642. Impact of External excitations on the Dynamic properties of Negative-stiffness Vibration Isolation Table

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Abstract. This paper presents a study of the efficiency of a negative-stiffness vibration isolation table Minus K 500BM-1 in the frequency range of 2–110 Hz. An overview of other possible methods of vibration isolation is provided. Experimental testing of vibration transmissibility was conducted. Special vibration excitation equipment was tested, which enables the planned studies of the object in question to be carried out.

Keywords: negative stiffness, vibration isolation, low frequency, vibration isolator.

Introduction

Vibration isolation is a procedure whereby the undesired impact of external vibration is reduced. This procedure normally involves the introduction of resilient dampers between the vibrating mass and the source of vibration, as a result of which, in the presence of specific conditions for excitation of the vibration, the dynamic reaction of the system is decreased [1]. Vibration isolation systems can be classified based on their characteristics. One of the classifications relies on the type of management systems—passive or active—used [2]. Only passive vibration isolation systems were used until approximately 1990.

Active vibration isolation systems. No active systems would be needed if it was possible to obtain a passive system with a very low natural frequency approaching zero. However, such low natural frequency systems require a stiffness, which is so low that it would be impossible for the system to maintain it. For this reason, we require active systems for vibration isolation in the vertical direction. It is easier to obtain low-frequency vibration isolation in the horizontal direction even with low horizontal stiffness [2].

Since most passive systems are effective and adequate for the isolation of high-frequency vibration, the need for active systems for the isolation of high-frequency vibration is smaller compared to their need in the case of isolation of low-frequency vibrations [2]. Furthermore, active vibration isolation systems require efficient vibration sensors, which is why they are also approximately ten times more expensive compared to passive vibration isolation systems [3].

Whenever the frequency of vibration impacting a machine or structure equals natural frequency of the system, the resonance phenomenon occurs, and this is associated with large amplitudes and failures.

Passive vibration isolation systems. The most common systems, which best protect against disturbances, are based on the principle of passive control of vibration/noise. These systems reduce vibrations and noise by simply dissipating the energy as heat [4]. These systems comprise a spring (an elastic part) and an energy damper. Elastomers, liquids, or negative stiffness parts can also be used. Springs resist the movement of vibration by straining the opposite forces which are in proportion to their displacement. Dampers comprise a piston moving through a viscous liquid or a conductor moving in the magnetic field, which dissipates kinetic energy in the form of heat. It should be mentioned though that springs have their natural resonance frequency, which depends on the force constant k, and, if the frequency of vibrations approaches the natural frequency of the system, the spring becomes an amplifier. Such simple systems do not demonstrate adequate operation in the presence of vibrations lower than 10 Hz [5]. Therefore, their damping quality is fairly poor. However, such systems are inexpensive and simple, therefore widely used [4]. Passive systems have preconfigured properties, which cannot be adjusted as long as the system remains in operation [2].

Passive systems are efficient only in the case of disturbances with a frequency which is considerably higher than their natural frequency. In practical situations, though, vibrations may have frequencies varying in time or can form a frequency spectrum. In this case, the efficiency of vibration isolation systems decreases when the frequency of vibrations approaches the natural frequency of the system. In the latter case, the system reduces the impact of vibration from the high-frequency spectrum but increases the impact of vibration from the spectrum close to its natural frequency. When determining the damping properties of a system, when the isolation of vibrations with different frequencies must be combined and the overall efficiency of vibration isolation must be ensured, the end results may be far from satisfactory [2].

Object and methods of study

The study involved the negative-stiffness vibration isolation table Minus K 500BM-1, which ensures good isolation of low-frequency and amplitude vibration in floors and buildings [6, 7]. This vibration isolator is completely mechanical and vibration isolation is achieved by using a spring and a negative stiffness mechanism.

Masaki Hosoda established that a condition of key importance is that the frequency of periodic force almost or fully equals the natural frequency f_0 of the vibration isolator [8]. Transmissibility in the range next to f_0 exceeds 1. It means the increased base movement vibration transmitted to the measurement equipment. If transmissibility is smaller than 1, the measurement equipment on the vibration isolator is isolated from the vibrations coming from the base. The expression of natural frequency f_0 is presented according to the following formula:

$$f_o = \frac{1}{2\pi}\sqrt{\frac{k}{M}} \qquad (1)$$

According to this formula, natural frequency may be very low if the structure of the equipment has very low stiffness and is very heavy. This is not the best solution, however. Instead of modifying the structure of the equipment, natural frequency can be reduced by using a vibration isolation system, which must filter the floor (base) vibrations before they reach the surface of the table.

Isolation is firstly ensured by retaining the required ratio between the disturbance frequency and the natural frequency of the system. The damped natural frequency of the system is expressed as follows:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M} - \left(1 - \left(\frac{C}{C_C}\right)^2\right)}; \qquad (2)$$

where: C – damping coefficient [Kg-sec/m]. C_C – critical damping [Kg-sec/m]. C/C_C – dimensionless relative damping parameter.

