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Е.Л. Селезньов, Д.Е. Селезньов, Б.С. Колтунік

Луцький національний технічний університет

В роботі висвітлено результати експериментальних досліджень змодельованого у програмному пакеті MATLAB навчального електромеханічного стенду.

Ключові слова: *балансування роторів, роторний вузол, вібрація, дисбаланс, електромеханічний стенд, гнучкий ротор.*

E.L. Seleznov, D.E. Seleznov, B.S. Koltunik

RESEARCH OF THE ROTOR ASSEMBLY IMBALANCE SYSTEM OF THE TRAINING ELECTROMECHANICAL STAND

The paper highlights the results of experimental research on a training electromechanical stand modeled in the MATLAB software package.

Keywords: *rotor balancing, rotor assembly, vibration, imbalance, electromechanical stand, flexible rotor.*

Introduction. In any production, the continuity of the technological process is ensured by the trouble-free operation of equipment, the condition of which must be constantly monitored. The efficiency of equipment operation is most often determined by the quality of technical and repair maintenance, for which various balancing machines are used. The study and application of such devices in industry is a promising direction in the development of the theory of automatic balancing and is an urgent scientific and technical task.

The emergence of modern vibration measuring equipment provides the possibility of dynamic balancing of rotors at the place of operation and reducing the vibration load on the supports to permissible limits.

The causes of imbalance can be: failure to comply with exact dimensions when manufacturing shafts; inaccurate centering of connected parts relative to each other; non-uniform density of the part and non-uniformity of the material; presence of gaps in the joints of assemblies and parts; deformation of shafts due to damage during operation and damage during thermal and mechanical treatment.

Relevance of the work. At the moment, there are many types of balancing stands[1, 2], which, depending on the natural frequency of the rotor in the machine supports, are divided into: pre-resonant , resonant, post-resonant . Since the studied rigid rotor, starting from a certain rotation speed, can exhibit the qualities of a flexible rotor, the choice of one or another balancing method is determined by both the rotation speed and the rotor configuration. Existing balancing stands do not take into account the possibility of studying different configurations of rotor assemblies, which can be represented not only by different types of rotors, but also by different types of connecting couplings.

Presentation of the main material. The educational stand, designed within the framework of the master's thesis, the scheme of which is shown in Figure 1, consists of two main parts: control system; rotor unit imbalance system.

The advantages of this scheme are the possibility of studying both rigid and flexible rotors, and even rotors of different configurations. An example of a rigid rotor is a short-circuited rotor of an asynchronous electric motor, and a flexible rotor is a rotor of a powerful current generator. Also, for example, a turbine rotor consists of the following main parts: disks or drums, blades, shaft, thrust comb and connecting coupling, an unloading piston is installed on the rotor of jet turbines, i.e. the design may be different, so it is impractical to study a rotor of one configuration.

In addition, the design provides for interchangeable couplings that allow you to explore different types of connections. The longitudinal-compensating type of couplings provides for compensation of axial displacements of the rotors. This type of couplings includes elastomeric couplings, which are designed to transmit torque to units powered by the engine, for example, a generator, a hydraulic booster pump or an air conditioner compressor and to dampen vibrations between two connected shafts , etc. Elastic couplings are especially recommended for use where there is a violation of the shaft alignment or axial movement. The universal-compensating type of couplings allows you to compensate simultaneously for two or all three types of displacements of the connected shafts. This type includes bellows couplings, which are used where it is necessary to provide protection against twisting of the connected parts, as well as where accurate backlash-free transmission of angle and torque is required. An example of the use of such couplings can be precision speed and positioning systems for industrial robots and high-precision machine tools.

