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More Flexible Damping Systems for Blades and Vanes

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The blades and the vanes of aero engines are subject to very high thermo-mechanical loads. In some cases, an additional damping system is necessary to reach the lifetime goals. Commonly, damping systems based on energy dissipation due to friction are used, e.g. under platform dampers for blades and spring dampers for the vanes. These damping systems have some limitations: under platform dampers work well mostly for just one mode family, their effectiveness is limited relative to rotational speed (because of the associated contact forces) and is dependent on the excitation order. The spring dampers work well for more than one mode family but their effectiveness is limited concerning the available contact force (just one value). Additionally, the use of the spring dampers requires a significant, sometimes suboptimal design change of the vane cluster. In this paper, some alternative damping systems are introduced and analyzed. All these new systems offer additional possibilities for damping and give more design flexibility. Two of them: insert damping and rocking damping are also based on frictional energy dissipation. The third one, impulse mistuning, adopts a special kind of absorption and is based on the so called targeted energy transfer. The analytical results for the insert damping systems were presented previously in Borufka et al. (2009), while in this paper the experimental validation by shaker tests is shown. The rocking damping was not presented so far – to the knowledge of the authors. Impulse mistuning was first presented in: Hartung and Retze (2011) and Hartung et al. (2016). In this work, an overview of such damping systems and some additional information on the experimental validation of some impulse mistuning systems are presented.

1 Damping Systems for Blades and Vanes: State of the Art

As mentioned above, the blades and vanes of modern aero engines are subject to very high thermomechanical loads and in some cases an additional damping system is necessary to reach the life targets, before high cycle fatigue (HCF) damage occurs. The HCF damage could be caused by different kinds of vibrations: forced synchronous vibrations (SV), driven by resonances with excitation or engine orders (EO), forced non-synchronous vibrations (NSV) and self-excited non-synchronous vibrations (flutter). In Figure 1, a sketch of a Campbell diagram with all these kinds of vibrations from Krack et al. (2016) complemented by indication of excitation orders is plotted. In Figure 2, the measured Campbell diagram of a blade stage from a MTU test engine is shown. Different kinds of vibration are visible, however an example of NSV is not absolutely identified. All of this would be analyzed during the analytical development and during engine validation. The need for a damping system arises when one (or more) of types of vibration lead to an HCF problem. The state of the art of damping systems for the blade stages are under-platform dampers, for the vanes, respectively, vane clusters – damper-springs. Both types of designs are depicted in Figure 3. The acting principle of both damping system is energy dissipation due to friction. These damping systems could be designed to be very effective in damping of the (mostly one, sometimes two) mode shape(s) or mode family causing an HCF concern. Hereby only mode shapes with enough relative motion between the blade or vane and the damper can be damped effectively. Additionally, the dampers are designed for limited rotation speed intervals only – due to the fixed geometry and the fixed mass of the damper. Consequently, the limitations of the under-platform dampers are:

- Mode family (mostly just one)
- Rotation speed interval
- Excitation order

and the limitations of the spring-dampers:

- Mode family (mostly 1-2)
- Friction force (just one value)

The strategic aim of turbine makers is to invent and to integrate the damper design with significantly fewer limitations. In this paper, three different damping systems with reduced effectiveness' limitations than the state of the art dampers are analyzed.

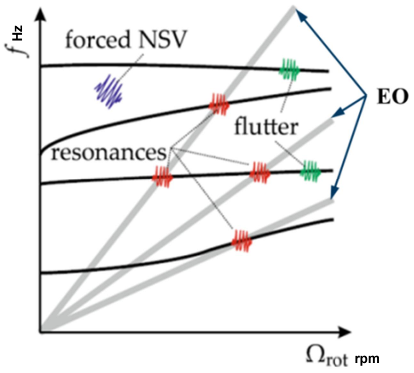

Figure 1. The schematic Campbell diagram from Krack et al. (2016) including excitation orders (EO)

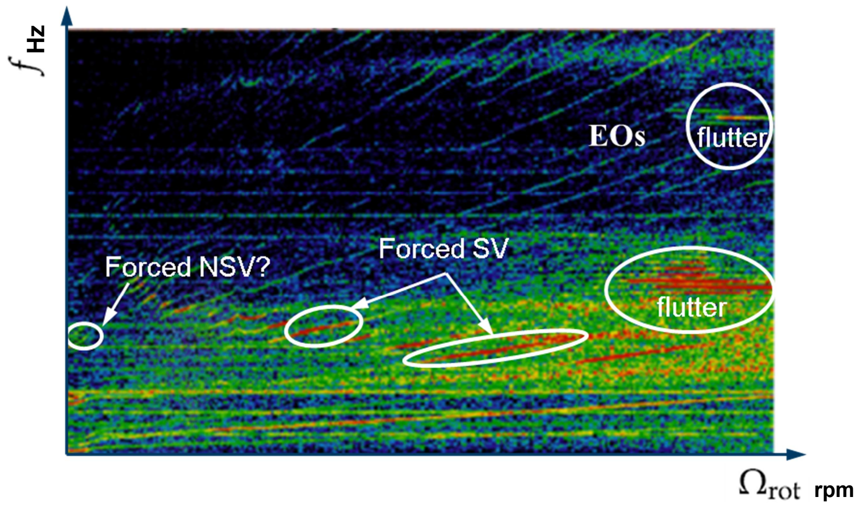

Figure 2. The measured Campbell diagram of a blade stage, from an MTU test aero engine

