567. Research of Dynamics of a Vibration Isolation Platform

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Abstract. Performance of negative-stiffness vibroisolating table „Minus K500BM-1“ was analyzed in a frequency range of 2-110 Hz. The paper also reviewed other possible ways of vibroisolation. Performed experimental vibrations transmissibility tests were reported as well. A special vibration excitation equipment was tested in research facilities. Subsequently, experimental test methods were described together with presentation of results and conclusions of experimental analysis.

Keywords: negative-stiffness, payload, “HVAC” systems, actuators, elastomers.

Introduction

Efficient vibration damping systems are needed in order to be able to effectively control the accuracy of the measurement results, particularly in the case of sensitive laboratory measurement equipment.

According to Enrico L. Colla (2003), a payload (term used in this work to refer to the object to be isolated from vibrations) can generally be subject to four sources of perturbations arising: from ground vibrations; from ambience, i.e. acoustic noise or air flow; from mechanically coupled external equipments and from internal mechanisms. Measuring equipment from extraneous vibrations can be isolated using active or passive vibroisolation systems.

An active vibration isolation systems creates equal, but opposite direction forces to directly cancel unwanted vibration from the system by using actuators, sensors and controllers.

With passive methods of isolation, by using a mechanical connection, the vibration energy is dissipated or redirected before reaching the isolated object. Passive systems may employ elastomers, springs, fluid or negative stiffness elements. One of the main passive vibration isolator is a spring that is placed between the transmitting shock or vibration surface and an insulating object. The spring resists to impulse and absorbs a lot of energy through deformation. Fluid or elastic elements are attached to the spring to perform the function of inhibition. A representative example is a shock-absorber in a car.

In this case, the mechanical energy from shock or vibration operates in the fluid and is converted into heat energy, thereby reducing the transmission energy exerted onto vehicle body.

Passive vibroisolation systems usually cost less than the active ones and their relative simplicity makes them more reliable and safe. Passive systems are classified into the negative stiffness systems that can isolate the effects of vibration along all six degrees of freedom when the resonant frequency is 0,5 Hz or even less. These vibroisolation systems are better 50-100 times in comparison to most air tables in 5-10 Hz range. Considering the nature of vibrations, several vibroisolating systems may be used which act as one.

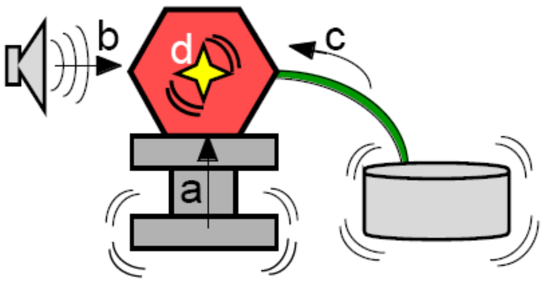

Fig. 1. Most relevant vibration sources for a payload (hexagonal object). a) ground vibrations, b) acoustic noise, c) external and d) internal vibration sources

The object of research

For current research a negative-stiffness vibroisolating table "Minus K 500BM-1 is used, which isolates low frequency and amplitude vibrations generated in buildings and floors. This vibroisolator is fully mechanical and isolation is achieved by using springs and a negative stiffness mechanism.

Fig. 2. Negative-stiffness vibroisolating table „Minus K500BM-1“

According to Masaki Hosoda (2009), a particularly important condition occurs when the frequency of the periodic force nearly or exactly corresponds to the natural frequency f_0 of the device. In the region around f_0, the transmissibility is over 1, which means the amplified vibration of a ground motion transmits to the equipment. If the transmissibility is lower than 1, the equipment is isolated from the ground motion. The natural frequency f_0 of the equipment is given by the equation:

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

According to this equation, the natural frequency may be small, if one designs the equipment of low stiffness and mass. But it is not a reasonable way.

Instead of changing the design of the equipment, the natural frequency can also be reduced by the means of vibroisolating system, which has to filter floor (base) vibrations.

Isolation is attained primarily by maintaining the proper relationship between the disturbing frequency and the system natural frequency. The damped natural frequency of the system is described by the equation:

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

Here C is a damping coefficient with a unit of [Kgsec/m]; C_C is a critical damping with a unit of [Kg-sec/m]; C/C_C is a damping factor.

Damping is advantageous when the mounted equipment is operating at or near its natural frequency because it helps to reduce transmissibility.

