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EXPERIMENTAL EVALUATION OF THE EFFICIENCY OF GAS TURBINE ENGINE PARTS DAMPING WITH DRY FRICTION DAMPERS USING LASER VIBROMETER

A.Balakirev*, B.Bolotov*, A.Golovkin*, M.Nikhamkin*,
N.Sazhenkov*, L.Voronov*, I.Konev*
**Perm National Research Polytechnic University, "Aircraft Engines Department*

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Abstract

The article discusses problem of experimental data obtaining for verification of mathematic models of dry friction dampers for gas turbo engines (GTE) units. The method of damping characteristics determination was developed for three critical units of GTE: hollow fan blade, compressor blisk and turbine blades. The paper describes the methodology and results of experiments.

1 Introduction

There is always necessary to solve the problem of reducing dynamic stresses caused by parts vibration and provide critical parts resistance to fatigue failure during the creation of an aircraft gas turbine engines (GTE). GTE blades which operate under static and dynamic loading, and always at risk of foreign object damage are the mostly subjected to fatigue failure. The problem of reducing dynamic stress is a key task in terms of reliability and engine lifetime. Due to the tendency of the engine weight reduction and increasing of engine parts loading, this problem continues to be relevant [1 - 6]

GTE blades have a dense spectrum of natural frequencies. For example, the wide hollow fan blade may have more than 50 of mode shapes in the range up to 3000 Hz [7,8]. For this reason is not always possible to avoid resonant vibration of the engine elements within the whole operating range. One of the main ways to prevent destruction during resonant oscillations is to provide structural damping. To prevent this in to construction installing the dampers which operating on the principle of energy dissipation in pairs of dry friction (for example, see [1])

Mathematical modeling of dry friction dampers using modern numerical methods allow to describe complex phenomena in dry friction damper contact [9-11]. Assumptions on which the mathematical models are built require experimental verification.

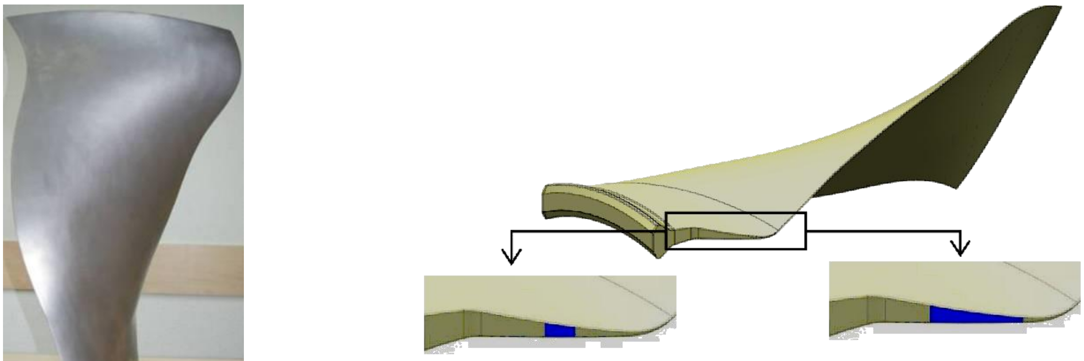
In this regard, the aim of this work is the obtaining amount of reliable experimental data about friction dampers efficiency and its basic laws for damping typical GTE elements for subsequent use of the this data for verification of computational models.

2 Methodology of investigation

2.1 Investigation objects

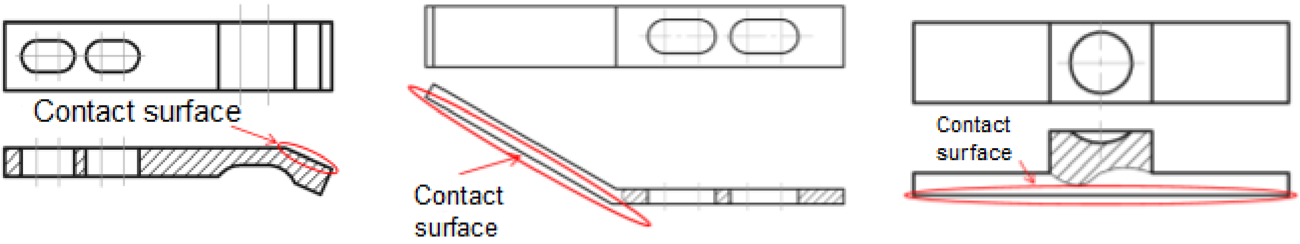
In this study, the following GTE elements were considered:

- Full-scale hollow titanium fan blade with the corrugation inside (Figure 1). Blade is made of three sheet metal parts: backrest, a trough and corrugation filler combined by superplastic forming and pressure welding method.
  For fan blade 3types of dampers with different values of the contact surface and stiffness were studied (Figure 2). Dampers installed on a smooth flat surface near the blade lock in side of the trailing edge.



a) b)

Figure 1 – The hollow fan blade with corrugation filler and contact areas for damper No. 1– (a) and for dampers No.2 & No.3 – (b)



a b c

Figure 2 – dampers for fan blade

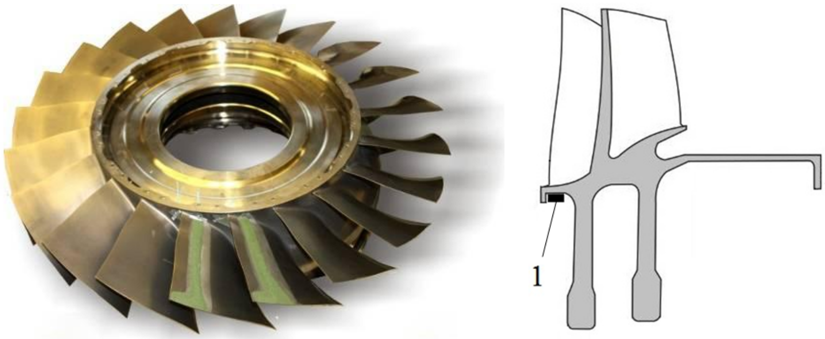
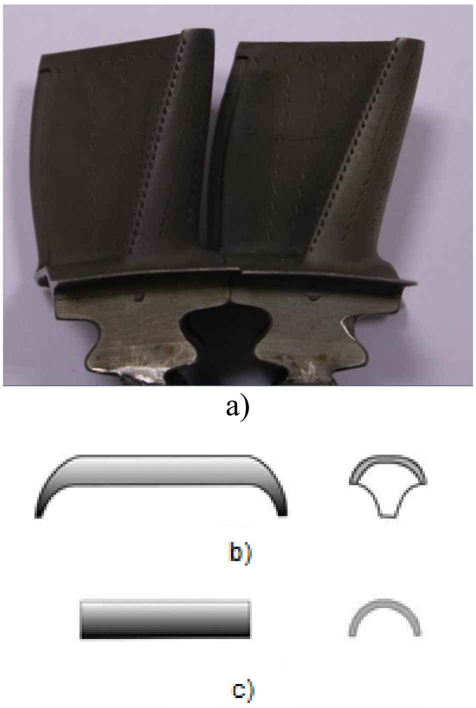


Figure 3 – HPC first stage blisk of gas turbine engine and the location of the ring damper installation – (1)

a b

Figure 4 – Ring dampers No. 1 – (a) and No. 2 – (b) for blisk and its expanding devices

- The full-scale blisk of the first stage of HPC (Figure 3). The blisk consisting of a disc part with double canvas and wide rim, and of 21 wide blade with complex profile. Blisk made of a titanium alloy.
  For blisk, there two dampers been tested which looking like a cut rings with different cross sections and different expanding devices (Figure 4). Dampers installed in a cylindrical groove under the Blisk rim from the side of the inlet flange.
- Cooled rotor blades of the first stage of high-pressure turbine made of nickel-base alloy with underplatform damper (Figure 5). There were considered two dampers. Damper number 1 is a curved plate, which shape follows the shape of underplatform cavity formed by two adjacent shelves of working blades. Damper number 2 has a shape of a cylindrical shell.


a)

b)

c)

Figure 5 – Blades of a high pressure turbine (a) and dampers No. 1 – (b) and a No. 2 – (c)

