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APPLICATION OF VARIATION-ANALYTICAL METHODS FOR ROTATING MACHINE DYNAMICS WITH ABSORBER

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ABSTRACT

A new variation-analytical method for the rotor systems with dynamic vibration absorber (DVA) is proposed. The essence of the method is to introduce few parameter models providing the effective analyses of differential equations for flexible rotor. The presence of DVA may significantly affect the dynamic and improve the characteristics of rotor system.

Key words: dynamic vibration absorber, rotor system, variation-analytical methods

INTRODUCTION

The improved dynamic vibration absorbers (DVA) design with taking into account complex rotating machines dynamic is the basic aim of the paper. It is often impossible to balance the rotating elements to reduce the vibration to an acceptable level. The paper contemplates the provision of DVA or number of such DVA. Originally designed DVA reduces vibration selectively in maximum mode of vibration without introducing vibration in other modes. The final results are achieved at far less expense than would be required to replace the concrete and steel foundation with one massive enough. By installing DVA, one can minimize excitation virtually at the source. In order to be more effective, a vibration absorption system should react in all frequency domains. The present absorber also has as an advantage that it can be constructed to get a wide-range vibration absorption property. This construction allows for the easy connection of above rotor equipment.

PURPOSE OF RESEARCH

Optimal parameters of DVA are obviously demanded to determinate the complete modeling of dynamics of rotating machine. Two degrees of freedom model are totally inadequate to calculate the vibration frequencies of the construction with accuracy and therefore, for a sufficiently accurate determination of its dimensional characteristics and to determine such frequencies as well. It is therefore necessary in practice to dimension the construction through more complex modeling. In particular, concentrated mass and rigidity calculation methods may be adopted on the basis of even more accurate theoretical determination.

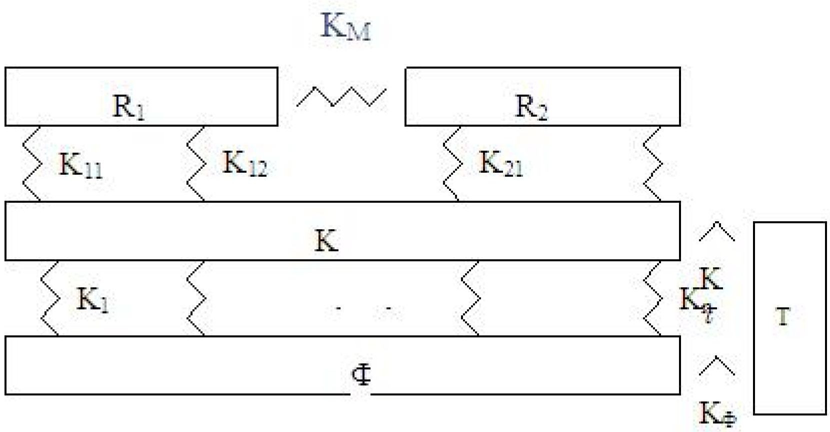
METHODS AND RESULTS OF RESEARCH

The numerical schemes (NS) row is considered for the complex vibro – exitated constructions. The methods of decomposition and the NS synthesis are considered on the basis of the new methods of modal synthesis. Complex NS are led of discretely-continuum type that enables in the adaptive mode to calculate tension not only in the continue elements, but in places of most tension concentration in joints. The absorbers in accordance with this project may be applied not only to electric machines or aeronautic structures, but also to any other type of vibro – exitated structure, such as cars, chisel installation, optical, magneto – optical disks, washing machines, refrigerators, vacuum cleaners, etc. Rotating machinery will typically introduce both acoustic and vibration energy into any fluids or structures surrounding the machinery. Both random and deterministic processes related to the operation of the machinery can cause the acoustic and vibration energy. Random processes result in noise or vibration spreading over a wide band of frequencies. Deterministic processes, on the other hand, often generate energy that is confined to a family of distinct frequencies radiated as "pure" tones.

