Simulation of the Vibratory Condition of the Compressor Blade with a Pressed wire Material "MR" Damper Which Located Around the Root Attachment

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Abstract. When you create a modern gas turbine engines urgent task is to improve the reliability by preventing fatigue damages of rotor blades. Such damage is largely determined by the level of vibration stresses. In this paper, using the finite element method and transient analysis of propose a method calculating the damping characteristics of the plates of the pressed wire material "MR" around the root attachment of the compressor blades of a gas turbine engine. Where taken into account contact interaction between the blades and the impeller disk.

1. Introduction
When you create a modern gas turbine engines urgent task is to improve the reliability by preventing fatigue damages of blades. Such damage is largely determined by the level of vibration stresses. The most important factor limiting the vibration is damping capacity of gas turbine engine blades. This characteristic depends on the energy dissipation in a streamlined gas flow, the blades material and structural damping [1-4]. Structural damping it is dissipation of energy in the root attachment and node of contact root platform. In [5], the various schemes of fixing of the compressor blades with elements from the MR material. It can be classified into three groups: A) the impeller disk and blades are in contact only via the plate of the MR material. B) The design of blade and the joggle are provides the active work of plate of the MR material, which is located under the base of the root attachment. C) The plate of the MR material is installed practically free of change in the joggle of the impeller disk under the base of root attachment blade. To calculate of interactions need to create a model that takes into account the joint work blades parts root attachment with small damping and plates of the MR material with a lower hardness, but great damping.

The aim of this work was the construction of the model and the creation of methods of calculation of vibrations blade of compressor with elements of the MR material in the root attachment.

2. Materials and methods
Only one finite element model with three modifications is used in the developed calculation methodology. Each modification depends of fixing conditions. The circuit in Fig. 1 shows the main parameters of a mathematical model that you need to pre-calculate for the successful calculation.

A type of fixing of the compressor blades in the impeller disk has been selected:
1. The root attachment of compressor blades from all sides surrounded by the spacer plates of the MR material. The compressor blade cooperates with the impeller disk through "capsule", which consists of separate plates of the MR material. The plate of the MR material is installed only at the base of the root attachment for the following three types of fixings.
2. The joggle in the impeller disk does not change when the anchorage. Stiffness bracing of the compressor blades of the impeller disk is much higher than the stiffness of the MR material.
3. In soft fixation stiffness bracing of the compressor blades of the impeller disk is comparable with the stiffness of the MR material.
4. "Free" fixation. The joggle design allows the blade to shift lightly to the center of the impeller disk. The plate of the MR material is strongly deformed. The contact surfaces may drift apart.

As a result of analysis of the oscillation curves for each option of fixings are defined amplitude $A_i$ fluctuations and logarithmic damping decrement of oscillations $\xi_i$. At this stage are selected the ultimate option of fixing and characteristics of MR material.

**Figure 1.** Block diagram of the algorithm for calculating the compressor blade vibrations

In [6, 7] are given expression that describes the stiffness $C$ and damping $C_v$ of the thin plates of the MR material. In [7, 8] indicates that the MR material has a nonlinear physical properties. The coefficients of stiffness and energy dissipation of the MR material depend on amplitude of its compression and level of the preliminary static strain. In this paper the static and the dynamic characteristics of MR material are assumed to be equal. This hypothesis is supported by the similarity of the dynamic characteristics which calculated based on the static experiment with data of the dynamic experiment [7, 9, 10]. In [7] given the formula of the transition of the energy dissipation coefficient $\psi$ to the equivalent damping coefficient $C_v$, used in ANSYS.

$$C_v = \psi \cdot C / 2 \pi \nu,$$

where $C$ is the stiffness; $\nu$ is oscillation frequency.

The expressions for calculating the parameters $C$ and $\psi$ can be represented in the form [7]:

$$C = 14 \cdot (1 + 363 \varepsilon A^{1.5} \left(1 + 727 \varepsilon Q^{3.45} \right) \cdot d_w^{1.2} \cdot H^{-0.33} \left(\varepsilon Q / 0.262 \right)^{0.063 H} S \left(\rho / 0.2 \right)^{0.100H/d_w},$$

$$\psi = 1.71 \cdot (1 - 7.23H^{0.13} \left(\varepsilon A - 0.048 \right)) \cdot \left(1 - 0.1 \cdot (10.5 \varepsilon Q - 1)^2 \right) \cdot d_w^{0.138} H^{0.15} \times$$

$$\times \left(0.095/\varepsilon Q \right)^{0.057H} \left(0.2 / \rho \right)^{0.37 + 0.015H / d_w}$$

