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The Study the Vibration Condition of the Blade of the Gas Turbine Engine with an All-metal Wire Rope Damper in the Area Mount of the Blade to the Disk

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Abstract. Improving the reliability of modern turbine engines is actual task. This is achieved due to prevent a vibration damage of the operating blades. On the department of structure and design of aircraft engines have accumulated a lot of experimental data on the protection of the blades of the gas turbine engine from a vibration. In this paper we proposed a method for calculating the characteristics of wire rope dampers in the root attachment of blade of a gas turbine engine. The method is based on the use of the finite element method and transient analysis. Contact interaction (Lagrange-Euler method) between the compressor blade and the disc of the rotor has been taken into account. Contribution of contact interaction between details in damping of the system was measured. The proposed method provides a convenient way for the iterative selection of the required parameters the wire rope elastic-damping element. This element is able to provide the necessary protection from the vibration for the blade of a gas turbine engine.

1. Introduction

Improving the reliability of modern turbine engines is actual task. This is achieved due to prevent a vibration damage of the operating blades of turbo machines. The main factor, which limits the level of vibration, is a damping capacity of the blades.

At the Department of construction and design of aircraft engines and in the laboratory "Vibration resistance and reliability of the engines and aircraft systems" for many years studied the problems with damping of the gas turbine engine blades. There many designs of damping elements from tapes, wire ropes and the pressed wire material MR, used in modern gas turbine engine have been developed. This work engaged Ulanov A. [1, 2], Frolov V. [3], Ponomarev Yu. [4], Ermakov A. [5], Belousov A. [6], Chegodaev D., Lazutkin G. [7], Kotov A., Davydov D. and many others [8]. Among the publications of foreign scientists, who are engaged in similar studies, can be distinguished [9-14].

Elastic-damping elements made of wire rope are common in aeronautical engineering and transport engineering. Their work is based on the structural damping. The main advantages of these elements is a good load-bearing capacity, resistance to temperature and aggressive environment, high damping, compact size and long life cycle [1, 4]. The metal wire rope can be laid separate segments or in one piece in the form of a serpentine.

In [6] given the various schemes of fixing of the compressor blades using elastic-damping elements. Schemes can be distributed into three groups: I. The disc of the impeller and the blade of a compressor interact only through the elastic-damping element; II. The design of the root attachment of blade and the retaliatory groove of disk provides active work the elastic-damping element, located under the base of the root attachment. III. The elastic-damping element is established under the base of the root attachment of blade with virtually no change in the retaliatory groove of disk.

The purpose of this work was to create a model and development method of calculation of vibration of the turbine engine blade with a wire rope element installed in the root attachment of blade.



The proposed method allows to achieve collaboration a system "disk - blade" (is used the contact interaction) and elements from the wire rope in the same model. The finite element method and the method of Euler-Lagrange are used. To calculate is required to create a model that takes into account the joint work a relatively "rigid" attachment (having some flexibility) and more "soft" the elastic-damping elements. At that, "rigid" attachment has little damping in comparison with more "soft" elements. The 3D model of the blade was created in the CAD-package [15]. In this paper we used the software ANSYS. We used a finite element COMBIN14 for modeling the elastic-damping elements. We used contact interaction to simulate the "rigid" attachments. The study of the application of a contact in ANSYS to simulate structural damping is dedicated to a number of foreign works [16, 17].

2. Materials and methods

The method uses the same finite element model of the compressor blade with three modifications that depend on attachment conditions. The diagram (Fig. 1) shows a condition of attachment and the main parameters, that you must to calculate for the simulation.