Large rotating elements, particularly such as exhaust fan rotors used in electric power generating plants or in gas compression, are unbalanced in operation due to their exposure to varying factors. It is often impossible to balance the rotating elements to reduce the vibration to an acceptable level. The two degrees of freedom model described in [1-3] are totally inadequate to calculate the natural frequencies of vibration of the rotor with proper accuracy and therefore, for a sufficiently accurate determination of its dimensional characteristics and to determine such frequencies as well. It is therefore necessary in practice to dimension the rotor through more complex modeling [4-6]. The detailed description of DVA application may be found in [7]. The lot of examples of DVA application for suppression of building vibrations is presented in [7] and [8]. In particular, concentrated mass and rigidity calculation methods may be adopted on the basis of even more accurate theoretical determination. The numerical schemes (NS) row is considered for such complex vibro – exitated constructions. The methods of decomposition and the NS synthesis are considered on the basis of new methods of modal synthesis [9-10]. Complex NS are provided of discretely-continual type that enables in the adaptive mode to calculate tension not only in the stratified elements, but in places of their higher concentration – in joints. It also considers the numerical schemes for research of halving elements that also have been got on the basis of kinematics hypotheses. The research of local tensions on verge of stratified structure at the different kinds of its fixing is conducted on the basis of simple and more complex NS. Traditional design methodology, based on discontinuous models of structures and machines, is not effective for high frequency vibration. Present research develops a modern prediction and control methodology, based on complex continuum theory and application of special frequency characteristics of structures. Complex continuum theory allows to take into consideration system anisotropy, supporting structure strain effect on equipment motions and to determine some new effects not described by ordinary mechanics of the continuum theory.

In Fig. 1 the typical discretely-continual scheme rotating machine is presented, namely - the compressor (though this scheme is general for many rotating machines), with 5 continual elements – rotors R_1, R_2, the case K, the base Φ, pipes T, and discrete ones: bearings K_{ij}, a clutch K_M, elastic joints of pipes to the case and the base of the compressor and elastic joints of case to the base (though there maybe several points of a compression, they are schematically represented in fig. 1 by elements: K_1, K K_η^K and K_Φ).

Fig. 1. Scheme of rotating machine



Let one to take that vibration spatial kinematics loading on bearings of rotors (it will be defined on the basis of the complex calculation scheme from Fig. 1) is given. The following reduced NS should be concerned: U_{ij}^{0} – the dynamic displacement affixed on the part of the case as vectorial magnitudes (rolling contact bearings of such type are considered, that it is possible to disdain longitudinal gains and the moments)

$$U_{ij}^{0} = (U_{ij}^{0x}, U_{ij}^{0y})$$

Let one to consider the dynamics of the rotating machine shaft taking into account flexibility of the shaft both on curving and on torsion. One shall proceed from a variation principle of Hamilton. The kinematics hypotheses for deviations of an axis from a certain statically counterbalanced position of the shaft on bearings should be accepted. One supposes that the center of masses of the given cross-section of the shaft coincides with its center of weight in the immovable frame rigidly connected to statically equilibrium position of the shaft on elastic support. Then the kinetic energy will consist of two items

$$T = T_w + T_{rot} \tag{1}$$

T_w – bending component, T_{rot} – rotation one. The first item is equal to

$$2T_w = \int_0^L \rho\, V_C^2 dz = \int_0^L \rho\, (V_r^2 + V_\varphi^2) dz \tag{2}$$

($\rho(z)$ - a linear mass of a shaft, V_C^2 – quadrate of tangential velocity of shaft equal the sum of quadrates of radial V_r and tangential $V_{\varphi r}$ velocities of the center of the given cross-section of the shaft, multiplied on its mass of relative movement will be equal to

$$2T_{rot} = \int_0^L \rho R^2 \gamma_\varphi^2 dz \tag{3}$$

Here $R(x)$ - radius of the given part of the shaft, γ_φ - axial angular velocity. The gyroscopic composites [1-3] are neglected.
Let one to accept kinematics hypotheses for deformation of the shaft. It may be assumed, that in an immovable frame (z, r, φ), the shaft realizes elastic rotating oscillations with a changeable angle of twisting

$$\varphi = q_\varphi(t)\lambda_\varphi(z) \tag{4}$$

and radial oscillations in a mobile frame (z, r, φ), rigidly connected to shaft in orthogonal planes [1]

$$W_1 = \sum_i q_i^1(t)\lambda_i(z) \qquad\qquad W_2 = \sum_i q_i^2(t)\lambda_i(z) \tag{5}$$

Here $q_i^j(t)$ – are time dependent unknown functions and multipliers by its $\lambda_i(z)$ – known coordinate functions.
Let one to note kinetic and potential energy. For the last resistance of the shaft may be assumed as linearly elastic. According to (1) - (5) one may obtain [7-10]

$$(M^c \ddot{q}^c + \overline{K}^c \cdot q^c) \cdot \delta q^c + \sum (M_i q_i^n) \delta q_i^n = 0 \tag{6}$$

Here M^c – the mass matrix of continua elements, M_i – the discrete elements masses, δq_i^n – independent time functions variations. By equating the terms by these variations one obtains the system of ordinary differential equations (detail description may be found in [9,10]).