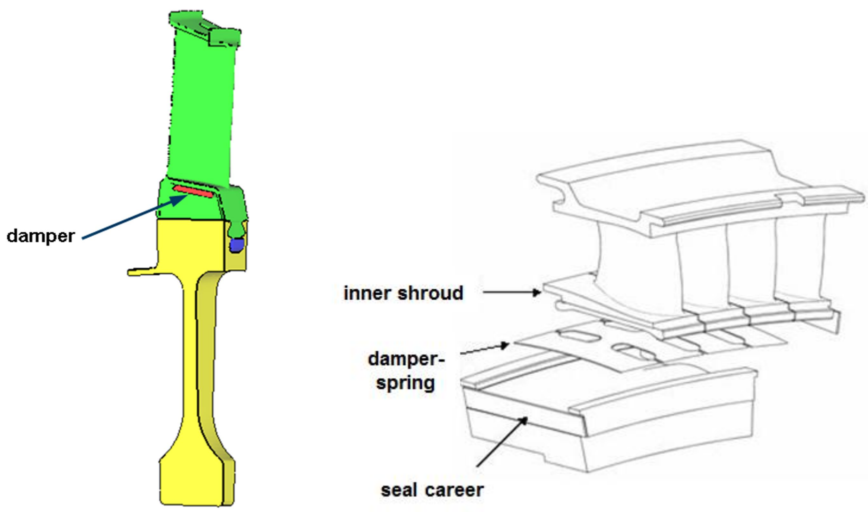

Figure 3. Examples of the common blade and vane damping systems – under-platform damper and damper-spring

2 Insert damping Systems

An insert damping system is a combination of a damping device and a cooling system which requires the use of hollow blades. For this reason, the insert damping system is independent of the excitation order. The idea of combining the damping and cooling system for hollow blades and vanes is not new. In Borufka et al. (2009), some design ideas from different patent applications were shown, followed by the presentation of the MTU design "Insert Damping System" with analytical results of the damping effectiveness. In the present paper, a brief overview of the most important topics from Borufka et al. (2009) will be given, supplemented with new experimental results.

The design idea of the insert damping system is shown in Figure 4. It entails complementing the cooling system "insert" with "pedestals", which are in frictional contact with specific positions of the airfoil. These positions will be designed dependent on the resonant mode shapes. In Figure 5, this is indicated for two resonant mode shapes, first bending (1F) and first torsion mode (1T). During the resonance passage, the energy dissipation due to friction between pedestals and the airfoil leads to the reduction of the vibration stresses and avoidance of HCF damage.

The proof of the damping effectiveness for the design shown in Fig. 4 and 5 was given in Borufka et al. (2009) as well and is plotted in Figure 6. The amplitude reduction for the 1F mode was 40% and for the 1T mode 70%. The analysis was performed to engine conditions. Due to a moved static equilibrium, the curves in Figure 6 are not symmetric anymore. In the same paper, robustness of this design concerning friction coefficient and hence concerning contact conditions in general was proven.

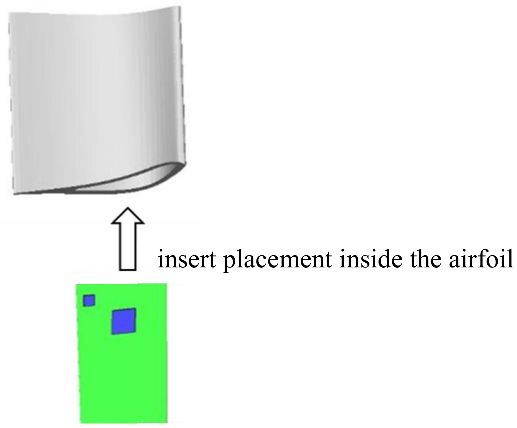

Figure 4. Design idea of the insert damping system: cooling system insert with "pedestals" from Borufka et al. (2009)