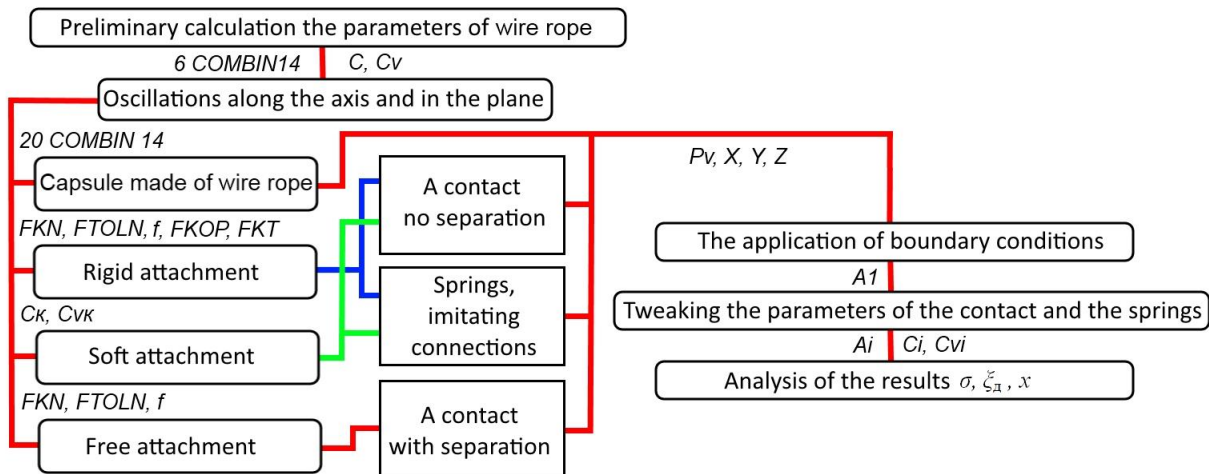


Figure. 1. Block diagram of the algorithm for calculating the compressor blade vibrations with a wire rope element in the root attachment

The desired type of attachment must be selected:

1. The root attachment of blade from all sides surrounded by elements from the wire rope. The blade interacts with the disk of a rotor through "the capsule".

For the following three types of attachments the elastic-damping element is mounted only at the base of the root attachment of blade.

2. If attachment is "rigid", the retaliatory groove in the disk does not change. Stiffness of the connection of the compressor blades with the disk of the impeller is many times higher than the stiffness of the wire rope.

3. If attachment is "soft", stiffness of the connection of the compressor blades with the disk is commensurate with the stiffness of the wire rope. Attachment type 2 and 3 can be modeled in two ways: without the use of contacts (springs, imitating connections), and with contacts.

4. Attachment of the compressor blades in the disk is performed so as to restrict only lateral movement and upward movement (Y-axis). It is "free" attachment. The standard contact can be used to describe such behavior. The contact surface of the disc and blades may drift apart.

The oscillation amplitude A_i and logarithmic damping decrement ζ_i of the oscillation are determined as a result of analysis of the graphs of the oscillation.

In the first step required to determine the dependence of the stiffness coefficient C and the damping coefficient C_v a wire rope when working on the compression from its design parameters and deformation amplitude.

The essence of the experiment is described below. For this study, the standard wire rope (diameter 6 mm) was divided into separate strands. Then the separate strands were re-merged in form of the quasi-continuous ring [18]. Therefore rings have a different radius.

Then the quasi-continuous rings were subjected to cycles of loading and unloading with the help of specially equipped a tensile testing machine. During the experiment, the values of displacement of the plate 1 (Fig. 2) and the corresponding reaction force recorded using a dynamometer in real time.

As can be seen from Fig. 2, equipment includes a fixed base (thickness 10 mm), the movable plate 1. The parts 3, 5 controls a movement of the plate 1. Between the base 1 and the plate member 2 was placed a wire rope. On the outer side of each plate is welded tube 3 having a square cross section and dimensions of 40x40 mm. The thickness of the pipe wall is 3 to 5 mm. The welding is made on the contour 4 at several points (Fig. 2). During compression the steel rod 5 presses directly on the plate 1 due to the through-hole in the pipe 3.

For this work were used the tested rings with the smallest radius. The parameters of the element is the thickness $H = 5.5$ mm, the outer diameter $D_1 = 53.4$ mm, inner diameter $D_2 = 42.4$ mm, material steel $E = 200\,000$ MPa.

It was assumed that an element to the beginning of the calculation already has deformed. This is necessary to compensate for the shift of the compressor blade by centrifugal force (a static load). The biggest hysteretic loop for the calculation of the shift amount was analyzed (Fig. 3).

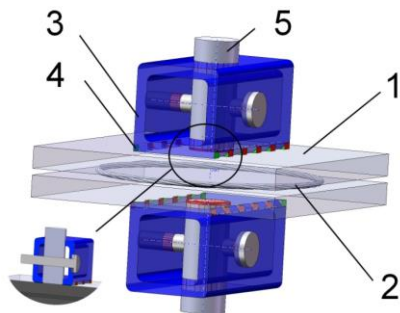


Figure. 2. Equipment for compression testing the wire rope

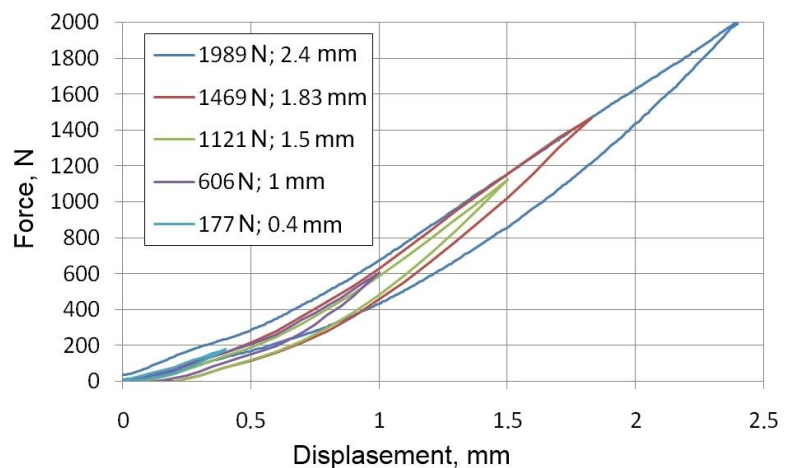


Figure. 3. The hysteretic loops of wire rope at different amplitudes

Several variants of the static characteristics with different maximum load for determination the dynamic rigidity of the elastic-damping element have been received (Fig. 5).