Increasing the damping factor C/C_C reduces natural frequency of the isolation system. Transmissibility is given by the equation:

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

Transmissibility can also be expressed in decibel (dB):

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

Methodology and experimental research

Vibration transmissibility and other dynamic parameters were investigated for the negative-stiffness vibroisolating system „Minus K 500BM-1“. A 164 kg minimum load and honeycomb composite plate were employed.

The experiments enabled determination of:

- Vibration transmission characteristics by applying impulse excitation;
- Vertical vibration transmission characteristics acting in the excitation frequency 2-110 Hz;
- Damping efficiency for low frequencies.

Tests were performed with Danish „Bruel & Kjaer“ dynamic measurement and analysis equipment: movable measuring equipment „Machine Diagnostics Toolbox Type 9727” with computer DELL; Vibrometer 2511; Seismic accelerometers 8306 and 8316; Amplifier 2706; Generator 1027; Oscillator 4810; Shock hammer 8202;

Measurement signals were processed in PC with „Origin 6“ and „Pulse“ software packages.

The measurement results were analyzed and signal spectra as well as statistical parameters (arithmetic mean, the standard deviation and the standard deviation of the mean) were calculated using the following formulae:

$$\bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i,\ S_X = \sqrt{\frac{1}{n-1}\sum_{i=1}^{n}\left(x_i - \bar{x}\right)^2},\ S_{\bar{X}} = \frac{S_X}{\sqrt{n}} = \sqrt{\frac{1}{n(n-1)}\sum_{i=1}^{n}\left(x_i - \bar{x}\right)^2}, \tag{5}$$

here n is the number of measurement results, x_i – measurement result.

Vibration measurements (fig. 5) were performed on the negative-stiffness vibroisolating table surface (9) attached orientation log (7) with seismic accelerometer 8306 (6). Other seismic accelerometer 8318 (12) was mounted on the rigid optical table (4).

Negative-stiffness vibroisolating table was mounted on a rigid optical table.

Orientation log allows fixing or reversing an accelerometer in the right direction.

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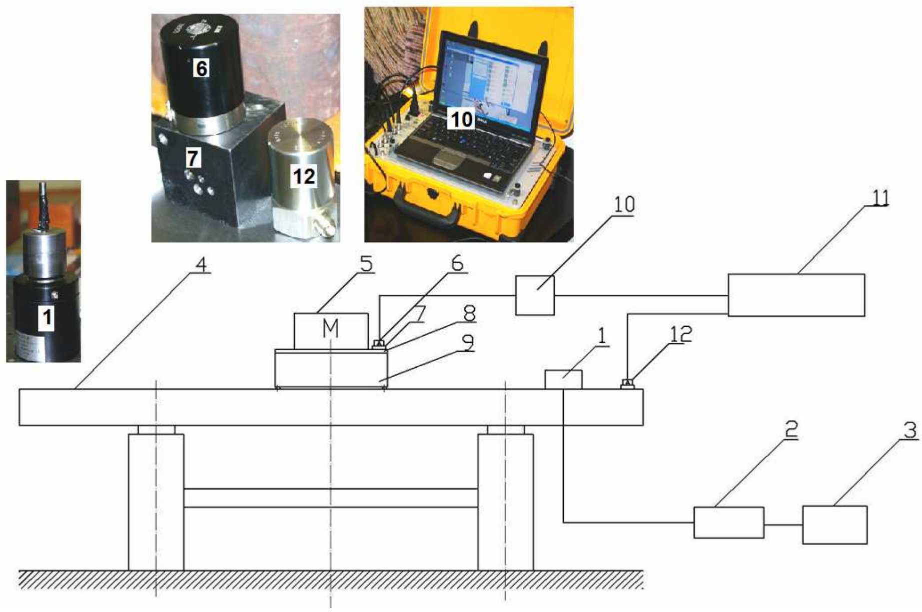

Fig. 5. Oscillation caused by vibrator. 1 – Oscillator 4810; 2 – Amplifier 2706; 3 – Generator 1027; 4 – Optical table; 5 – Mass – 164 kg; 6 – Seismic accelerometer 8306; 7 – Orientation log; 8 – Fix board; 9 – Negative-stiffness vibroisolating table „Minus K 500BM-1“; 10 – Vibrometer 2511; 11 – Movable measuring equipment „Machine Diagnostics Toolbox Type 9727” with computer DELL; 12 – Seismic accelerometer 8318

During research with oscillator exerted harmonic vibrations in 2-110 Hz range in vertical direction.