(3)
The expressions (2) and (3) comprise the dimensional and the dimensionless parameters. The parameter measurement ranges investigated at [7] given in square brackets. The dimensionless parameters: 

\[ \bar{\rho} = \frac{m_{MR}}{S \rho_S} \] - the relative density of the MR material [0.143...0.26];

\[ \varepsilon_A = \frac{A}{H} \] - the relative deformation amplitude [0.043...0.183];

\[ \varepsilon_Q = \frac{Q}{H} \] - the relative preliminary static deformation [0.087...0.262]. The dimensional parameters:

\[ A \] - the deformation amplitude;

\[ d_w \] - the diameter of the wire, [0.09...0.12 mm];

\[ H \] - the plate thickness, [1.15...4.35 mm];

\[ S \] - the plate area,

\[ m_{MR} \] - the plate weight;

\[ \rho_S \] - the steel density, 7800 kg/m\(^3\);

\[ Q \] - the preliminary static deformation.

Due to the nonlinearity of the MR material the calculation of oscillation in this case is of iterative nature [7]. Initially, the required parameters of the MR material are unknown. Therefore, the values of the initial parameters were taken close to the average: 

\[ H = 2.1 \text{ mm}; \]

\[ d_w = 0.09 \text{ mm}; \]

\[ Q = 0.5 \text{ mm}; \]

\[ A = 0; \]

\[ \rho = 0.2; \]

\[ \varepsilon_Q = 0.238. \]

Then, according to equations (2) and (3): 

\[ C = 227 \text{ N/mm}, \]

\[ \psi = 1.316. \]

The frequency of oscillation of the system corresponds to the natural frequency.

The model to calculating the vibrations along and the relatively axis the compressor blade is shown on Fig. 2. The area of the blade base is taken as the calculated area of the plate \( S \) of the MR material.

The fixation of the compressor blades is performed over all axes, except the one about which the oscillations occur.

**Figure. 2.** The modeling by ANSYS the plate of the MR material located at the base of the blade

Firstly, the character of the linear movement the model along the vertical axis has been defined. Secondly, the behavior of the model with plates of the MR material was evaluated in the swing plane.

To simulate the spacers on the lateral surfaces of the root attachment needed to add to the model 16 elements COMBIN14. The pressure \( P_v \) has been applied to the surface of the pressure side of the airfoil the blade.

Further were carried out the studies of "rigid" and "soft" fixing. In this case, the compressor blade along the lateral surfaces is supported on the impeller disk.

At that the elastic-damping element of MR material is mounted only in the lower part of root attachment and practically does not receives a static load. The "rigid" and the "soft" fixing are different by the method of modeling the interaction of the system "blade - disk." At first the "rigid" fixation was investigated. The slot design of the impeller disk and the blade root is almost unchanged. In this case, the joggle in the impeller disc being done deeper for placement of MR material plate.

Initially the modeling was carried out without the use of contact interaction.

Further were taken the following parameters: 

\[ C_e = 16820000 \text{ N/m}, \]

\[ C_{Ve} = 10 \text{ N·sec/m}. \]

The angular velocity is not applied. It was assumed that at the beginning of calculation our model is deformed under a static load by the field of centrifugal forces. The amplitude of the external pressure \( P_v = 84 \text{ kPa}, \) duration 0.005 sec.

By simulate the interaction between the impeller disk and the compressor blade instead of COMBIN14 springs can be used contact. The auxiliary elements, which modeling the impeller disk, to
be able to use a contact in the model were designed. The surfaces on the impeller disk were chosen as targets. The surface of the compressor blade has been chosen as the contact. The contact of type «No separation» was chosen. The option "Close gap" has been set to the initial moment of calculating the surfaces are brought into contact state even when there is a small gap. The following values of real variables contact were set:  

\[ FKN = 1, \quad FTOLN = 0.1, \quad f(MU) = 0.1, \quad FKOP = 1, \quad TAUMAX = 10^{25}, \quad FKT = 1. \]

However, the contact interaction can be softer. Constructively this can be achieved by the following method. For example, this can be done by changing the construction of the root attachment or by the use of inserts of wire-rope elements. The deformation of elastic-damping element is increased at such fixation, characterized by a decreasing value of the contact stiffness \( FKN \) and/or \( FKOP \) and increasing value \( FTOLN \). Therefore, the element more active involved in the work.

The following real constants of contact were used to implement the "free fixation" at the calculation:  

\[ FKN = 0.1, \quad FTOLN = 0.1, \quad f(MU) = 0.1. \]

Further were performed several calculations at the different properties of the MR material. The following parameters of the contact were taken:  

\[ FKN = 0.1, \quad FTOLN = 0.1, \quad f(MU) = 0.1, \quad FKOP = 0.5, \quad FKT = 1. \]

The value \( FKOP \) has been reduced in order to increase the influence on the result of the parameters MR material.

The frequency \( Cv \) at which the calculation was made was 178 Hz. It is the average frequency. The first peak frequency is 100 Hz (Fig. 3). The dissipation of energy on the first peak has a significant proportion. Therefore, the assumption of a certain average frequency of oscillation is acceptable.

