Experimental Measurements of the Particle Damping Effectiveness Under Centrifugal Loads

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Introduction

Particle damping is believed to be a promising loss mechanism for engine turbine blades, as some particle damping materials have the capability to withstand the extremely high temperatures inherent with operational aircraft engines turbines. The ability to provide damping under centrifugal loads however had never been shown experimentally. In this paper, initial measurement results from a series of particle damping configurations tested under true centrifugal loads are reported upon. The interplay of various parameters such as fill ratio, particle size, shape, and material on achieved damping levels were investigated with nine different particle damping configurations loaded up to 5,000 G's. Damping, in terms of frequency domain peak response reduction, was seen at this G loading for one configuration with no indication of reduction due to centrifugal load. This best performing particle damper, based on irregular tungsten carbide granules, was also proof tested to more than 50,000 G's with no noticeable degredation. While damping still remains to be shown at centrifugal load levels comparable with real engines, these initial test results show the particle damping is worthy of further consideration.

Why are new damping methodologies needed?

The design trend towards integrally bladed rotors (IBR) and a reduced use of shrouds is eliminating traditional platform and inter-blade dampers.

Thus the levels of blade mechanical damping is being reduced towards the level of material damping while at the same time, blades are being exposed to a more dynamically rich and harsh environment. While many technologies and materials exist for damping structures, no established damping material has the ability to provide damping between 500 deg F and 1800 deg F. A wide range of viscoelastic materials and configurations can be designed for under 500 deg. F (Johnson and Kienholz, 1982). Other loss mechanisms such as magnetic eddy current, viscous oils, shunted (or active piezoelectrics for that matter) are similarly limited to low temperatures. Enamels, a material that acts like a viscoelastic, only functions at high temperatures above 1800 F. There is thus a wide gap between 500˚ and 1800˚ F where no establish damping material can work. Mechanisms such as internal blade friction dampers or tuned mass absorbers can offer some damping but are subject to wear concerns or are limited to single modes.

Particle damping however, is capable of functioning over wide temperature ranges and modes. Work is currently ongoing at CSA Engineering to demonstrate the effectiveness of particle damping for non-rotating hot structure components. The biggest question however with regards to implementation in aircraft engines is “Can particle damping work under centrifugal loads?” These loads can reach 50,000 to 75,000 G’s in larger engines, and sometimes even higher in smaller ones. For particle damping to work, the particles must be capable of moving relative to one another. Of course, other issues exist as well, such as compatibility with blade manufacturing techniques, lifetime, wear, etc. but these only become important if it can be shown that particle damping can function under centrifugal loads at all.

Initial Centrifugally Loaded Particle Damped Test Results

In order to begin providing data that would address the question of whether or not particle damping could work under centrifugal loads, a series of nine different particle damping configurations were subjected to centrifugal loads over two days of testing during April 1998. All tests were performed at Test Devices, Inc. facilities in Hudson MA. These tests were intended to provide a direct answer to the above question as well as provide data that could be used in further developing analytical design tools similar to those discussed at last years HCF Conference (Olson et al., 1997).

As the goal of the tests was to subject the particles to as high of a centrifugal load as possible for an affordable price, a specialized blade-like test object, Figure 1, of 7075-T6 aluminum was fabricated. The test blade was sized to withstand up to 13,000 RPM, or the equivalent of 70,000 G’s, at the capsule test location.

Changing damping configurations was simplified by the use of a removable capsule, Figure 2, with an internal cavity 0.5” in diameter and 3/8” deep. Measurements of the inherent damping of the test object and mounted capsule were made by performing the standard tests with an fixed added mass that was equivalent to the nominal particle fill mass fixed to the otherwise empty cavity. The blade root was constrained roughly 4” from the spin axis and the blade protruded another 12” beyond to the blade tip. The blade was 8” wide at the base, 3” wide at the blade neck and remained a constant 0.150” thick. At zero RPM the first mode of the blade was approximately 45 Hz.