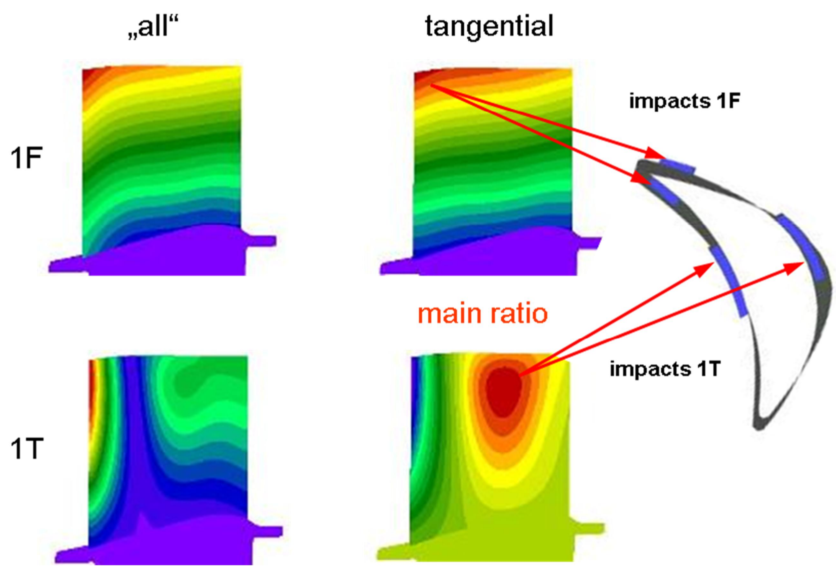

Figure 5. Design development of the insert damping system, Borufka et al. (2009)

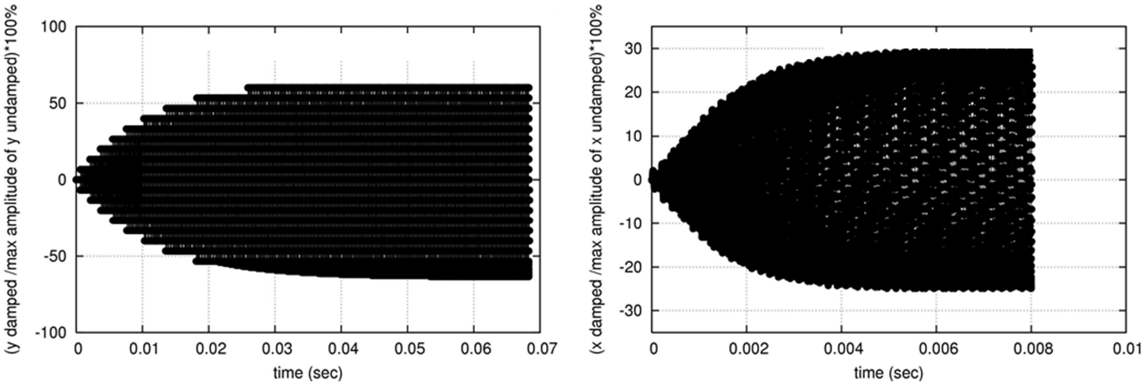

Figure 6. Proof of the damping effectiveness of the insert damping system, Borufka et al. (2009)

Recently, the experimental validation of the insert damping concept was performed at MTU Aero Engines. For this reason, a shaker test specimen was developed and manufactured. The specimen including design features (pedestal positioning) is shown in Figure 7. The blade root cannot be shown for confidentiality reasons. Because of the significant difference between laboratory and engine conditions, it was not possible to test exactly the same design as analyzed in Borufka et al. (2009). However, the test constitutes an appropriate concept validation. Just mode shape 1T was tested, the comparison of the measured and calculated mode shapes including the placement of the characteristic node for experimental vibration comparison is plotted in Figure 8. Additionally, the positioning of the substitute excitation for the linear plausibility proof is explained. The experimental results as measured forced response curves on this characteristic node with and without insert are plotted in Figure 9. Additionally to the measured curves, linear steady state dynamics calculations with measured equivalent linear damping ratios are depicted as well. The substitute excitation for the linear calculation was adjusted to the measurements without inserts. The measured amplitude reduction of 56% compared to the measurements without inserts confirm the possibility of high damping potential of the insert damping. The plausibility proof with the linear analysis (amplitude reduction 60%) proceeded successfully as well.
Summary of the conclusions concerning insert damping system:

- Independent on the excitation order
- Damping of more than one mode family simultaneously conditionally possible
- Robust concerning contact conditions
- Experimentally validated in a shaker test
- Flexibility concerning rotation speed is limited: it deals with one contact force similar to the state of the art damping systems