2.2 The method of tasting

The method of the dampers efficiency developed within experimental modal analysis using a scanning laser vibrometry [12-14]. The method involves the excitation of the object vibrations and the registration of vibration velocity in the scan points with following determination and analysis of the transfer functions matrix *[H]*. Determination of natural frequencies, mode shapes and logarithmic decrements comes to this matrix analyze, each element $H_{ij}(\omega)$ of which is a function of the oscillation frequency ω and represents the frequency response (FRT) [15]:

$$H_{ij}(\omega) = X_i(\omega) / F_j(\omega) \qquad (1)$$

Where $X_i(\omega)$ frequency function of vibration velocity at a scan point i excited by the force $F_j(\omega)$, attached at the point j.

Evaluating the dampers efficiency is based on the quantitative determination of the logarithmic decrement δ of the test object with installed damper. Decrement calculated by the width of the resonance peak at the frequency response averaged over the ensemble of scan points:

$$\delta = \pi \frac{\Delta f}{f_n} \qquad (2)$$

Where Δf – the width of the frequency response band with a back-3dB from the value of the local maximum in the resonant mode, f_n – resonant frequency.

The method realized by using three-component scanning laser vibrometer Polytec PSV-400-3D, which controls the excitation of vibrations and providing vibration velocity registration at the scanning grid. The main advantage of scanning laser vibrometer consists in possibility of non-contact measurement of the three components of vibration velocity in a large number of scan points.

Method of fixing research object and excitation of oscillations is different for all three investigated structures. For fan blade fixing performed on a special device rigidly fixed on the basis of the shaker. The excitation of oscillations was performed using a shaker LDS M850. The installation diagram is shown in Figure 6.

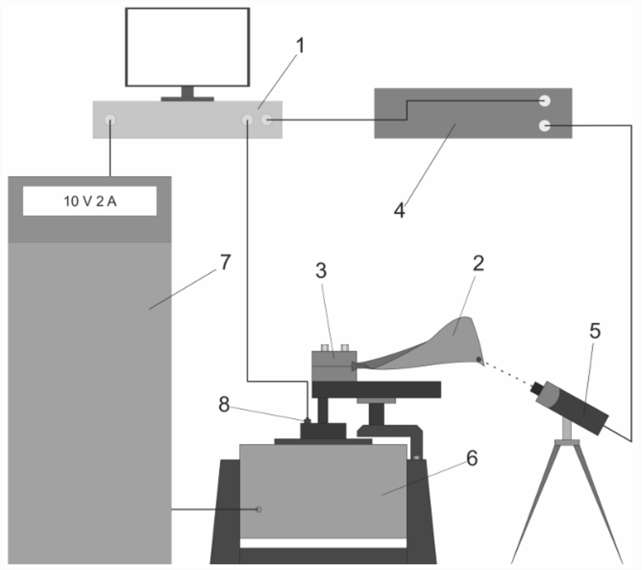


Figure 6 – Schematic of the experimental rig: 1 – vibration control system, 2 – studied fan blade, 3 – a device for testing the fan blades, 4 – controller of the laser head, 5 – laser vibrometer, 6 – LDS shaker M850, 7 – amplifier, 8 – Accelerometer PCB Piezotronic.

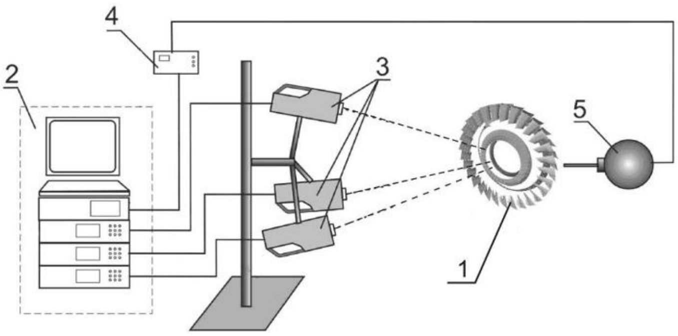


Figure 7 – Experimental rig for blisk tasting: 1 – investigation object, 2 – laser vibrometer control system, 3 – laser scanning heads, 4 – amplifier 5 – shaker.

