DAMPING OF A COMPRESSOR VANE CLUSTER

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OVERVIEW

Because of smaller High Fatigue Strength of cast components compared to forged materials an additional damping system will generally be added to cast compressor vanes. To achieve this the inner shroud of a vane cluster will be separated for each airfoil and warped over a spring within an additional seal carrier. The spring acts as a mechanical damper because of friction contact to the inner shroud of the vane cluster. The separating of the inner shroud is required in order to make the relative motions in the join connections possible and hence to constitute the energy dissipation.

The essential design parameters of this damping system are the spring press force and friction coefficient which have to be adapted to the actual excitation amplitude. Usually only the damping of mode shapes with sufficient amplitudes on the inner shroud is successfully.

Some potentially critical mode shapes of a compressor vane cluster with a new designed spring (FIG 1.) were analyzed analytically and experimentally. The damping possibilities by different springs for different excitations were investigated. The comparison of analytical and experimental results shows very good correlations.

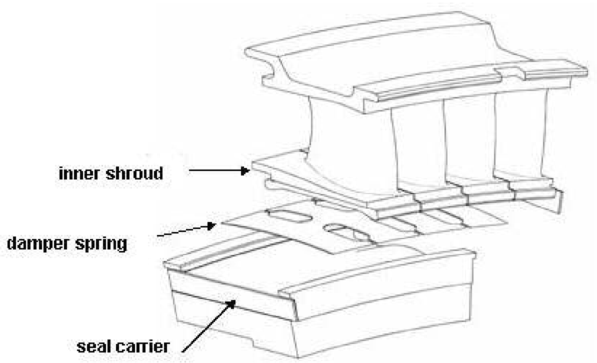

FIG 1. Damping concept

Some examples of analytical and experimental investigations of other designed vane cluster damping systems are published in [1] and [2].

1. INVESTIGATION ACCOMPLISHMENT

The mode shapes shown in FIG 2. were chosen for the measurements of the damping effectiveness. These mode shapes could be critically according to experiences concerning the resonance situation of an engine. While mode shapes 1-3 are fundamental, the mode shapes 4 and 5 are higher in frequency and show smaller motions at the inner shroud. The analytical dimensioning of the damper was done for mode 1.

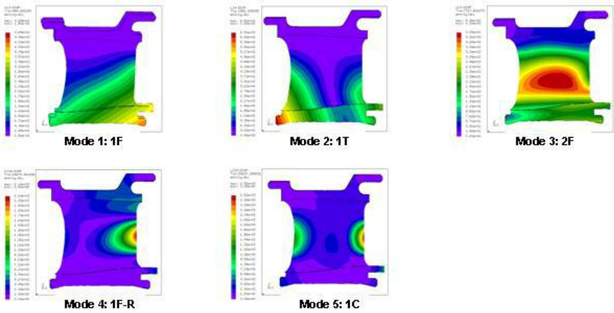

FIG 2. Investigated mode shapes

For the experimental vane damping evaluation a vane cluster of 4 airfoils was clamped on a shaker and excited in vertical direction (FIG 3.). The shaker tests were performed with harmonic excitations, the vibration velocity was measured contact less and the modes were identified using a scanning laser.

The measurements were performed with two different damper springs:

- damper spring with a nominal stiffness according to analytical dimensioning;
- damper spring with 50% higher stiffness.

FIG 3. Experimental setup

Two different levels for the excitation were chosen to simulate engine relevant stress levels in the airfoils (level 1 and level 2).

The analytical analysis was done for modes 1, 2 and 4. Two different analytical models were used: a relative simply multy body and a finite element model, FIG 4. All calculations for both models were done nonlinear in the time domain until steady state conditions were achieved.