The graphs of the mean-cyclic stiffness C and the energy dissipation coefficient ψ from the deformation amplitude A were approximated by the expressions (Fig. 4, 5):

$$C = -681.6 \cdot A^2 + 1775 \cdot A + 546, \quad (1)$$

$$\psi = -0.34 \cdot \ln A + 0.916. \quad (2)$$

The dimensionless criteria were introduced in this work: the relative amplitude $\varepsilon_A = A/H$; the relative stiffness $\varepsilon_C = C/(l \cdot E)$. Graphs of the characteristics of the elastic-damping element made of wire rope in dimensionless coordinates are shown in Fig. 4 and 5.

Area of the base of a compressor blade $S = 199 \text{ mm}^2$. Thickness of the wire rope $H = 5.5$ mm. Therefore, the segments of the wire rope with a maximum length $L = 36$ mm can be placed under the compressor blade.

A wire rope as MR material is a material with non-linear properties. Therefore, the calculation of oscillations using elastic-damping element made of a wire rope, has several iterations [1].

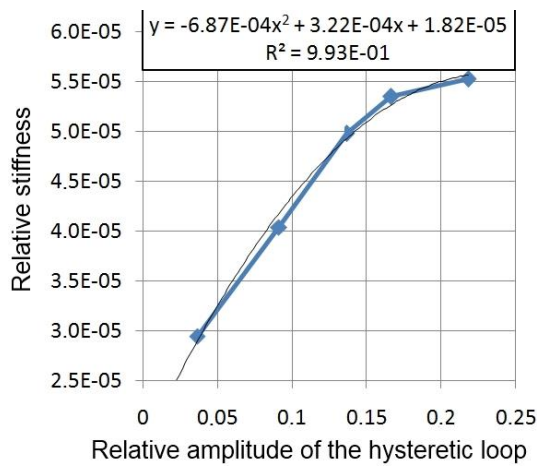


Figure 4. Relative mean-cyclic stiffness

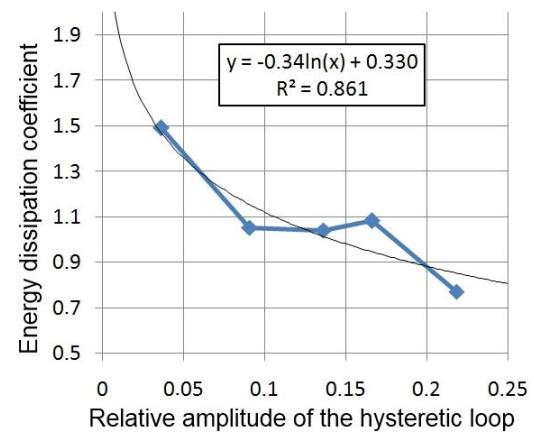


Figure 5. The dependence of the energy dissipation coefficient ψ from the relative amplitude

Parameters of the first iteration point: $A_I = 1$ mm, $\varepsilon_A = 0.182$, $\varepsilon_C = 5.4 \cdot 10^{-5}$, $\psi = 0.917$, $\nu = 86$ Hz, $C_1 = 388000$ N/m, $Cv_1 = 660$ N·sec/m. The final amplitude of the deformation is 1.08 mm. The discrepancy is considered acceptable. The parameters are $C_i = 388000$ N/m, $Cv_i = 660$ N·sec/m. Auxiliary elements simulating the impeller disc for use a contact in model were created.

Modeling "rigid attachment" was in the first calculation. The values of the real variables of the contact was: $FKN = 1$, $FTOLN = 0.1$, $f(MU) = 0.1$, $FKOP = 1$, $TAUMAX = 10^{25}$, $FKT = 1$. Parameters of elastic-damping element ($C = 172000$ N/m, $Cv = 114$ N·sec/m) were left unchanged.

"Soft" contact was modeled in two subsequent calculations. The values of the real variables of contact for the first calculation are $FKN = 0.1$, $FTOLN = 0.1$, $f(MU) = 0.1$, $FKOP = 1$, $TAUMAX = 10^{25}$, $FKT = 1$. The average frequency $\nu = 200$ Hz was used to calculate the value of Cv .