Research Results

Obtained experimental results were processed statistically and are listed in Table 1. The experimental results are presented in Figs. 6 - 13.

Table 1. Statistical parameters of the vibration acceleration measurement in harmonic oscillation using the excitation signal

Accelerometer	Excitation frequency, Hz	Standard deviation S_x, m/s^2	The standard deviation of average $S_{\bar{x}}$, m/s^2	Minimum value x_{min}, m/s^2	Maximum value x_{max}, m/s^2
8306	2	3,98E-4	3,11E-6	-0,00151	0,0016
8318	2	0,00654	5,11E-5	-0,04248	0,0313
8306	5	0,00149	1,17E-5	-0,0028	0,00313
8318	5	0,0268	2,10E-4	-0,1395	0,150
8306	20	0,00338	2,64E-5	-0,012	0,0098
8318	20	0,372	0,0029	-1,184	1,066
8306	50	0,014	1,09E-4	-0,0372	0,0429
8318	50	1,615	0,0126	-4,09	3,816
8306	80	0,00453	3,54E-5	-0,0094	0,0113
8318	80	0,380	0,00296	-1,002	0,799

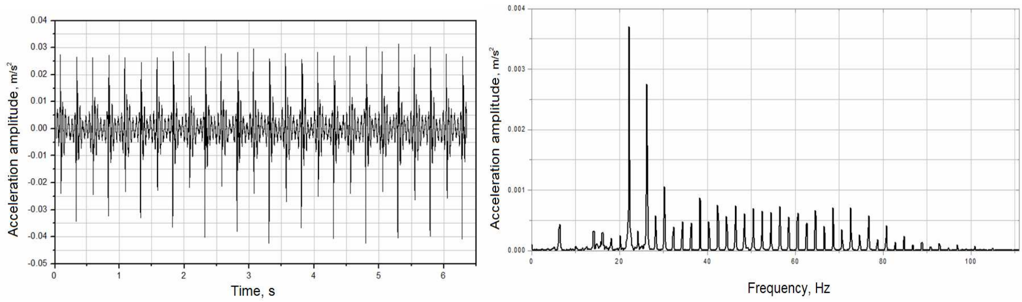

Fig. 6. Temporal signal and spectrum. Vertical excitation vibration with 2 Hz frequency on the optical table

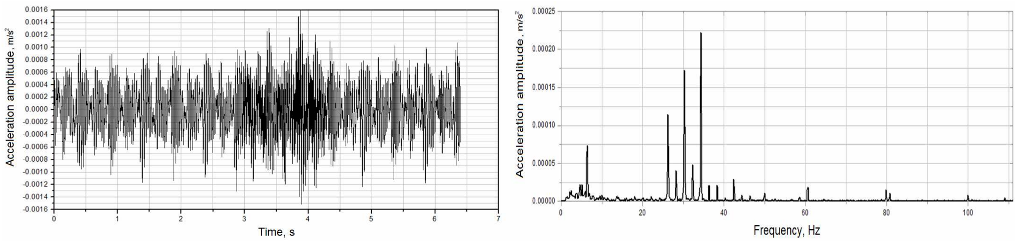

Fig. 7. Temporal signal and spectrum. Vertical inhibition vibration with 2 Hz frequency on the „Minus K 500BM-1" table

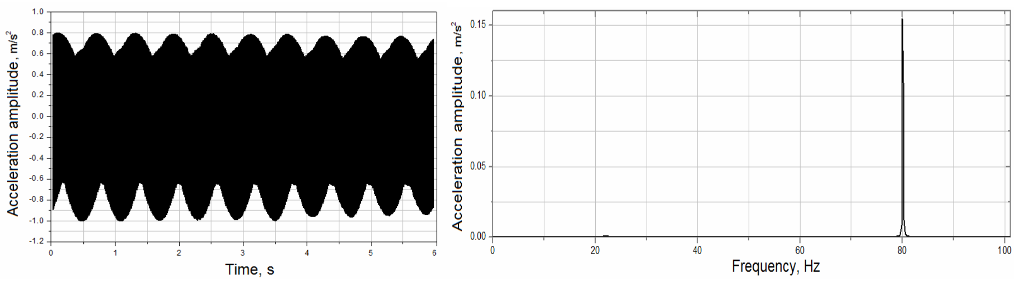

Fig. 8. Temporal signal and spectrum. Vertical excitation vibration with 80 Hz frequency on the optical table