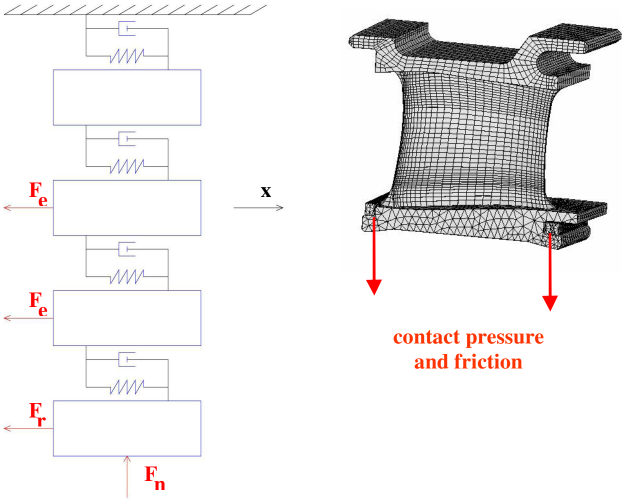

FIG 4. Rigid body and finite element models

2. ANALYTICAL RESULTS

The analysis of the first mode shape using the multy body model shows that the pressure forces of the spring cause nearly full damping, FIG 5. That could be approved by experiments. Also the frequency shift of the resonant frequency to lower values was predicted by the rigid body model.

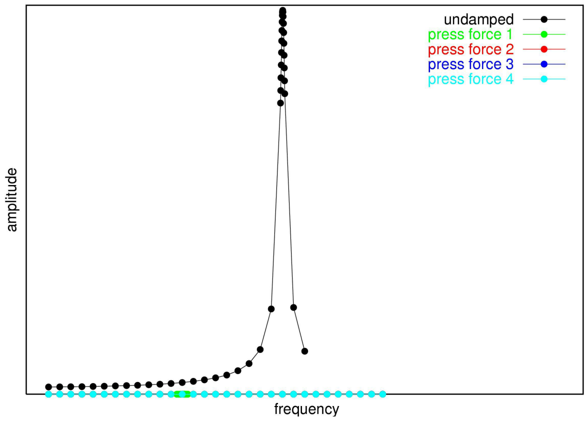

FIG 5. Forced responses for mode 1 calculated by the rigid body model for different pressure forces of the spring

The prediction of damping for mode shape 2 (1T) was smaller compared to mode 1 (FIG 6.), but also considerably high. This is confirmed by the experimental results. Also the prediction of nearly the same resonance frequencies for the damped and undamped configuration could be approved. The reaction force 2 was chosen for the experimental realization and the vibration amplitudes are depicted at the bottom body.

The calculations using the finite element model lead to similar results. FIG 7. shows the vibration in the time domain. Note: solutions in the time domain are depicted at characteristically nodes with maximum amplitudes for each mode respectively (red areas in FIG 2.) - so, these nodes are different for different mode shapes.

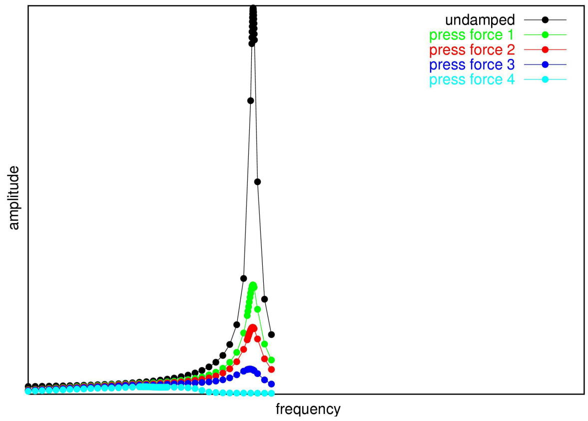

FIG 6. Forced responses for mode 2, calculated by the rigid body model for different pressure forces of the spring

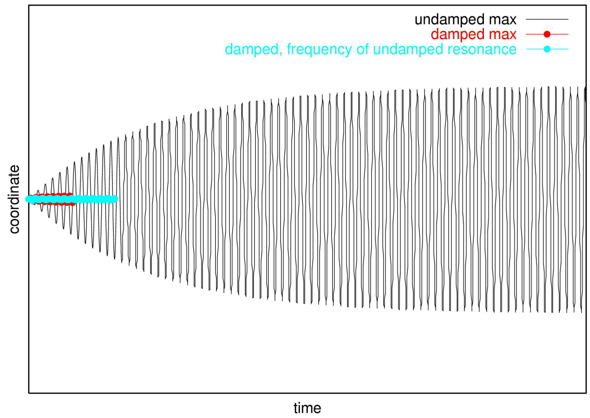

FIG 7. Solutions in the time domain, calculated using the finite element model for mode 1

Contrary to the rigid body model, the finite element model is able to analyze every mode shape. For this reason the finite element model was used to analyze an airfoil mode with very low motion at the inner shroud: mode 4, FIG 2. The results of the calculations showed that for the chosen pressure force no damping can be predicted (FIG 8.), although the experiments showed a small degree of damping.

