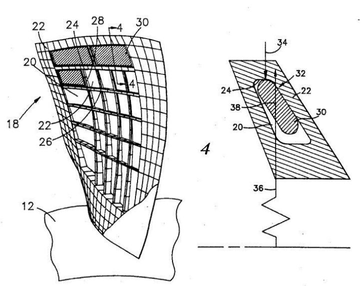

Fig. 3: United Technologies, 1993, US 5,232,344;

- a particle damping system [6]:

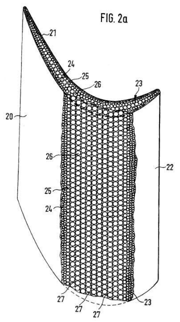

Fig. 4: MTU Aero Engines, 1997, US 5,634,189.

Analytical investigations of a single free friction-impact damper and of multi free friction-impact dampers (multi-body damping system) inside of hollow airfoils were done in [7] and [8]. Free dampers inside of vane clusters are not analyzed yet. For the investigated damping concept there are two physical causes for reducing vibration amplitudes. One hand, energy dissipation by friction between the damping body and the cavity takes place. On the other hand an energy dissipation by impulses between the damping body and the cavity wall takes place.
The concept of free damping bodies mean, that the dampers are not clamped and are sliding if the tangential contact force to the cavity surface is equal to the sliding friction force.
In case of free dampers in rotating blade cavities the dampers are pre-loaded by centrifugal forces and the influence of gravity can be neglected. In case of vanes or vane clusters the damper bodies in the cavities are only pre-loaded by the gravity force.
The approach for vibration damping of a non-separated vane cluster aims to reduce the vibration amplitudes of the fundamental vibration cluster mode shape. Higher vibration modes are not considered..

1 NEW DAMPING SYSTEM FOR VANE CLUSTERS

Multi-body damping systems consist of some free impact damper bodies. They are placed in cavities of the inner shroud of a vane cluster (Fig. 5). Investigations of this damping approach are main part of this publication.

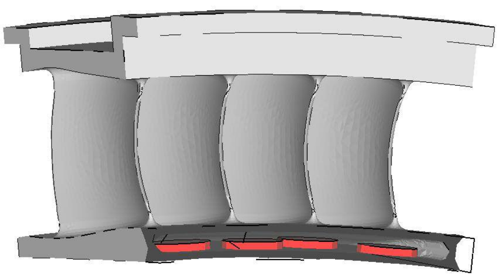

Fig. 5: Example of a vane cluster multi-body damping system.

Basically, the number of cavities, the number of used damping bodies and the placement of the cavities (inner shroud or airfoil) is variable. This damping method can be used for compressor clusters and turbine clusters. The main parameters defining the damping effect are: material combination, damper body mass and gap between the damping body and cavity. Based on these parameters the cavity and damping body dimensions can be defined and the number of required cavities can be specified. The basic operation and the essential design parameters are also given in the patent application [9].

2 ANALYZED VANE CLUSTER

A high power compressor vane cluster consisting of 4 airfoils was selected for the following investigations. In the inner shroud a cavity was eroded by taking the material load into account (Fig. 6).

Fig. 6: Analyzed vane cluster with cavity.

For this cluster three different damping configurations were experimentally analyzed: single damper (Fig. 7a), double damper (halves of the single body) (Fig. 7b) and quad damper (quarters of the single body) (Fig. 7c).

a) b) c)

Fig. 7: damping configurations: a) single b) double c) quad damper

Analytical simulations were done for the configuration of the single and double damping systems..

3 SIMPLIFIED RIGID BODY MODEL ANALYSIS

The analytical approaches are based on two models: a simplified rigid body model and a 3D Finite Element (FE) full contact model. The reason for two models is the large computation time for the solution of the 3D FE model with friction and impacts. A numerical optimization of the damper mass and the gap size would therefore not have been possible. Therefore these optimizations were done using the rigid body model. The computation using the 3D FE model were only done for optimized single damper and some damper configurations with small scattering masses. Initially, the rigid body model was calibrated by the 3D undamped model results. The simplified rigid body model for the first bending cluster mode in case of a single damper body is depicted on the Fig. 8:

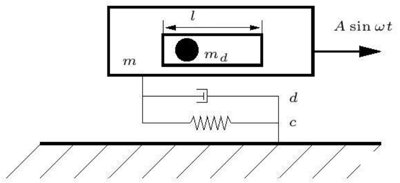

Fig. 8: Simplified rigid body model.