For HPC blisk tests were conducted for the two fixing schemes: When blisk rigidly fixed to the rear flange on a special building berth and when hanged on an elastic suspension. Excitation was performed by using a miniature piezoshaker stuck to the disk part of the blisk. The experimental rig is shown in Figure 7.

Investigation of the damping in the block of turbine blades performed at the rig, which includes a special block of two full-scale turbine blades with underplatform damper between them (Figure 8). Blades 1 are welded to the base 2, which simulating the disk. Welded connection eliminates the damping in the lock of blades to make the underplatform damper the main element determining the structural damping. The block of blades fixed in a clamping device by the side surfaces of the base.

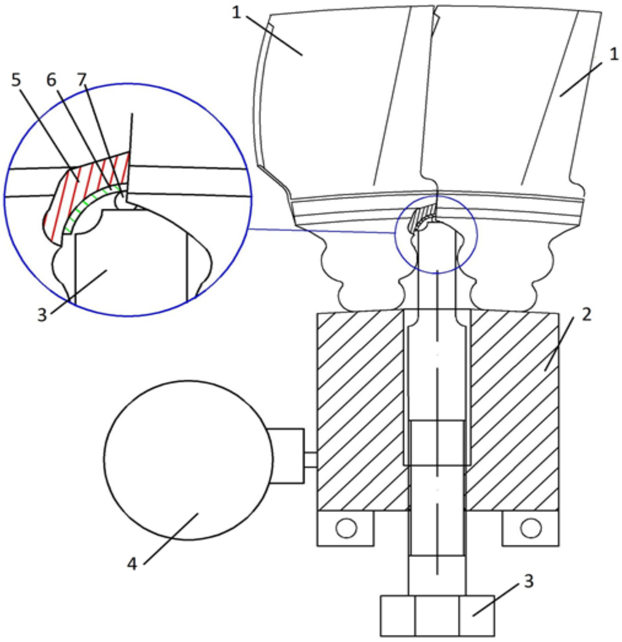


Figure 8 – The test unit blades: 1 – blade, 2 – base, 3 – clamping screw, 4 – shaker 5 – shelf blade 6 – damper, 7 – steel ball

In all the tests, the dampers were provided special devices to control and measure pressing force of dampers modeling the centrifugal load.

3 Experimental results

In all the tests, the dampers were provided special devices to control and measure pressing force of dampers modeling the centrifugal load.

To test the damping efficiency at first there were performed experiments with non-damped constructions. According to the obtained frequency response in the studied range, for each natural frequency by the peak width was determined the logarithmic decrement δ. Then on the test objects were installed dampers with a predetermined pressing force, and the next series of experiments was carried out. On the obtained frequency response by visual analysis of modal shapes was calculated natural frequencies of the object corresponding to the frequencies of undamped structure, and then was calculated the change of the logarithmic decrement.

So the increasing of the logarithmic decrement for the studied dampers, for the investigated mode shapes amounted to a maximum of 7.81 times for the fan blade, to 13 times for blisk and to 18 times for the system of turbine blades.

However, these values are largely dependent on their own mode shapes of the research object, which is particularly well illustrated by example with blisk. Figure 9 is showing a diagram of the logarithmic decrement changes depending on the number of mode shape for the damped blisk with the maximum and minimum damper pressure on the background of undamped system.

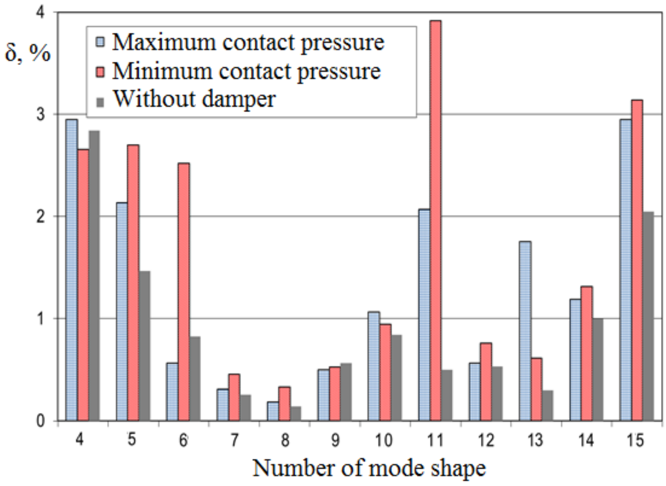


Figure 9 – Blisk with damper No. 1 logarithmic decrement for various mode shapes

As can be seen from the diagram, tested damper was not effective for the mode shapes (forms numbered 7-10 and 12 on the diagram) under which mostly fluctuated only blisk blades, and the contribution of the disc part was small. However, for mode shapes with mainly fluctuations of a disk part (Numbers 4-6, 11, 13-15, in the diagram) damper looks pretty effective.