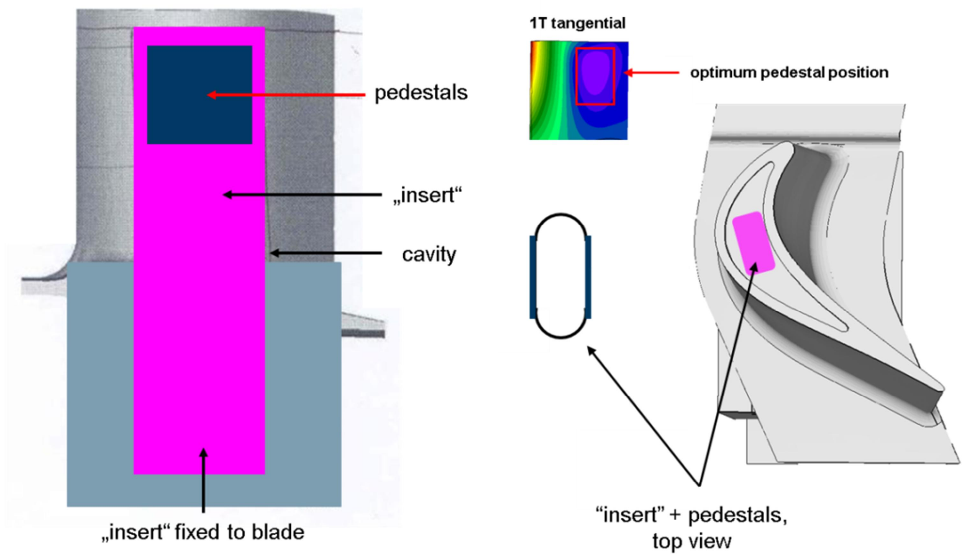

Figure 7. Specimen for the shaker test and explanation of the pedestals positioning

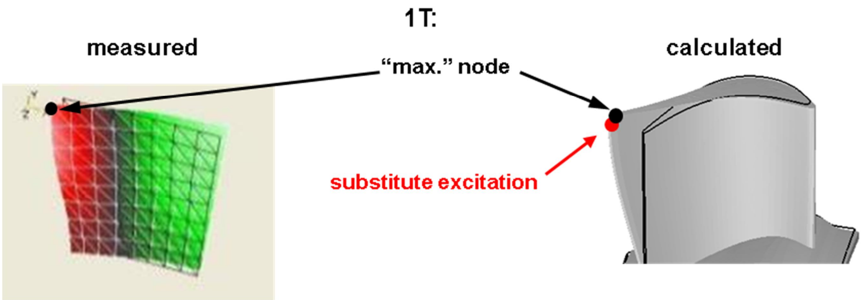

Figure 8. Measured and calculated mode shapes 1T and explanation of the characteristic node to comparison of the experimental results and linear proof of its plausibility as well as positioning for the substitute excitation in the numerical analysis

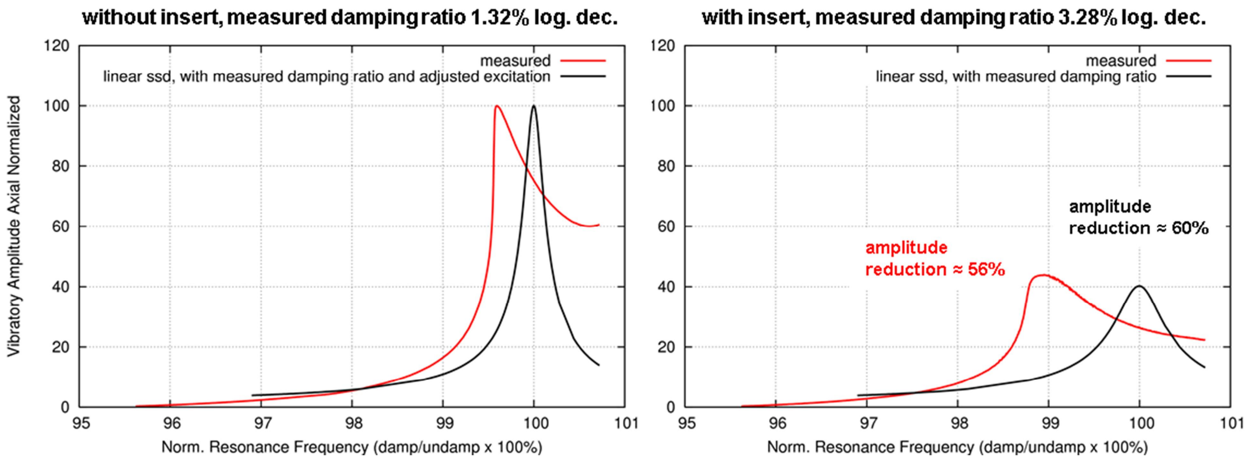

Figure 9. Measured and calculated results on the characteristic node

3 Rocking Damper

The second damping system, which is independent of the excitation order, is the rocking damper. This is only possible for blades with a "cover plate" as a special design feature. Cover plates are seals placed on the face side of the blades. Hence rocking dampers can only be used together with cover plates or similar features. The design of a rocking damper will not be shown in this paper, but two substitute models are analyzed in Erbts (2011). Here, two new damping systems are analyzed, the rocking damper being one of them. In the following, the most important results of this work are presented.