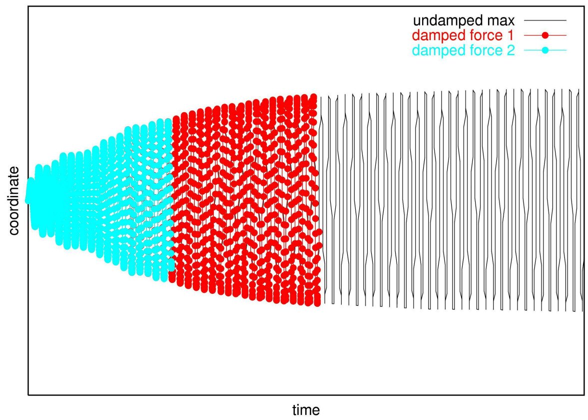

FIG 8. Solutions in the time domain, calculated using the finite element model for mode 4

4. EXPERIMENTAL RESULTS

For the verification of the results and for closer investigations of other, not analytically analyzed mode shapes, some measurements were performed on a shaker, FIG 3. The measurements were performed with different measurement systems and the repeatability of the results was checked by different built-ups on the shaker. Additionally, measurements on different vanes of the cluster were performed and compared. The following results should be presented as an example. If relevant the position of the vane in the cluster is given.

The worst measured damping effect for mode 1 (for the vane in the middle of cluster) is shown in FIG 9.

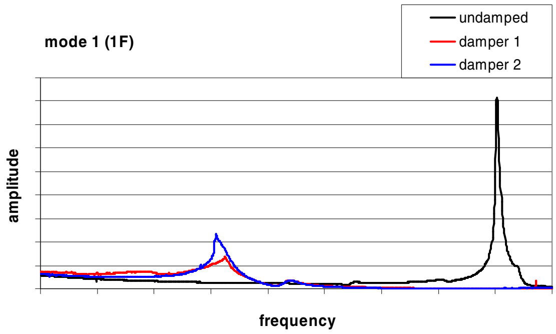

FIG 9. Forced responses for mode 1, measured at the vane in the middle of the cluster

The application of a 50% stiffer damper (in FIG 9. damper 2) leads to a smaller damping effect. Thus the chosen damper 1 is nearly optimal. The frequency shift to lower values is significant and was predicted analytically. The influence of the higher (double) excitation amplitude is depicted in FIG 10.

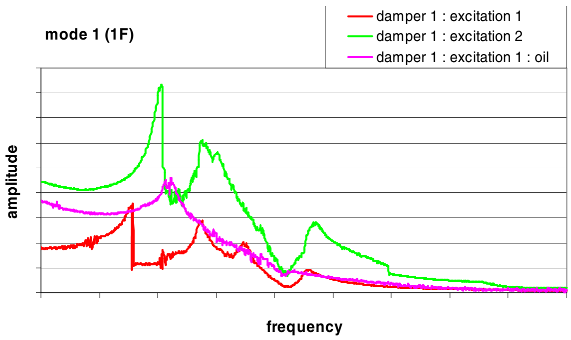

FIG 10. Influence of the excitation amplitude and friction coefficient, measured at a “border” of the cluster

In order to investigate the effect of a different friction coefficient, the seal carrier was lubricated in one test with oil. The influence on the damping was relative small, FIG 10.

FIG 11. shows the test results for the mode shape 1T. For this mode the coupling of adjacent vanes is obvious also for the undamped case (double peak). The damping effect is very good for all vanes, also for the stiffer damper - 90% amplitude reduction can be achieved.

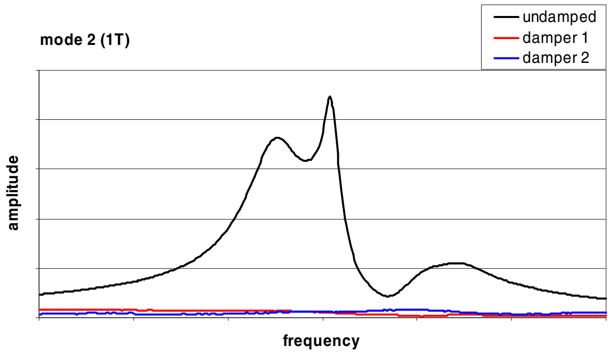

FIG 11. Forced responses for the mode 1T

Even for mode shape 3 (2F) the damping is very good (FIG 12.), although the maximum vibration amplitudes are at the airfoil, FIG 2. A shift of the resonant frequency to the right was observed. In FIG 12, the damping effect for vane 1 is depicted, it shows the best result for this mode shape. Even for the worst case a reduction of the amplitude by about 60% was observed.

