Experimental study on active damping of compressor blade forced vibrations using piezoelements

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Abstract. This paper is dedicated to active damping of gas turbine engine blades forced vibrations using the capabilities of piezoelements. Experimental investigations are carried out with a non-rotating compressor blade, rigidly clamped on a shaker. Vibration damping was carried out by using of feedback based on a piezoelement pair fixed on the blade surface and used as a sensor and an actuator. A significant decrease of resonant stresses in the blade is shown.

1. Introduction
The problems of vibration durability of gas turbine engines blades are well studied [1-2], however, the blade failures due to increased alternating stresses continues to occur, significantly complicating the operation and reducing the economic performance of engines.

It is necessary to find newer methods of damping to ensure low- and high-cycle fatigue, especially at resonant and near-resonant modes of details operation. Besides the passive methods consisting in energy dissipation of diverging vibrations by various types of friction forces [2] the attention of researchers is aimed at finding active methods that would prevent the appearance of increased stresses. One of such methods is using the electromechanical devices (piezoelements) based on the piezoelectric effect of certain materials.

The piezoelectric effect is that under the influence of applied voltage to a piezoelement (PE) its size changes and, conversely, the deformation of PE leads to the appearance of an electric charge on it.

It is possible to use this effect to make a PE, which is attached to a structure, deform in such a way as to compensate for the vibrations of this structure.

Many studies are dedicated to theoretical and practical problems of the active vibration control using PE [3-5]. Studies are mainly directed on researching the influence of PE location, parameters and configurations of damping circuit, control laws to the efficiency of vibrations damping for various objects.

In the area of the active damping of forced vibrations of gas turbine engine parts, there are works [6-10] showing the effectiveness of PE-based damping methods.

In this work, the experiments were carried out on a full-scale blade using bimorph PE combined into a feedback system with the possibility of phase shift and gain of the control signal. Also, when using PE as an additional source of excitation, the new data, which may be useful in fatigue tests, were obtained.
2. Experimental setup

Active damping of a blade vibrations with PE is meant as the organization of an electromechanical system (blade → PE-sensor → signals processing unit, amplifier → PE-actuator) so that vibration impact of PE-actuator glued on the blade weakens the resulting values of the blade resonant stresses and displacement amplitudes. Figure 1 shows the experimental setup scheme and its basic elements.

In the works in which experimental studies of active piezo-damping are carried out, the experimental setup schemes contain the following main elements: an object of damping, PE-sensor, signals processing and formation unit, PE-actuator [3-5].

In this work tests were carried out using electrodynamic shaker with a digital control system. The laser vibrometer was used to register the blade tip displacement under forced vibrations, and a sensor strain – to record the alternating mechanical stresses amplitudes.

The amplitude amplifier has the ability to increase the amplitude of the input voltage signal in required frequency range without changing of the phase between input and output signal. A frequency filter was used to control the phase of input signal from piezoelectric element. The filter has the ability to set phase between input and output signal either 0 ° or 180 °.

The PE-sensor wires are connected to the active resistance, which, at some value of resistance and due to the electrical properties of PE, allows the phase of the voltage to be shifted by -90 °.

During active vibration damping, a signal from the PE-sensor passed through the active resistance is fed to frequency filter input. Further, the signal is processed and the signal from frequency filter output, where it can change phase by 180 °, is fed to amplifier input, where it can be amplified in the range from 1 to 90. The amplified signal is fed to PE-actuator pasted on the other side of the blade, operating at phase shift of -90 ° in the active vibration damper mode, and when the phase shift of +90 ° – in the «anti-damping» mode.

The PE-actuator vibrating under the action of the control signal is the same source of vibration excitation for the blade as the shaker. Therefore, for effective action on the blade at resonance, it is necessary to set the control signal phase that differs from the signal phase of PE-sensor by ±90 °. It is obviously, the vibrations of PE-actuator and blade occur with the frequency of excitation.
Figure 2 shows the investigated compressor blade with two glued PE. In tests bimorph type PE with dimensions of $40.0 \times 12.0 \times 0.6$ mm are used.

The PE placements were selected on the basis of a preliminary calculation of natural frequencies and vibration modes so, that these are the areas of increased deformations for the blade first bending mode of vibration. PE on the blade pressure side is a sensor, PE on the suction side is an actuator.

Figure 3 shows the investigated compressor blade mounted on the shaker rig with two glued PE connected to each other by the electrical network described above. Using the shaker the forced vibrations were excited in the blade, rigidly fixed on it, with passage through the natural frequency of the first bending mode of the blade vibrations at a constant rate of increase of the exciting frequency.

Figure 2. Compressor blade with strain sensor and PE, view from the pressure (left) and suction (right) side.

Figure 3. Compressor blade with PE on shaker. 1 – compressor blade; 2 – PE on suction side; 3 – PE wires; 4 – shaker table; 5 – shaker rig; 6 – accelerometer; 7 – laser vibrometer reflector (for measuring blade tip displacement); 8 – strain sensor wires.
When processing the test results, special attention was paid to the accuracy of measurements of the blade tip displacements range and alternating mechanical stresses at the resonant frequency of the blade without PE, with switched on/off PE, as well as the reliability of measuring the resonant frequency of the system for different experimental conditions with varying excitation frequency.