In Figure 10a, a sketch of dummy rocking damper with a dummy carrier is given (necessary to illustrate the boundary conditions on the damper because of the cover plate), Figure 10b shows a sketch of the assembly with a dummy blade, and Figure 10c the placement for a real blade.

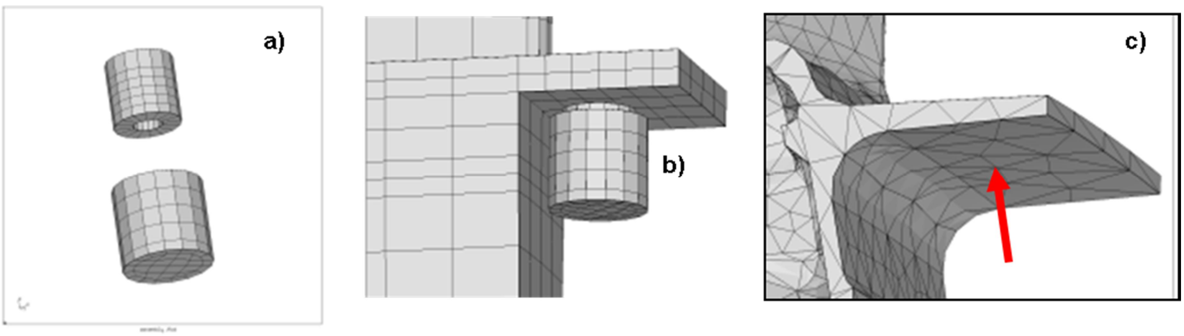

Figure 10. Geometrical features of the rocking damper derived from Erbts (2011). a) dummy damper and carrier, b) assembly of these dummy with a dummy blade, c) placement of the rocking damper for a real blade

In Erbts (2011) a real blade with substitute damper was analyzed, whereas two substitute models were analyzed, a lumped parameter model and a finite element model (in CalculiX) with a regularized contact formulation realized in a user-subroutine. The preparation of the lumped parameter model for the mode shape 1F free shroud is exemplarily explained in Figure 11.

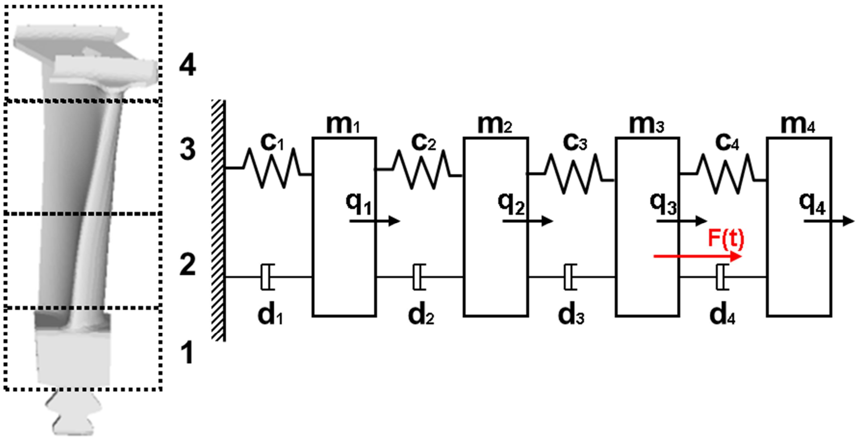

Figure 11. Lumped parameter model of the undamped blade, Erbts (2011)

The parameters shown in this Figure are:
c_k – substitute stiffness, derived from the relevant mode shape and eigen frequency,
d_k – substitute damping ratios, derived from the substitute stiffness,
q_k – generalized coordinates chosen concerning the dominant direction of the relevant mode shape,
F(t) – substitute aero-elastic excitation force of the relevant resonance crossings, derived by forced response prediction or from testing.
In the simulations, harmonic excitation was assumed.
The modelling of the damper influence was proposed in Erbts (2011) as follows: In case of sticking damper conditions, the motion of the substitute lumped mass m_1 is suppressed:

$$q_1'' = q_1' = 0, \tag{1}$$

hence a three D.O.F. linear ordinary differential equations system with an additional algebraic condition for the contact force F_C to prove the end of sticking (whenever the contact force is larger than Coulomb's sliding friction force F_T) needs to be solved. In case of sliding, a four D.O.F linear ordinary differential equations system with the additional Coulomb friction force is to be solved. The condition for the end of the sliding becomes

$$q_1' = 0 \; and \; F_C < F_T \tag{2}$$

Such lumped parameter models have to be prepared for each mode shape separately. The analysis with the lumped-parameter models were performed in the time domain until steady state conditions were reached. In Erbts (2011), two mode shapes, 1F and 1T, both with free shrouds, were analyzed.
The second substitute model, prepared and analyzed in Erbts (2011), was a standard FE model of the blade with a simple follow regularized contact formulation:

$$F_T = -\mu F_N \psi, \quad \psi = \begin{cases} -1, & if \quad q_T' < -\varepsilon \\ q_T'/2\varepsilon & if \quad -\epsilon < q_T' < \epsilon \\ 1, & if \quad q_T' > \varepsilon \end{cases} \tag{3}$$