Damping is useful when the vibration isolation system operates close to its natural frequency. In this manner, the transmissibility is reduced. Increasing the damping coefficient C/C_C reduces the natural frequency of the vibration isolation system.

Transmissibility is expressed using the following formula:

$$T = \frac{|x|}{|y|}. \tag{3}$$

Transmissibility can also be expressed in decibels (dB):

$$T = 20\log\frac{|y|}{|x|}dB. \tag{4}$$

The studies of transmissibility of vibrations and other dynamic parameters were conducted using the negative-stiffness vibration isolation system Minus K 500BM-1 with minimum 164-kg distributed load, which was mounted on a slab with a honeycomb structure.

The system in question was impacted by means of a vibrator, whose output parameters were obtained by amplifying the signal received by an amplifier.

Measurement of the vibrations was carried out on the surface of the negative-stiffness vibration isolation table. Seismic accelerometers 8344 were attached to the four angles of the table. One seismic accelerometer 8344 was attached to a vibration isolation table with air supports. The negative-stiffness vibration isolation table was mounted on a stiff optical table with a slab provided with a honeycomb structure. A scheme of the experimental board is provided in Figure 1.

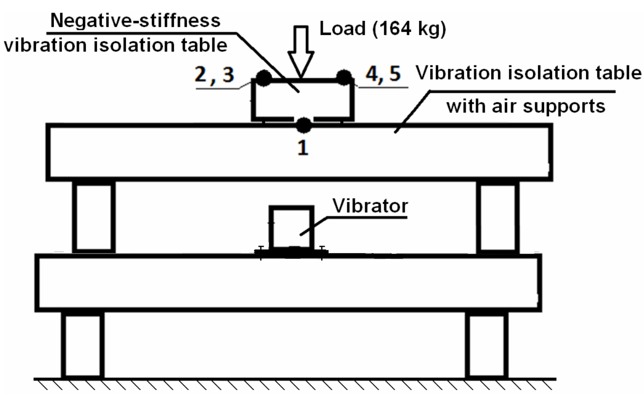

Fig. 1. The scheme for the measurement of transmissibility of vibrations by using vibrators. Points 1, 2, 3, 4, and 5 indicate places where seismic accelerometers 8344 are mounted

Results of experimental studies

Vibration measurement and analysis equipment produced by the Danish company Brüel & Kjær was used for the testing. The measurement signals were processed by means of a computer using the Origin 6 software. The signal spectra and statistical parameters were determined.

Fig. 6 provides a graph illustrating transmission degree dependence on frequency with respect to the first point.

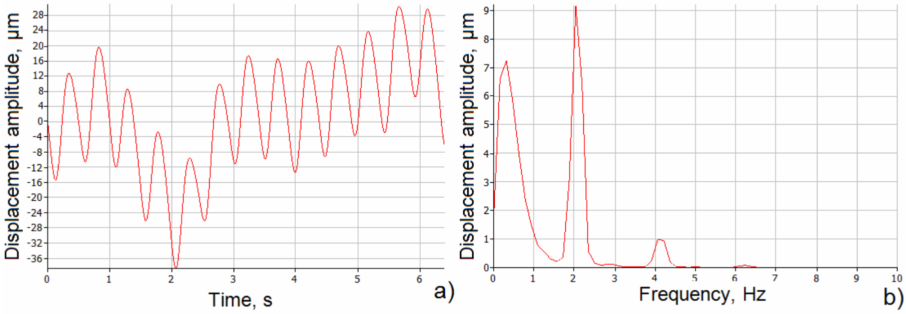

Fig. 2. Graphs of absolute vibrations (a) of point 1 (see Fig. 1) and vibration spectrum (b) (the frequency of excitation is 2 Hz)

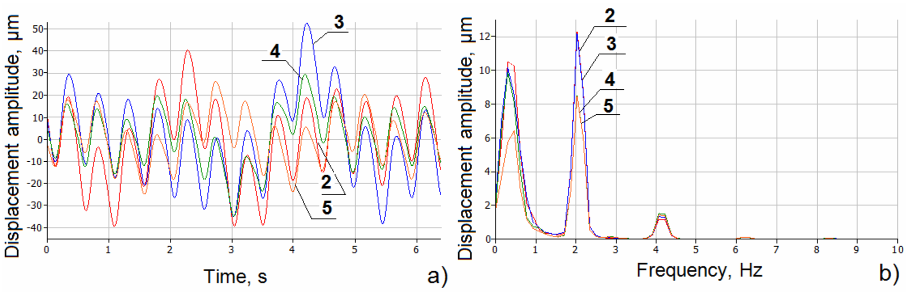

Fig. 3. Graphs of absolute vibrations (a) of points 2, 3, 4 and 5 (see Fig. 1) and vibration spectrum (b) (the frequency of excitation is 2 Hz)