A variety of excitation and sensing options were considered. In the end, to achieve controllable excitation while rotating, the blade was instrumented with two piezoelectric patches which were operated out of phase to induce bending moments into the test blade. Four additional patches were used to measure the response of the blade as they provided a high voltage response signal as compared to the pre-amplified milli-volt traditional strain gage response. The utility of piezoelectric patch excitation and sensing is further seen by the ease with which mode shapes were tracked and excited. Figure 3 shows some example test results of a patches response to broad band excitation provided at several different RPM levels. The response of the piezoelectric patches was calibrated in a laboratory environment with a blade mounted accelerometer. These measurements allowed the direct correlation of measured piezoelectric voltage levels to accelerations at the particle damping cavity location.

Results

Measured frequency response functions from broad band excitation and time domain ring downs from narrow band sinusoidal excitation were used to extract estimates of the test objects damping with and without particle damping. Capsule added mass was nearly identical for all damped and un-damped cases to ensure mode similarity and help reduce variations in dynamic balance. During spin testing, the first and second bending mode and the first torsional mode of the test object were excited.

Nine different particle types and fill ratios were tested. Realized mass ratio of particles to blade was approximately 0.4%. All configurations were tested from 0 G’s centrifugal load to near 1,700 G’s in the first day of testing. The majority showed indications of loosing their damping effectiveness with RPM. The best performing damper as well as the comparative un-damped baseline were further tested to 5000 G’s on the following day with an encouraging outcome as shown in Figure 4. These results are for the second bending mode and were averaged from broad band tests performed at three excitation levels. One can see that as the G loading increases there is an initial decrease in damping performance (the negative value around 1500 G's indicated there is some scatter in the damping extraction methods) followed by a gradual increase in particle damping effectiveness with RPM. Time limitations prevented further tests except for a high speed, limited duration exposure of the particle damped capsule to slightly more than 50,000 G's.

Conclusions

This work was motivated by the need to answer the fundamental question of "Can particle damping work under centrifugal loads at all?" These initial experimental results show that particle damping can indeed work under centrifugal at least up to 5000 G's. Based on these promising results, further investigation was felt to be warranted. Phase II of the effort is now underway with the express goal of testing as many different particle damping parameters as possible. Work is currently focused on defining the test blade, representative capsule configurations and in specifying the required spin facility characteristics.

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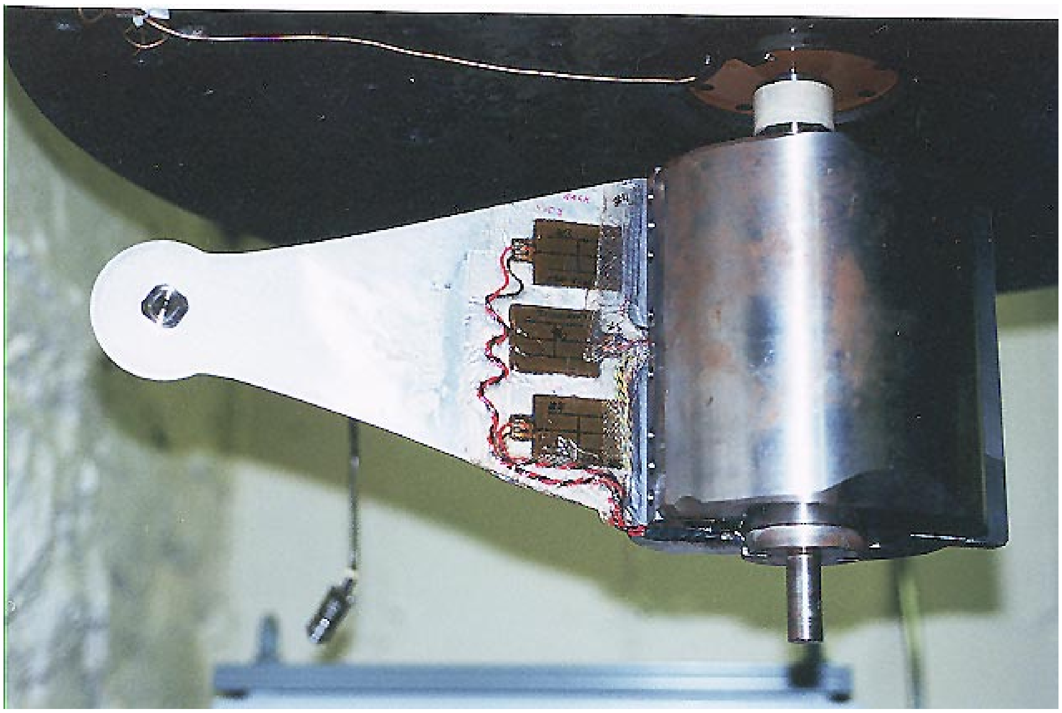