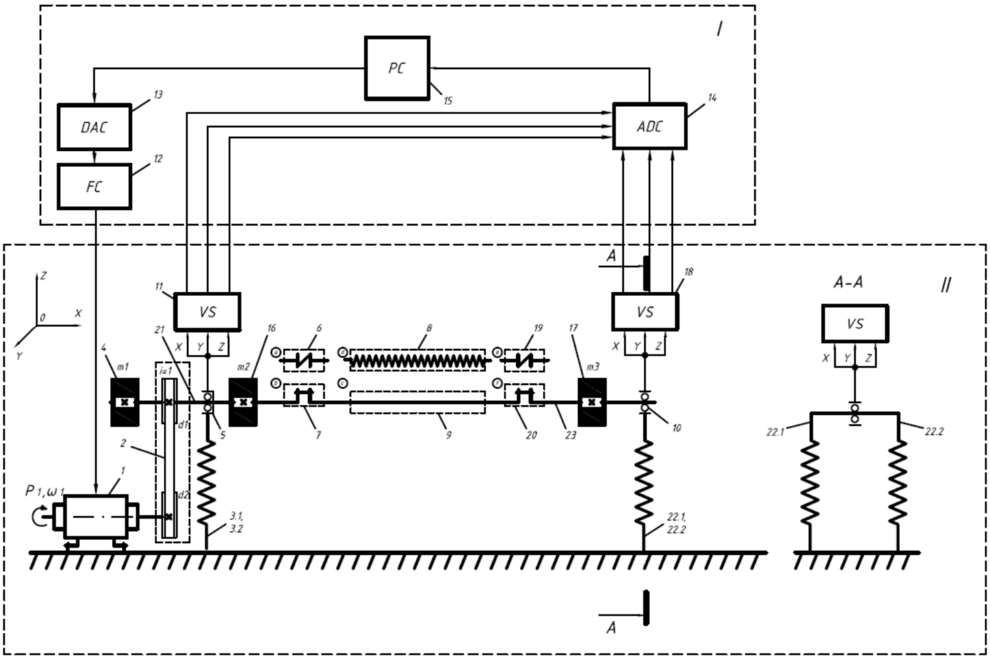
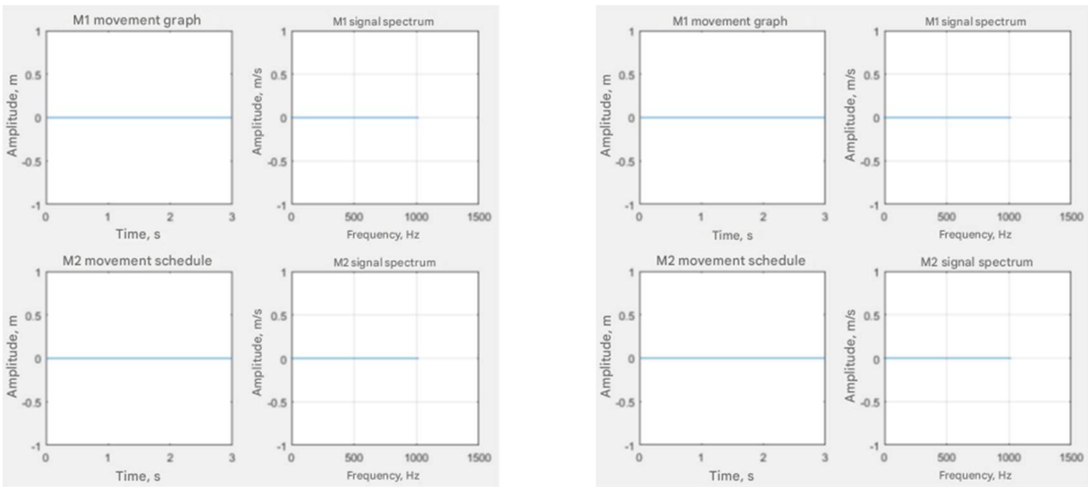


Figure 1. **Schematic diagram of the rotor unit unbalance system: I – control system, II – rotor unit unbalance system; 1 – tested electric motor on legs; 2 – toothed belt transmission; 3.1, 3.2, 22.1, 22.2 – support racks in the form of a flat direct spring; 4,16,17 – flywheel 1,2,3; 5 – rolling bearing; 6,19 – elastic connection (cross-compensating) of two coaxial shafts 21,23; 7,20 – elastic connection (universal-compensating) of two coaxial shafts 21,23; 8 – flexible shaft; 9, 21, 23 – rigid shaft; 10 – rolling bearing; 11,18 – vibration sensor 1,2; 12 - frequency converter (FC); 13 - digital-to-analog converter (DAC); 14 - analog-to-digital converter (ADC); 15 - personal computer (PC).**

Having compiled a mathematical model of the rotor unit balancing system, a study of the influence of system parameters on vibration characteristics was conducted. All input data for calculations, as well as the mathematical model of the system, will be covered in detail in the master's thesis. Mathematical modeling and research were performed in the MATLAB[3, 4] software package.