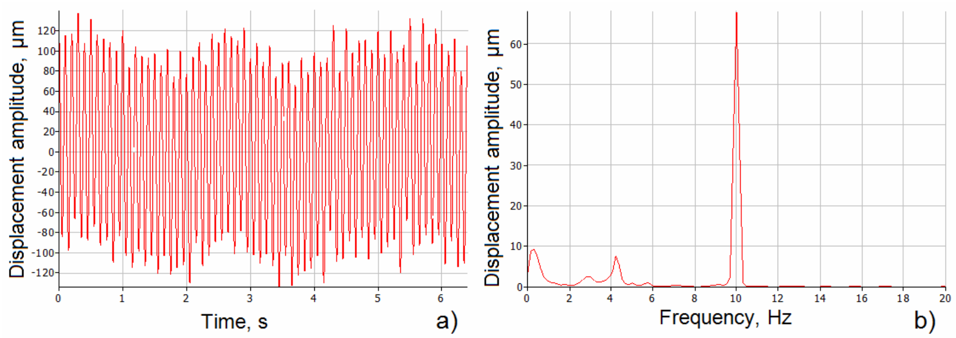

Fig. 4. Graphs of absolute vibrations (a) of point 1 (see Fig. 1) and vibration spectrum (b) (the frequency of excitation is 10 Hz)

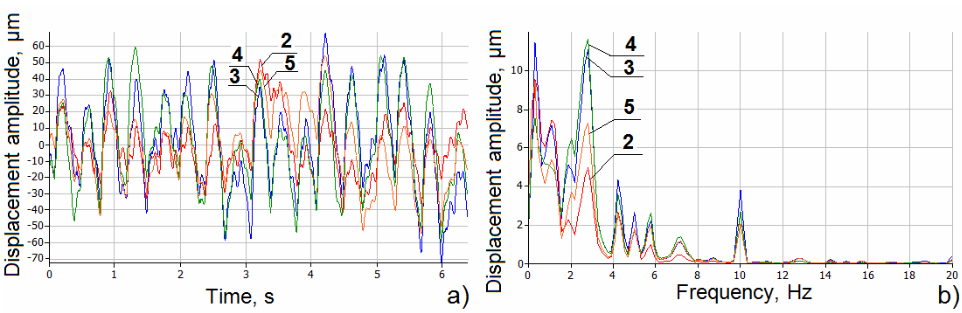

Fig. 5. Graphs of absolute vibrations (a) of points 2, 3, 4 and 5 (see Fig. 1) and vibration spectrum (b) (the frequency of excitation is 10 Hz)

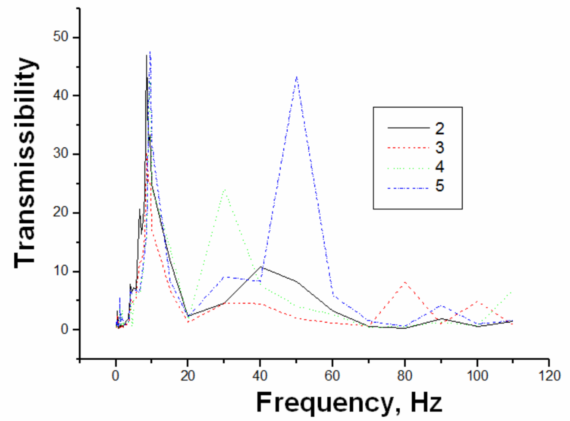

Fig. 6. Transmission degree dependence on frequency with respect to the first point (Fig. 1)

Results and discussion. Figs. 2-3 present the displacement amplitudes and spectra of the 1st point (located on the surface of the optical table) and of the 2nd, 3rd, 4th, and 5th points (located on the surface of the vibration isolation table). The obtained testing results demonstrate that the spectral amplitudes of the optical table surface vibrations are 9 μm at a frequency of 2 Hz, while the spectral amplitudes of the vibration isolation table surface are from 8.4 to 12.2 μm at a frequency of 2 Hz. Results provided in Figs 4-5 illustrate the displacement amplitudes and spectra of the 1st point (located on the surface of the optical table) and of the 2nd, 3rd, 4th, and 5th points respectively. These results reveal that the spectral amplitudes of the optical table surface are 68 μm at a frequency of 10 Hz, while the spectral amplitudes of the vibration isolation table surface are from 2 to 3.8 μm at a frequency of 10 Hz. The electrical amplitudes of vibrations of the carriage at frequencies from 1 to 3 Hz are from 0.01 to 0.03 μm when air is provided to the vibration isolation supports only, whereas when air is directed both to the vibration isolation supports and to the air bearings of the carriage, the spectral amplitude of the vibrations at frequencies from 21 to 23 Hz are from 0.01 to 0.020 μm. The amplitudes in other frequency ranges are insignificant.

The results indicate that the best characteristics of the negative-stiffness vibro-isolation bench are obtained in frequency range close 10 Hz (dumping ca. 47 times all four points), 30 Hz (dumping ca. 23 times in the fourth point) and 50 Hz (dumping ca. 43 times in the fifth point).

Conclusions

1. The paper presented the developed methodology and equipment for testing of vibration isolation tables. Reported research work was performed on a negative-stiffness vibro-isolating table "Minus K 500BM-1".
2. It was established that the best characteristics of the negative-stiffness vibro-isoliation bench are obtained in the vicinity of 10 Hz.

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