3. Results

The following equations for the wire rope which running to transverse compression, obtained and approximated. The relative mean-cyclic stiffness is ($R^2 = 99.3\%$):

$$\varepsilon_C = -6.873 \cdot 10^{-4} \varepsilon_A^2 + 3.218 \cdot 10^{-4} \varepsilon_A + 1.82 \cdot 10^{-5}, \quad (3)$$

The energy dissipation coefficient is ($R^2 = 86.2\%$):

$$\psi = -0.344 \ln(\varepsilon_A) + 0.331. \quad (4)$$

The maximum calculated deviation of the compressor blade for scheme "capsule" is 17 mm ($\xi = 3.23$). Displacement was measured at node of the bottom of the compressor blade airfoil (1/3 the height of the blade airfoil). A strain sensor was located in this place during the experiment.

It was found that the load-bearing capacity of the wire rope is less than that of the material MR investigated in [1]. The maximum displacement was 2.0 mm (the wire rope, $C = 183$ N/mm, $Cv = 780$ N·sec/m) versus 0.6 mm (MR, $C = 1709$ N/mm, $Cv = 810$ N·sec/m).

The maximum deviation of the compressor blade for "rigid" attachment without contacts is 1.07 mm. The maximum amplitude was 0.44 mm. Logarithmic damping decrement of oscillations was $\xi = 0.03$. The maximum displacement of the blade with the "rigid" attachment and contact amounted to 0.64 mm. The oscillation amplitude amounted to 0.037 mm, $\xi = 0.184$. The maximum displacement elements COMBIN14 were 0.16 mm. In general, the main energy dissipation comes in the place of contact the blade with the impeller disc groove, when using a contact for communication "the disk - the root attachment of blade". The elastic-damping element made of the wire rope in this case is an auxiliary element for case, when amplitude oscillation is large.

Equivalent stress (von Mises stress) has been determined for this model. Stress was calculated only from the vibration action. Stress from centrifugal force was not taken into account. The maximum value was 100 MPa, which is acceptable.

The results of studies of various parameters of the wire rope for "soft" contact were shown in Fig. 6. As a result of the work, the model with a variety of variants of placement of elastic-damping elements made of the wire rope in the root area of the disc of the impeller was created. A wide range of the logarithmic damping decrement of oscillations from 0.104 to 0.235, in the simulation of the most common attachments was obtained [6].

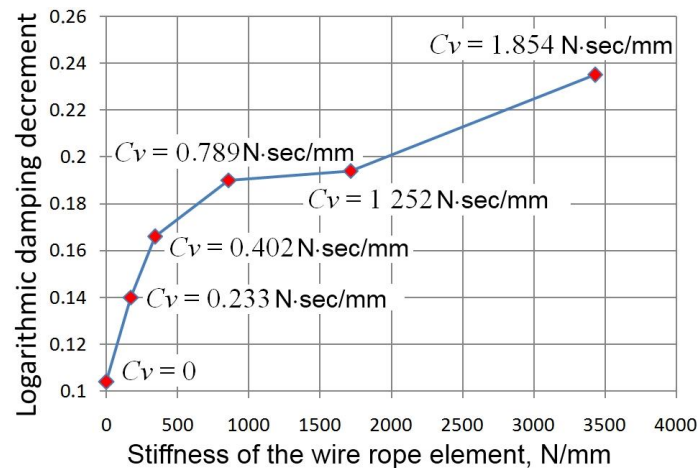


Figure. 6. Graph of logarithmic damping decrement at different stiffness of the wire rope:
 $P_v = 84$ kPa; $FKN = 0.1$, $FTOLN = 0.1$, $f(MU) = 0.1$, $FKOP = 0.5$

Calculations confirmed the effectiveness of the dampers based on the wire rope for damping vibrations of the compressor blades. The greatest efficiency of vibration isolation is achieved with a modified design of the root attachment of blade. In this case a slight slippage of the bottom of the compressor blade relative to the groove in the plane (normal to axis) of the disc rotor is possible.

4. Discussion and Conclusion

In general, the examined wire rope ($H = 5.5$ mm) was too soft for damping the compressor blades by scheme "capsule". It requires research a wire ropes with more the load-bearing capacity. This will require the tensile testing machine with a more wide range of loads.

The proposed method allows to consider a joint operation of the system "the disk - the blade" with the contact interaction and elastic-damping elements made of the wire rope in a single model. The method is based on the finite element method and the Euler-Lagrange method to calculate the contact. This allows the easy selection of the required parameters (by iterations) of the elastic-damping element made of the wire rope, capable of ensuring the protection from the vibration of the blade of a gas turbine engine.

Acknowledgment

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