As an illustrative example the one mass rotor model is applied [1-3]. For these purples the linearization of system (6) was provided. The DVA was attached to the case. The system of governing equations takes a form:

$$
\begin{aligned}
&M_O \frac{d^2 x_1}{dt^2} = -k_{1A}(x_1 - x_{1A}) - c_{1A}\left(\frac{dx_1}{dt} - \frac{dx_{1A}}{dt}\right) - F_1 \\
&M_O \frac{d^2 x_2}{dt^2} = -k_{2A}(x_2 - x_{2A}) - c_{2A}\left(\frac{dx_2}{dt} - \frac{dx_{2A}}{dt}\right) - F_2 \\
&M_A \frac{d^2 x_{1A}}{dt^2} = k_{1A}(x_1 - x_{1A}) + c_{1A}\left(\frac{dx_1}{dt} - \frac{dx_{1A}}{dt}\right)_1 \\
&M_A \frac{d^2 x_{2A}}{dt^2} = k_{2A}(x_2 - x_{2A}) + c_{2A}\left(\frac{dx_2}{dt} - \frac{dx_{2A}}{dt}\right)
\end{aligned}
\qquad (7)
$$

This model and the result of DVA application in frequency domain is presented in Fig. 2. The non-dimensional parameter relations are Ma/Mo=0.115

Fig. 2a. One mass rotor model with DVA

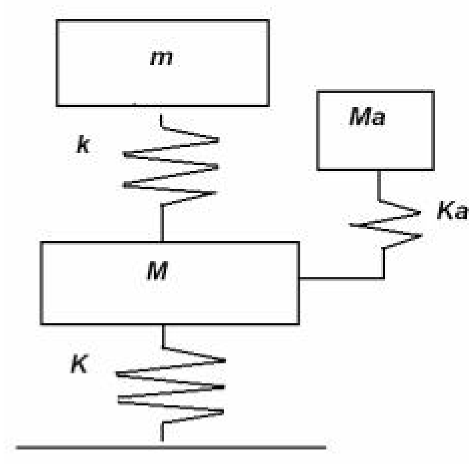
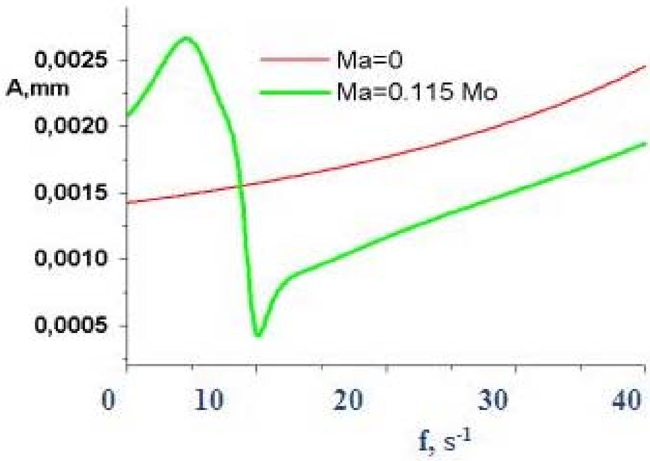


Fig. 2b. FRF characteristics



Here **Ma** is the mass of absorber and **Mo –** mass of the case. The frequency response lines **Ma = 0** and **Ma = 0.115Mo** presents the case of vibration: levels **A** without and with DVA with optimal mass.

Forces obtained on the bases of the given calculation schemes may be used further for determine deformed conditions of cases to define vibrations levels for elements of the rotating machine. Algorithms adduced in [9,10], and known programs – such as ANSYS, NASTRAN, COSMOS may be used by calculation of bearings and shaft stresses at the loading given above.

CONCLUSION

In order that optimal parameters of DVA are determined the complete modeling of dynamics of rotating machine should be made. Traditional design methodology, based on discontinuous models of structures and machines, are not effective for high frequency vibration. They do not give a possibility to determine strains and to predict durability. Present research develops a modern prediction methodology, based on complex discreet-continuum theory. This allows to take into consideration system anisotropy, supporting structure strain effect on equipment motions and to determine some new effects that are not described by ordinary mechanics, namely concentration of strain in junctions and chaotic oscillations. The result may be significantly improved by changing the special form of the DVA.

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