The model parameters are:

m – mass of the inner shroud;

c – substitute stiffness first bending cluster mode;

d – substitute viscose damper first bending cluster mode;

m_d – mass of the damper body;

l – maximum gap between damper and cavity walls;

A and ω – parameters of the harmonic excitation;

$\mu = 0.4$ - friction coefficient damper body/bodies – cavity;

$k = 0.7$ - coefficient of restitution, selection is founded in [10].

The amplitude A leads to 70% fatigue strength of the undamped 1st cluster mode shape. Friction was modeled according to Coulomb friction law without regularization. Impacts were modeled based on the Newton impact law. This modeling will lead to a model with a variable structure, a piecewise linear structure:

$$\left.\begin{aligned} &mx'' + cx' + dx = A\sin(\omega t) + F_R, \\ &m_d x_d'' = -F_R, \\ &F_R = -\mu\, m_d g \end{aligned}\right\} \qquad (1)$$

and

$$(m + m_d)x'' + cx' + dx = A\sin(\omega t). \qquad (2)$$

The linear subsystems correspond to the state of the motion of the damper: equations (1) to damper sliding, equation (2) to the damper sticking. Condition for switching from sliding to sticking is

$$x' = x_d', \qquad (3)$$

for switching from sticking to sliding:

$$|x''| = \mu g\ . \qquad (4)$$

The impact is modeled as a short-term change in the damper's ($x_d' = v_d$) and cluster's ($x = v$) velocities to $\tilde{v}_d$ and $\tilde{v}$ respectively by formulas

$$\left.\begin{aligned} \tilde{v} &= \frac{mv + m_d v_d - k m_d (v - v_d)}{m + m_d}, \\ \tilde{v}_d &= \frac{mv + m_d v_d + k m (v - v_d)}{m + m_d}. \end{aligned}\right\} \qquad (5)$$

Therefore the impact could not be denoted as a system state but as an additional switching condition between system states. The numerical analysis was performed in the time domain for each excitation frequency to reach the steady state condition. The switching points between sliding and sticking of the damper as well as in case of impact were calculated with high accuracy. First the case of single damper body is analyzed. Two parameters were varied: the damper's mass and the gap between the damper and the cavity. In the following the results for the optimized damper mass are shown. Fig. 9 includes the forced responses of the inner shroud for different gaps (reference, reference/2 and reference/10) between damper body and cavity.

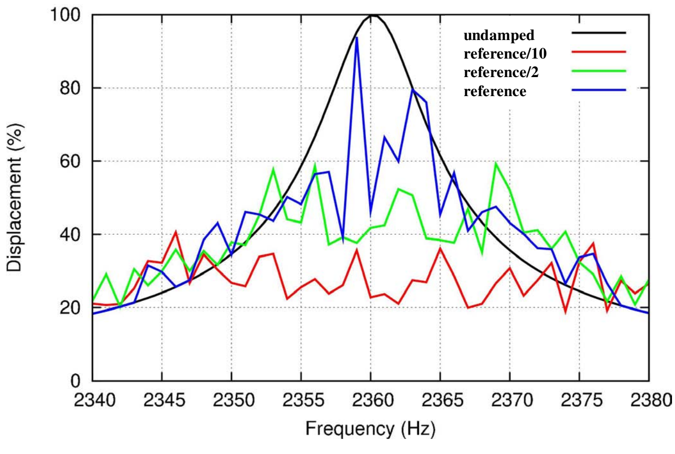

Fig. 9: Simplified rigid body model – variation of the gap between damper and cavity walls

Displacement amplitudes are normalized by the maximum amplitude of the undamped system. The green curve in Fig. 9 shows averaged 50% amplitude reduction. This is technically realistic and agrees with the experimental tested case. Fig. 10 shows the comparison between selected single damper system (optimized mass and gap “reference/2”) and the double damper system (two damper bodies with together equal mass to the optimized single damper). The basic result of this comparison is, that the averaged amplitude reduction in case of the double damper in separate cavities is not worse then in case of one shared cavity. The scattering of the curve in case of damped systems comes from the non-periodical solutions in cases of some excitations. Non-periodical solutions take place if there are impulses between damper body and the cavity. In these cases an excitation takes place for each impulse. Sometimes, the solution amplitudes will be caused by such excitations larger than before the impulse for a short time. In Fig. 9 and 10 the absolute maximums of the amplitudes in the steady states are plotted. The averaged forced response curves are smoothed. In order to exclude numerical reasons for these scattering results, numerical optimizations regarding computational accuracy and computational time were done subsequently.