Figure 1: Test blade mounted in the spin test hub

Figure 2: Open and closed capsule shown with a penny as scale

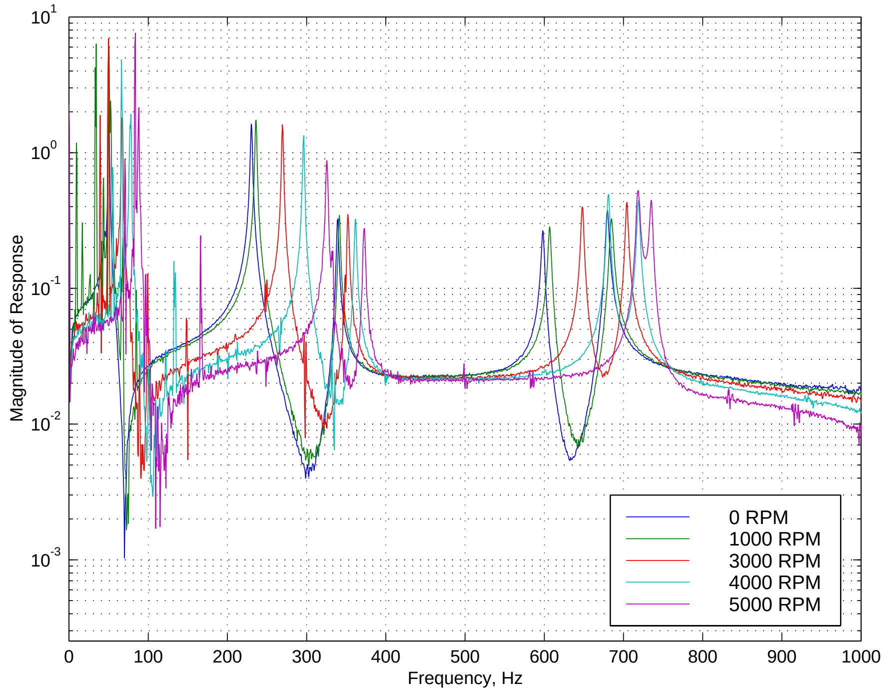

Figure 3: Frequency response function of representative piezoelectric patch to pseudo-random broad band excitation at several RPM levels.

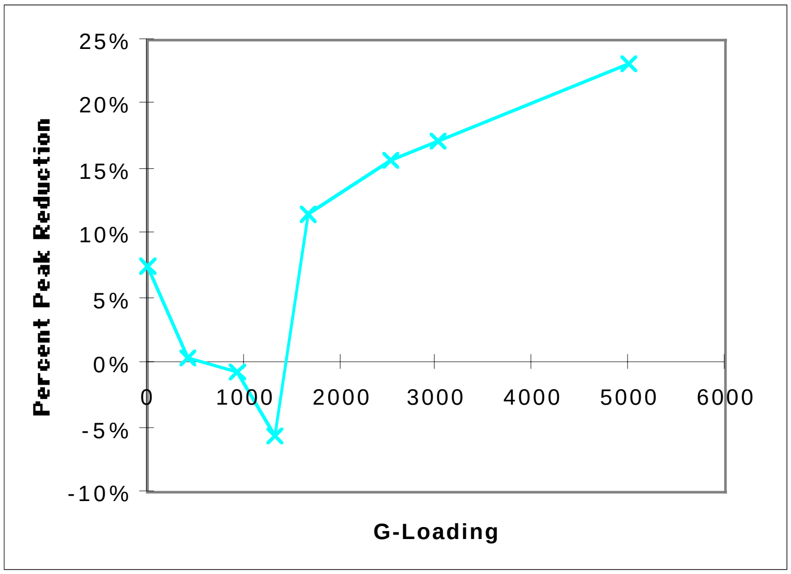

Figure 4: Example percent peak reduction versus centrifugal load for the second bending mode