### 3. Results

For all subsequent schemes fixing blades, the maximum deviations obtained along the X-axis.

Several calculations have been performed at the simulation of the "capsule" scheme:

1. The initial parameters of the MR material:  
   \[ P_v = 100 \text{ kPa}, \quad \xi_d = 0.847, \quad A = 69 \text{ mm}. \]
   Displacement are measured at the node on the blade airfoil of the compressor blades. This node was located in the lower part of the compressor blade to 1/3 of its height. At this place at the experiment was located the strain sensor. External pressure \( P_v \) acts sinusoidally only at the initial time.

2. The stiffness of the plate of MR material has been increased to increase loading capacity of the structure. Was obtained the value of the parameter \( C = 3115000 \text{ N/m} \), which is close to the maximum possible for the range investigated in [7]. Was obtained:  
   \[ \xi_d = 0.011, \quad A = 0.015 \text{ mm}. \]
   Damping is very small.

3. Parameters have been calculated \( C = 1709000 \text{ N/m}, \quad Cv = 810 \text{ N\cdotsec/m}. \) Parameter \( \xi_d = 1.615 \), the first peak amplitude of 0.6 mm, the amplitude of the third peak of about 0.1 mm.

![Figure. 3. The graph of vibrations under the loading of the sinusoidal pressure at amplitude \( P_v = 84 \text{ kPa} \)](image)
In general, the simulation result was considered satisfactory. However, of this type of fixation the MR material should have a stiffness that on one or two orders higher than the stiffness of the MR material, which investigated in [7]. At the same time should not be a significant reduction in the value of $\psi$. Fig. 3 shows a graph of vibratory displacement at "rigid" fixing without contact. The maximum deviation $A_n$ was about 1 mm, the maximum amplitude $A_{max}$ of about 0.35 mm; parameter $\zeta_n = 0.123$. The maximum stress of the compressor blade was about 95 MPa. This value correlates well with the stresses obtained in the study of the pinned damper, and corresponds to the expected result. The logarithmic damping decrement $\zeta_n = 0.123$ when using plate of the MR material is very close to the logarithmic damping decrement, obtained to the pinned damper by the same compressor blade $\zeta_n = 0.135$ [5,8,11].

For the scheme "rigid" fixation using contacts: the value of displacement was about 0.56 mm, the oscillation amplitude was 0.055 mm (versus 0.35 mm), the energy dissipation coefficient $\zeta_n = 0.188$. It value is greater than the previously obtained value of 0.123. The maximum amount of displacement of the COMBIN14 element was 0.14 mm.

In general, when using the contact at the coupled the compressor blade and the root attachment of the impeller disk, then the main energy dissipation occur into the "root attachment-blade" contact.

The maximum displacement at the "soft contact" was 1.08 mm, the maximum amplitude of 0.22 mm, $\zeta_n = 0.123$. For the case of "free fixation": $A = 5.2$ mm, $\zeta_n = 1.223$. At such a scheme of fixation you need stiffer elastic-damping elements to normal operation of the construction. The results of studies of the "soft fixation" with different plates of the MR material are shown on Fig. 4.

Figure 4. The logarithmic damping decrement at the different parameters of the MR material

It is evident that due to structural damping at the result of friction the compressor blade and the impeller disk is provided the value of the parameter equal to $\zeta_n = 0.104$. The use of spacers of the MR material can increase this parameter by 44% while maintaining of acceptable amplitudes not exceeding 1.71 mm.

A result of research was designed a model that takes into account the vibrations of compressor blade mounted in the joggle of the impeller disk. The different options for placement of the plate of the MR material into the root attachment of the impeller disk were investigated.

A wide range of the logarithmic damping decrement was obtained: from 1.615 to 0.08 [5]. The calculations confirmed the experimental studies of the ability of MR material to damping of the compressor blades of gas turbine engine [1, 5]. The high-performance of these dampers for vibration damping of the compressor blades. The proposed method is based on the finite element method and the method of the Lagrange-Euler to calculate of contact. The proposed method allows considering a joint operation of the "disk-blade" system and the plates of the MR material implementing contact interaction, into the single model. This model is a convenient tool at the iterative selection of the required parameters of MR material.
4. Discussion and Conclusion
By solve the problem of oscillations the compressor blades with restrictions on the plates of the MR material it is possible without the involvement of the finite element method. You can use the markers (by MSC.ADAMS or ANSYS/Rigid Body) for this method.

By analyzing calculations that are presented above, could be made a conclusion. The interaction of "rigid" and "soft" root attachment fixings of compressor blade at the disk of the impeller can be modeled both by the contact, and by the COMBIN14 elements. The difference between these two methods is only in the values of real constants of these elements (C, Cv, FKN, FKOP etc.).

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