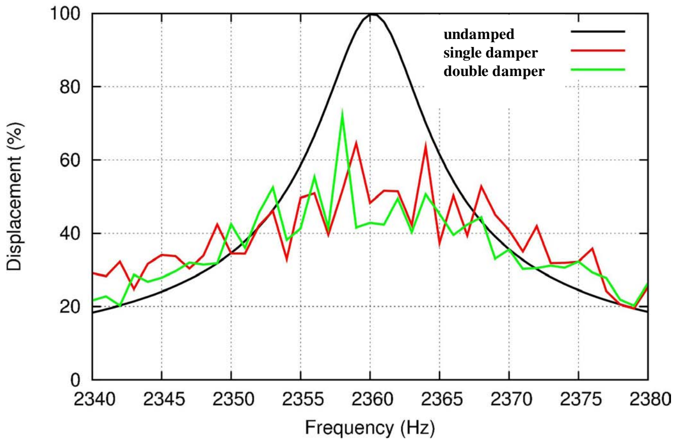

Fig. 10: Simplified rigid body model – comparison of single and double damper systems

3 3D FINITE ELEMENT MODEL ANALYSIS

The analytical analysis using the 3D FE model was done for the single damper system with optimized mass and gap “reference/2”. The model is presented in Fig. 11:

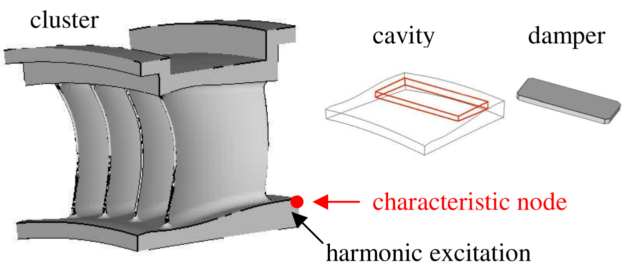

Fig. 11: 3D Finite Element model.

The model was analyzed by FE Code ABAQUS. All results are described at the characteristic node (Fig. 11). First, a realistic excitation amplitude was adjusted using steady state dynamic calculations without dampers. The final amplitudes lead to a 70% HCF loading in the forced response simulation. Second, forced response with the same excitation level and fixed damper is calculated on the similar way. “Fixed” means, the damper is a part of the cluster. This is modeled by multi points constraints between the damper and the cavity. Then calculations in the time domain by “ABAQUS Explicit” with Coulomb friction contacts between damper and cavity were accomplished (Fig. 12).

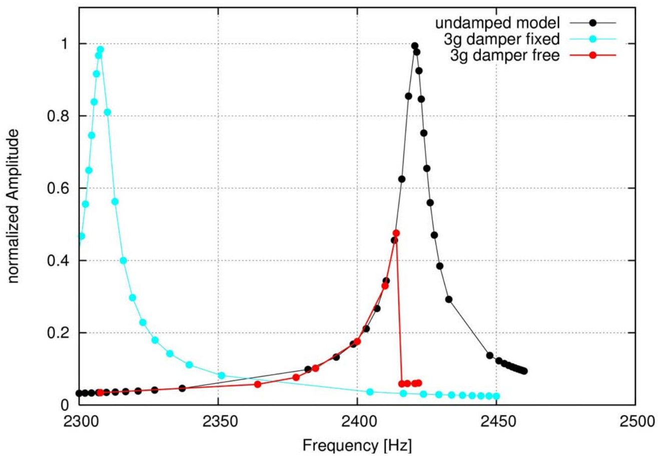

Fig. 12: 3D Finite Element results, forced responses

As main result the FE model leads to well comparable averaged amplitudes as the rigid body model analysis. Coincidentally, a small change in the eigenfrequency was determined. The small frequency shift is caused by the not significant role of the friction between damper and cavity. The curve of the damped forced response confirms this phenomenon. Until a certain excitation frequency the curve is identically to the undamped forced response curve. For higher frequencies the curve shows an offset and drops down to the forced response curve of the cluster with fixed damper. Such offsets were first observed in [7]. We decided to call this phenomena “impact mistuning”. Founded by the resulting impulses the eigenfrequency of the damper system is mistuned and becomes equal to the frequency of the fixed damper system. The resonance curve of the damped system from the 3D FE calculation does not scatter as strongly as in the case of rigid body modeling.

The time domain response of the damped 3D system is non-periodically as the rigid body model. This non-periodic behavior occurs when there are impulses between the damping body and the cavity in the steady state as mentioned in chapter 2.

An example of such a solution can be seen in Fig. 13 for the excitation of 2414 Hz. This time domain plot of Fig. 13 corresponds to the maximum red point in the Fig. 12.
The impulses occur in the time domain unevenly. This can be caused by physical as well as numerical phenomena. Therefore the solution is irregular (non-periodic). Caused by very long computation times, more detailed classification was impossible. In each case the amplitude deviations are less compared to the calculations based on the rigid body model. In contrast to the results of the evaluation in case of the rigid body models, averaged amplitudes are plotted in the forced response curve (Fig. 12).