3. Test results

Figure 4 shows the obtained resonant curves of displacement ($2A$) of the blade tip and alternating mechanical stress ($\sigma_a$) amplitudes at the input edge near the compressor blade root for different cases: without PE, with switched off PE, for active damping and «anti-damping» with different values of a feedback coefficient $k$ and constant value of vibration acceleration of the shaker $a = 4.9 \text{ m/s}^2$ and $a = 9.8 \text{ m/s}^2$. The value $k$ is defined as the ratio of the PE-actuator voltage amplitude and voltage amplitude received from PE-sensor. Active «anti-damping» is carried out by the joint unidirectional action of the exciting load and the PE-actuator, which leads to increase displacements range and alternating mechanical stresses in the blade.

![Blade displacement and mechanical stress curves](image)

*Figure 4.* Experimental resonance curves of compressor blades without PE, with switched off PE, with active damping and «anti-damping». 
As can be seen from figure 4, with using PE and vibration acceleration of the shaker \( a = 4.9 \text{ m/s}^2 \) and \( k = 30 \) (damping mode) the amplitude range of resonant displacements of the blade tip decrease from 2.26 mm to 0.25 mm and the resonant mechanical stresses from 83.0 MPa to 8.8 MPa, in comparison to a blade without PE. With vibration acceleration of the shaker \( a = 9.8 \text{ m/s}^2 \) and \( k = 18 \) the amplitude range of resonant displacements of the blade tip decrease from 3.38 mm to 1.26 mm, and the resonant mechanical stresses from 122.1 MPa to 43.8 MPa.

Such a decrease depends on two factors:
- additional rigidity in a structure and features of PE mounting;
- active damping directly.

Moreover, for \( a = 4.9 \text{ m/s}^2 \), the resonant mechanical stresses decreases by \( \sim 40 \text{ MPa} \), and for \( a = 9.8 \text{ m/s}^2 \) – by \( \sim 33 \text{ MPa} \) only due to active damping.

In the «anti-damping» mode, the increase of registered values was recorded.

4. Conclusions

Based on the experiment results, the fundamentally possible efficiency of forced vibrations damping of gas turbine engine compressor blade using capabilities of piezoelements is confirmed.

With the vibration acceleration \( a = 4.9 \text{ m/s}^2 \) and \( k = 30 \), the resonant stresses are reduced by \( \sim 85 \% \) when the blade oscillates on first bending mode, in comparison to a blade without PE for initial stress amplitudes of 83 MPa.

As the level of exciting load increases, the efficiency of active vibration damping decreases, which can be explained by the limited ability of its PE to withstand higher loads. Another type of PE, more powerful, could probably be more effective.

After gluing PE on the blade, the frequency of resonant vibrations increases and the amplitude of the blade tip displacement and resonant mechanical stresses decreases due to the introduction of additional rigidity into the structure and mounting PE features.

In the active «anti-damping» mode the increase of recorded parameters confirms the correctness of chosen vibration control scheme using PE. The new data on active «anti-damping» mode, which may be useful in fatigue tests of various objects, were obtained.

Further studies will focus on the issues of integrating PE into a structure of engine elements (stator and rotor blades, casings, rotor bearings).

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References

[1] Ayrapetov E L et al 1980 Kolebaniya mashin, konstruktsiy i ikh elementov (Vibratsii v tekhnike: Spravochnik v 6-ti t ed V N Chelomey) vol 3 ed F M Diementberg and K S Kolesnikov (Moscow: Mashinostroenie) p 544 (in Russian)
[2] Bauer V O et al 1981 Dinamika aviatsionnykh gazoturbinykh dvigateley ed I A Birger and B F Shorr (Moscow: Mashinostroenie) p 232 (in Russian)
[3] Akopyan V A, Rozhkov E V, Soloviev A N and Shevtsov S N 2007 Nano- i Mikrosistemnaya Tekhnika 10 pp 36–61 (in Russian)
[4] Kim S J and Jones J D 1995 J. Intelligent Material Systems and Structures vol 6 pp 610–23
[5] Gundage V V and Sonawane P R 2016 Int. J. Research in Engineering and Technology vol 2 pp 56–65
[6] Duffy K P, Choi B B and Provenza A J 2012 Proc. ASME Turbo Expo 2012 (Copenhagen, Denmark) p 11
[7] Schwarzendahl S M, Szwedowicz J and Neubauer M 2012 Proc. ASME Turbo Expo 2012 (Copenhagen, Denmark) p 10
[8] Kelley C R and Kauffman J L 2017 J. Intelligent Material Systems and Structures vol 28 (16)
[9] Kohler R, Kaletsch C, Rinderknecht S and Marszolek M 2011 20th Int. Symp. Air Breathing Engines (Gothenburg) vol 3 pp 1675–85

[10] Jiuzhou L, Lin L, Pengcheng D and Chao L 2016 Proc. ASME Turbo Expo 2016: Turbomachinery Technical Conf. and Exposition (Seoul) p 12