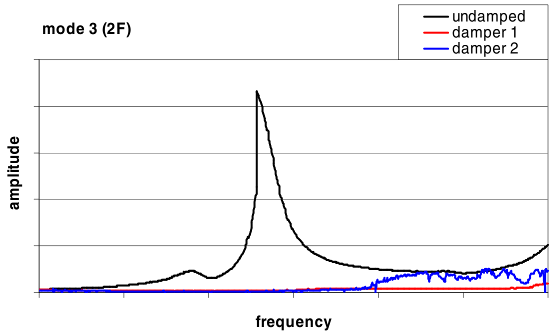

FIG 12. Forced responses for the mode 3(2F), a random vane

The characteristic of higher order mode shapes is an amplitude of less than 10% at the inner shroud compared to the amplitudes at the airfoil. Two of this mode shapes were investigated. For mode 4 (1F-R) a certain degree of damping could be observed, FIG 13.

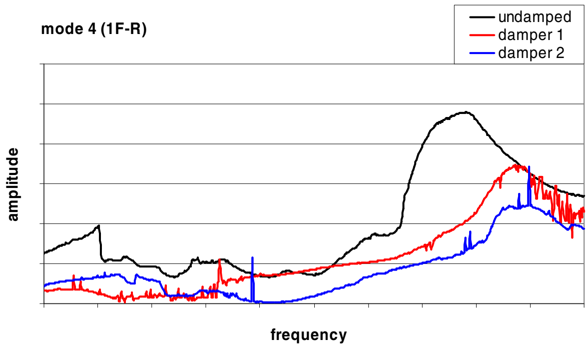

FIG 13. Forced responses for the mode 4(1F-R)

For mode shape 1C (FIG 14.) a definite statement is difficult. For the damper with smaller stiffness (damper 1) damping effect seams to exist. The measurements with the stiffer spring show only a very small damping effect. In some cases the amplitude of the damped configuration is higher than for the undamped configuration. This fact is well known and takes place if the contact pressure is to larger. However it's not critical because of small amplitude values for 1C in case of this vane. Examples like this are well known.

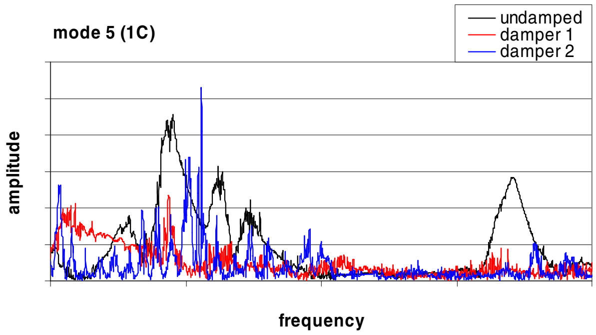

FIG 11. Forced responses for the mode 1C

6. CONCLUSIONS

The present investigations showed that:

- for the chosen design an effective damper can be realized and
- the analytically prediction of the damper effect is quite accurately.

As expected the damping of the fundamental mode shapes are quite effective. The influence of the 50% higher pressure force was marginally. The influence of the small variation of the friction coefficient is not significantly. An amplitude reduction of 90% can be achieved. The damping of high-frequency mode shapes is not effective because of small inner shroud motions. The prediction of amplitude reduction by finite element models for these mode shapes is also possible. The investigations for the repeatability of the measurement results were successful.

The analyzed damped vane clusters were built in an experimental aero engine. The damper design is proved successful, the results of the engine tests are comparable with the presented shaker tests results.

7. AKNOWLEDGEMENTS

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REFERENCES

[1] Petrov, E. and Peng, C.: "Prediction of Mechanical Damping for Core Compressor Blades and Vanes". Proceedings of the 10th National Turbine Engine HCF Conference, March 8-11, 2005, New Orleans, USA.

[2] Berruti, T. et al.: "Friction Damping of Interlocked Vane Segments: Validation of Friction Model and Dynamic Response". Proceedings of ASME TURBO EXPO 2002, June 3-6, 2002, Amsterdam, The Netherlands.