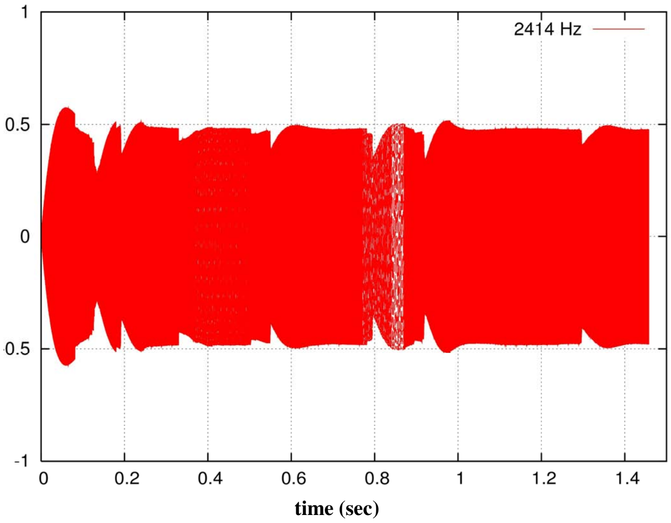

Fig. 13: 3D Finite Element results,
normalized tangential vibration on tip in the resonance

The ordinates in the Figures 12 and 13 are normalized tangential displacements by the maximum undamped amplitude, but in difference to the rigid body model analyses not multiplied by 100%.

3 EXPERIMENTAL ANALYSIS

The experimental investigations were done by the usage of a high pressure vane cluster. This cluster was prepared for the multi-body damping system by eroding a cavity in the inner shroud as shown in Fig.6. For the excitation of the first bending cluster mode a modal shaker was used. The cluster was clamped in a realistic way (Fig. 15) and excited via base point. Since all clusters have a different circumferential orientation, a different damping effectiveness depending on the orientation is expected. The limits of the cluster alignment are vertical and horizontal orientation as shown in Fig 14. All other orientations should lead to results between the vertical and horizontal orientations.

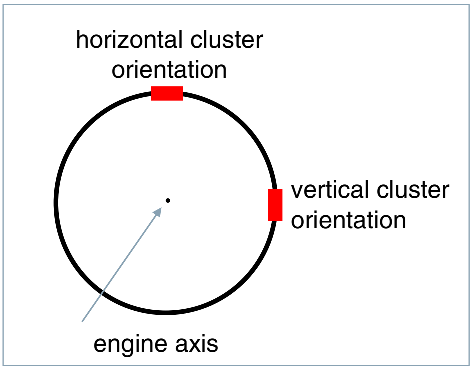

Fig. 14: Different vane cluster orientations

The first step was to identify the correct mode shape of the cluster. In Fig. 16 the first bending cluster mode was identified by a Laser Scanning Doppler Vibrometer. Afterwards a comparable amplitude was adjusted as in the simulation. All test results are normalized to these maximum undamped amplitudes. Afterwards the different damper configuration were tested with the same excitation level.

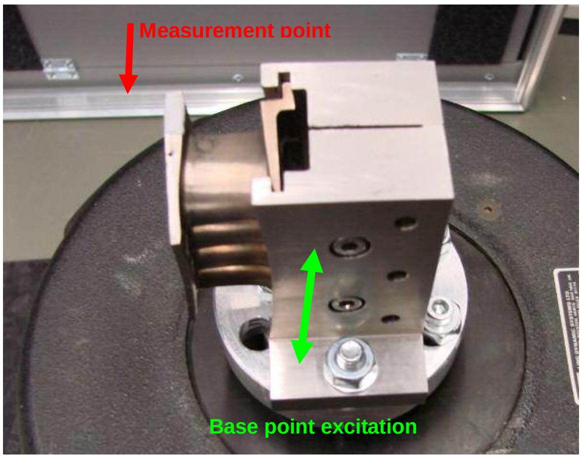

Fig. 15: Test configuration

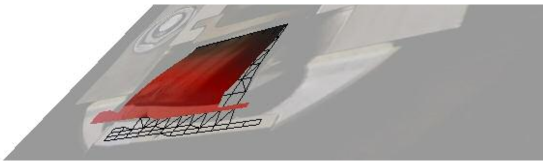

Fig. 16: Identified undamped first bending cluster mode shape

The results of the different damper configurations are shown in Fig. 17. The multi-body damper systems lead for the horizontal damper orientation to a better amplitude reduction as the single configuration. An reduction of more than 50 % is achievable.