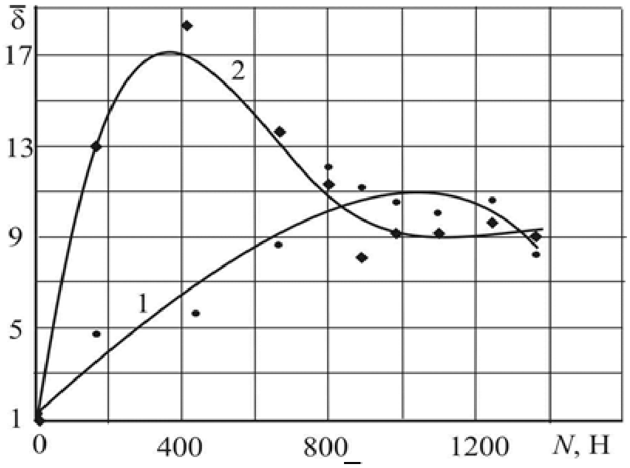


Figure 10 – Parameter $\overline{\delta}$ dependence on the contact load on the damper N for dampers No. 1 (1) and No. 2 (2).

Figure 10 shows the dependence of $\overline{\delta}$ (the ratio of the logarithmic decrement of blades with damper to the logarithmic decrement of blades without damper) on the magnitude of the simulated centrifugal load on the damper *N* for investigated block of turbine blades in one of the mode shapes. With increasing of force N the logarithmic decrement increases and reaching a maximum value at N = 400-900N and then somewhat reducing. The behavior of this dependence confirms the known ideas about reducing of the damping efficiency due to reducing of damper mobility zones square relative to blade flanges while the load increases.

During the tests the dependence of damping efficiency on the excitation level of the object were evaluated. Figure 11 shows an example of the obtained dependence of fan blade damping parameter for the three levels of excitation to one of the mode shapes, where *σ* - amplitude level of the stress in the blade lock. Damping characteristics significantly changes with the change of system excitation level. This is clearly seen nonlinear dependence with the explicit function extremes as in the case of changing of the pressing force of the damper.

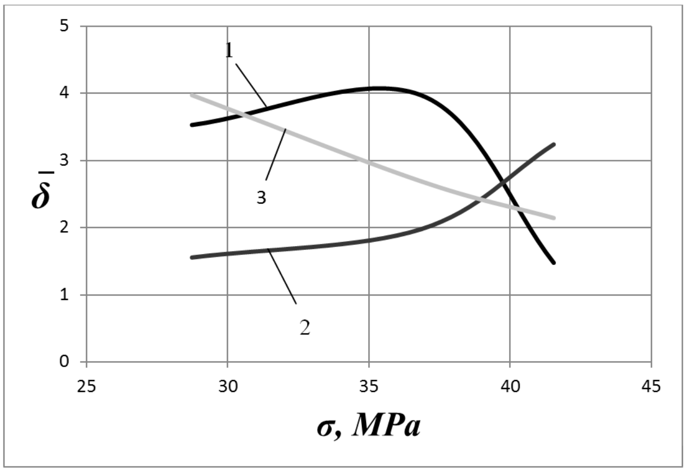


Figure 11 - Comparative characteristics of undamped fan blade with system with a damper No 3 by the excitation load, the bending vibration mode, the forward stroke. Damper contact pressure (1) – 10kg, (2) – 20 kg, (3) – 37 kg.

The experiments demonstrate the effectiveness and applicability of dry friction dampers for damping GTE elements. Obtained results will be used to verify mathematical models of dry friction dampers, which will develop the most effective damper design for the main units of GTE and optimize their parameters.

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