With a perfectly balanced rotor assembly, that is, when the axis of rotation coincides with the main axis of inertia of the shaft, no oscillations occur in the system, as proven by the experiments shown in Figures 2 and 3.



a b

Figure 2. **Dependence of displacement on time and amplitudes of speeds on the frequency of rotation (spectrum) on the supports of the rotor assembly with a balanced: a) rigid shaft; b) flexible shaft.**

A study was conducted on the influence of shaft stiffness when the center of gravity is displaced from the shaft axis *e* (determined based on the shaft tolerance field g 6 for a flexible shaft of 0.004 mm, for a rigid shaft of 0.0065 mm). The studies were conducted without installing imbalances , with minimal support stiffness, engine rotation speed, which is within the operating speed range of both the rigid and flexible shafts (2,000 rpm), and with the shaft connected by bellows couplings.

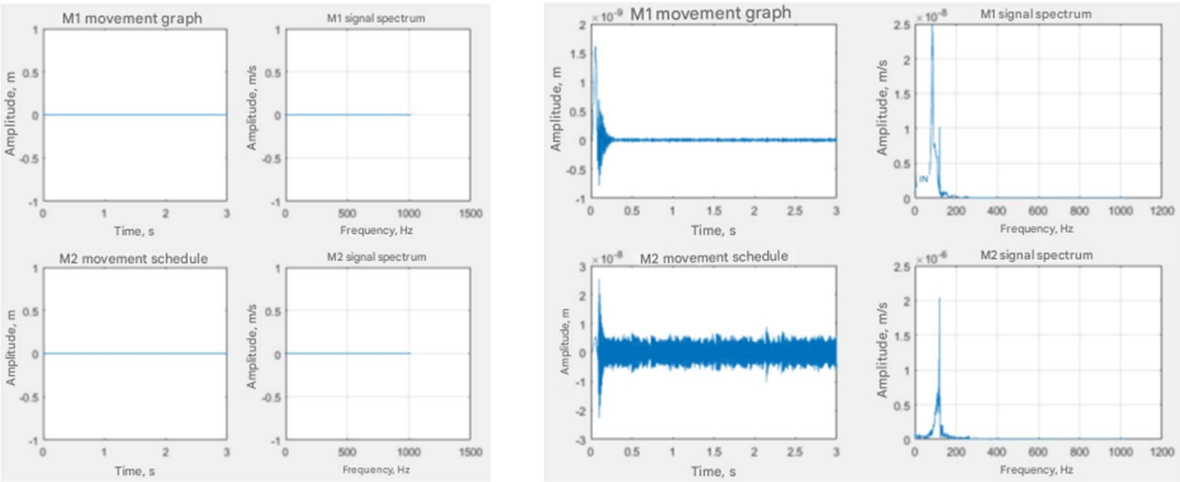


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Figure 3. **Dependence of displacement on time and amplitudes of speeds on the frequency of rotation (spectrum) on the supports of the rotor assembly when installing: a) a rigid shaft; b) a flexible shaft.**

Conclusions. Comparing the studies with rigid and flexible shafts, it was concluded that if an arbitrary force is applied to a rigid shaft, this force is completely balanced by the elastic reaction of the shaft. However, if the shaft is not rigid enough (flexible shaft), the forces acting on it can cause deflection. If the rotation speed is low, then small bending vibrations quickly die out. Under the conditions, the rectilinear shape of the shaft is stable. As a result of the above, we choose a flexible shaft for further research in our work.

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