The analysis with the second model was also performed in the time domain but in difference to the lumped-parameter model as a transient resonance passage through the resonance using the approach described in Hartung (2010). Care was taken to ensure that the frequency sweep velocity is slow enough to obtain stationary amplitudes during transient passage.
Based on the simulation with both models, the diagrams in Figure 12 are provided. The diagrams contain just amplitudes of the simulations, normalized to the analysis without the damper.

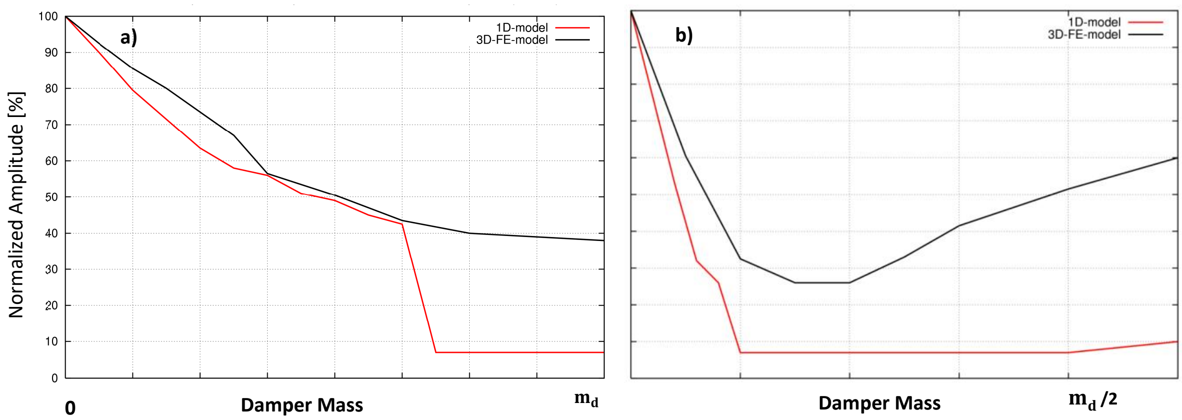

Figure 12. Normalized maximum amplitude over the rocking damper mass: a) 1F, b) 1T

The results show good agreements between 1D and 3D-FE-analysis for 1F until sticking. Discrepancies between 1D and 3D for 1T are related to the simple lumped parameter model. From the authors' point of view, these very different simulations give the numerical validation for the damping effectiveness of the rocking damping system. The finite element results in Figure 12 show that it is possible to damp simultaneously both mode shapes, 1F and 1T (Figure 13).

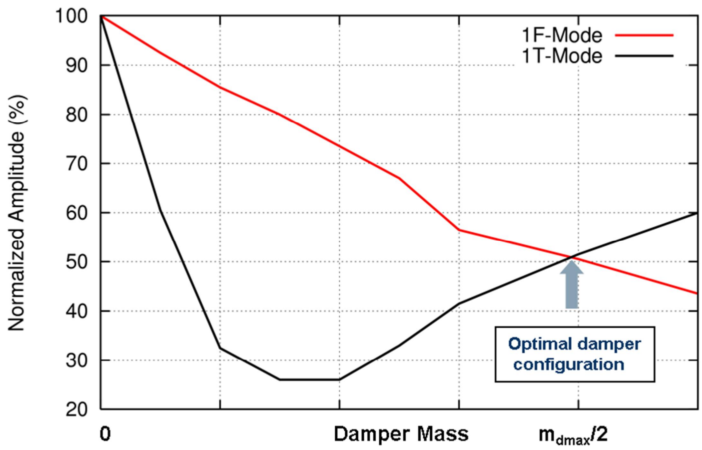

Figure 13. Evaluation of the optimal rocking damper configuration for simultaneously damping both mode shapes, 1F and 1T

For the analyzed blade, the evaluated damper mass is a realistic one – a rocking damper could be designed if necessary.

Summary of conclusions concerning rocking dampers:

- Independent of the excitation order
- Damping of more than one mode family simultaneously conditionally possible
- Robust concerning contact conditions
- Numerically validated on two different models
- Limited flexibility concerning rotation speed: it deals with one contact force similar to the state of the art damping systems

4 Impulse mistuning Systems

The idea of the Impulse Mistuning Systems goes back to the attempt to damp the blades and vanes by free moving bodies placed e.g. inside of the airfoils. The idea of the inside damping systems is not new generally, some references are given in Borufka et al. (2009), Hartung and Retze (2011) and Hartung et al. (2016). To make a classification of the inside damping systems for blades and vanes, the Figure 14 is prepared.