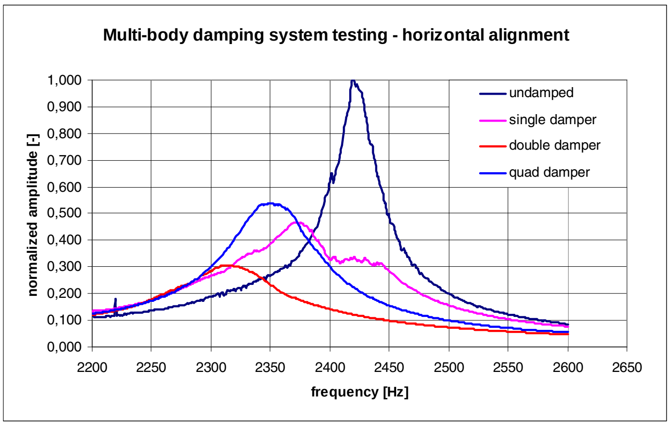

Fig. 17: Test results for different damping configurations – horizontal cluster orientation

The analytically predicted forced response drop is also visible on the measured curvature (Fig. 17, single damper).
Even for the vertical damper orientation the reduction of vibration amplitude can be demonstrated as shown in Fig 18. The double damper configuration showed in both cases (vertical and horizontal orientation) the best damping effectiveness.

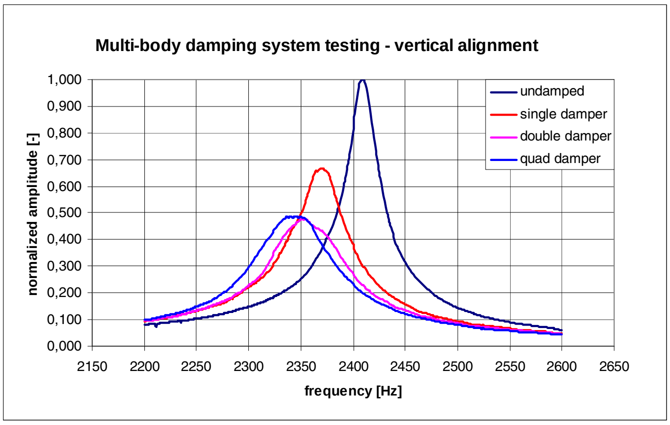

Fig 18: Test results for different damping configurations – vertical alignment

The next step was to identify the influence of the damper mass to the results. For this topic 3 different single dampers were tested. The Fig. 19 contains the results for varying the damper mass +/- 6 % (horizontal orientation). The mass variation occurs via geometry changing in the tangential direction, therefore the gaps between damper bodies and the structure were changed as well. It can be seen, that the damper mass and the gap between the damper and structure has a big impact to the amplitude reduction results. To optimize damper design it is inevitable to build up a high resolution simulation model in order to define the ideal damper mass and gaps.

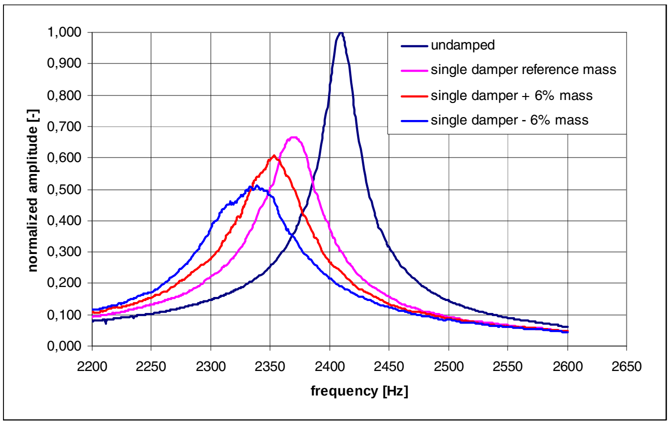

Fig 19: Test results for varying damper masses (single damper, horizontal orientation)

4 CONCLUSIONS

The present thesis describes a new approach to damp vane clusters – a multi-body damping system. The damping system was analyzed based on an example of a high power compressor vane cluster. Analytical analyses were done using a simplified rigid body model and a 3D Finite Element model with full friction contact modeling. The results of both analytical models are very similar to each other and show a high level of correlation to the experimental results. In the case of the optimized single damper about 50% amplitude reduction is reached. A high level of correlation of the amplitude reduction was observed analytically and experimentally. A new phenomena “impact mistuning” was identified as the main physical mechanism for amplitude reduction. A multi-body damping system is an alternative to common spring damper systems with big advantages that no changes in structural design are necessary.

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