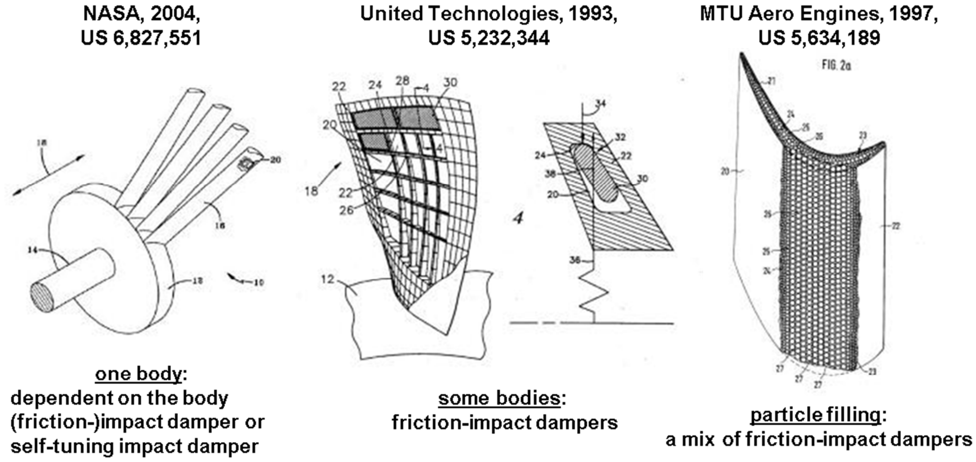

Figure 14. Some examples of inside, (friction)-impact based dampers from Borufka et al. (2009)

The most important features of the damping systems shown here are the geometry and the mass of the bodies, the number of bodies in a cavity and the geometry of the cavities. Dependent on all of this, the inside damping systems work as impact, friction-impact, particle or self-tuning impact dampers, as highlighted in Hartung and Retze (2011) and Hartung et al. (2016). In these papers was shown, that assumed only one body is placed in a cavity and for much smaller dimensions of the damping bodies and of the cavities another acting principle is active: mistuning of the resonance frequency due to impacts – "impulse mistuning". Impulse mistuning could be also described as a special kind of vibration-impact nonlinear energy sinks (VI-NES) with a large capability of energy absorption and negligible contribution of energy dissipation. The mass ratio of such VI-NES to the main system is <1%. There are also some other types of Nonlinear Energy Sinks: e.g. friction or non-linear stiffness based. The acting of the Nonlinear Energy Sinks lead to Targeted Energy Transfer (TET): one-way directed transfer of energy from a primary subsystem to a nonlinear attachment. References and more explanations are given in Hartung et al. (2016). In difference to the energy dissipation based damping systems, the impulse mistuning systems work for a larger interval of rotation speed (for all speeds below a critical value) in case of blade and has no dependency on the contact pressure in case of vanes. Additionally, every mode shape with enough movement at the cavity will be damped, independent of the direction of the motion – more than one mode family could be damped. In Hartung et al. (2016) damping of two mode shapes for a turbine blade was presented: from analytical prediction until experimental validation in a rotated rig. The validation in an engine demonstrator was mentioned. A vane Impulse Mistuning System with five bodies (mistuners) for a vane was analyzed in Hartung et al. (2016) as well. In this case, the robustness of the mistuning system concerning masses of the bodies and the gaps between the mistuners as well as the cavity walls was analyzed analytically and proved experimentally. The damped vane mode shape in Hartung et al. (2016) was a "1E cluster" mode (Figure 15a). Beside the cluster mode shapes so-called airfoil mode shapes could be HCF-critical as well. Because of design features of the vane clusters it is not possible to damp the airfoil mode shapes with a friction based damping system. For such a mode shape, "1T-Airfoil", experimental analysis of another vane cluster (Figure 15b) was performed at MTU recently and will be presented below for the first time. The test series consisted of five shaker tests of the vane cluster (Figure 15b): without dampers and with four impulse mistuning systems - two different designs of the impulse mistuning system with three different manufacturing approaches were tested. A special feature of the airfoil mode shapes is that a family of such mode shapes exist in the vicinity of each other instead a single resonance in case of the cluster mode shapes. So, first the identification of these was performed (Figure 16). Mode 3 is the most relevant for the engine. For this mode and the relevant excitation level, the highest amplitude reduction of 40% with one of the tested impulse mistuning systems was established (Figure 17). The second result of this test series is, that the impulse mistuning system produced with three different manufacturing technologies showed no significant difference in damping effectiveness.
Summary concerning Impulse Mistuning systems:

- Suitable for both, blades and vanes, respectively. vane clusters
- Action principle: frequency mistuning due to impulses, targeted energy transfer
- Independent on the excitation order
- Robustness concerning design parameters and production technologies proven

- Damping of more than one mode family simultaneously often possible
- Fully validated – including engine experiences, Hartung et al. (2016)
- A very large flexibility concerning the contact force

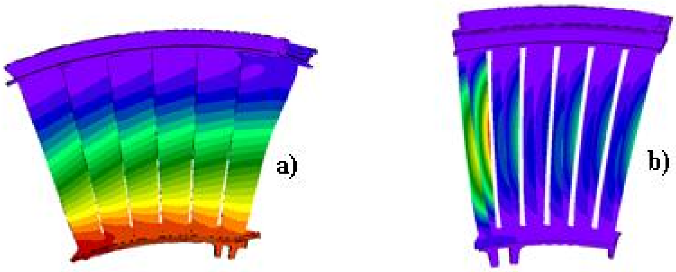

Figure 15. Analyzed vane cluster mode shapes a) "1E cluster", Hartung et al. (2016), b) "1T airfoil"

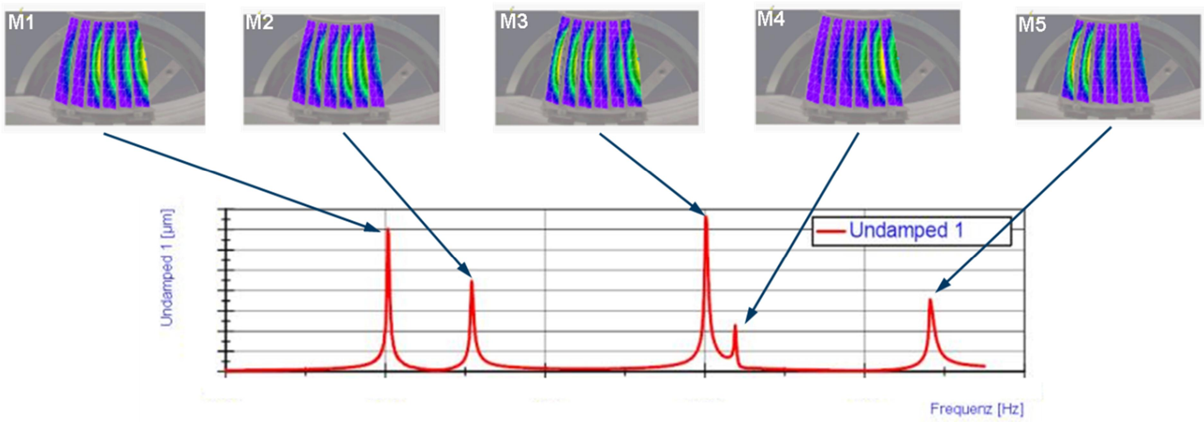

Figure 16. Measured and identified "1T airfoil" mode family

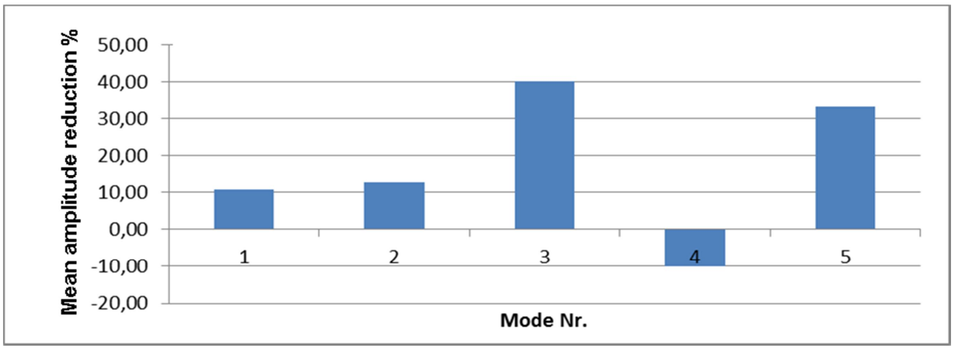

Figure 17. Measured mean amplitude reduction mode shape "1T airfoil" for the second impulse mistuning system averaged over all test samples manufactured using three different production technologies

5 Conclusions

Three different new damping systems independent of the excitation order are presented. All three systems are analyzed and validated using the different analytical and experimental methods. All three systems showed good capability of damping effectiveness. Insert and Rocking Dampers could be designed more flexible than under-platform dampers concerning simultaneously damping of more than one mode family but are limited concerning using for different rotation speeds. This restriction is of general quality because of the acting principle of energy dissipation due to friction. Impulse mistuning systems showed the highest effectiveness and flexibilities: concerning the rotation speed, mode families and production technologies.

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