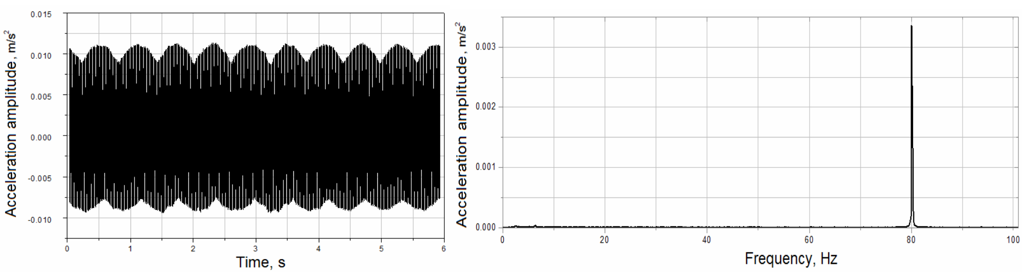

Fig. 9. Temporal signal and spectrum. 80 Hz inhibition vertical vibration on the „Minus K 500BM-1" table

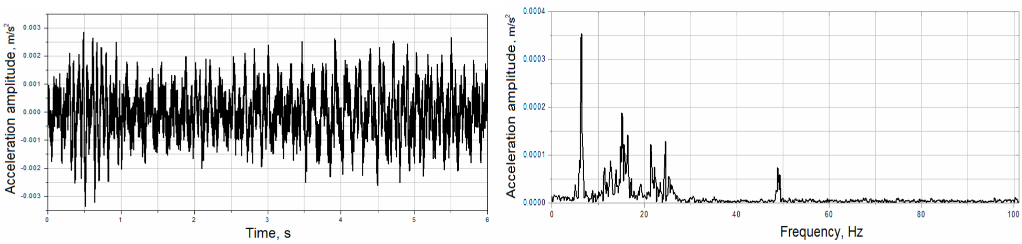

Fig. 10. Ground temporal vibration signal and spectrum on the optical table. Seismic accelerometer 8318

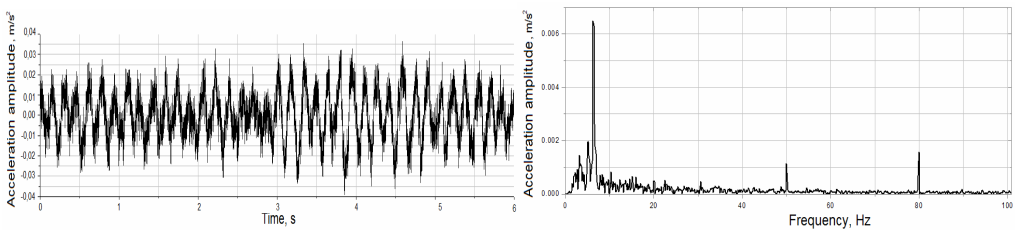

Fig. 11. Ground temporal vibration signal and spectrum on the „Minus K 500BM-1" table. Seismic accelerometer 8306

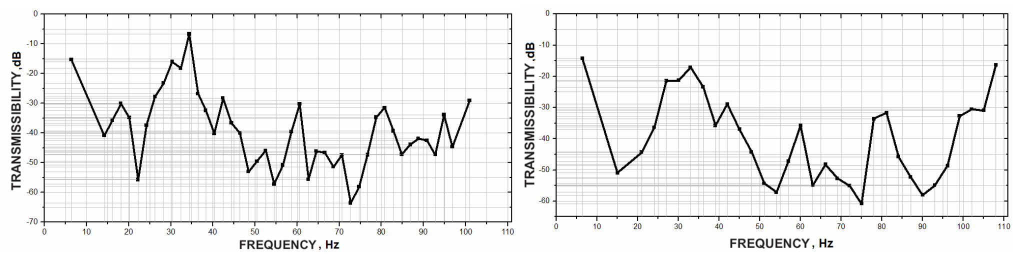

Fig. 12. Transmissibility curve of the „Minus K 500BM-1" table. Excitation vibration with 2 Hz (a) and 3 Hz (b) frequencies

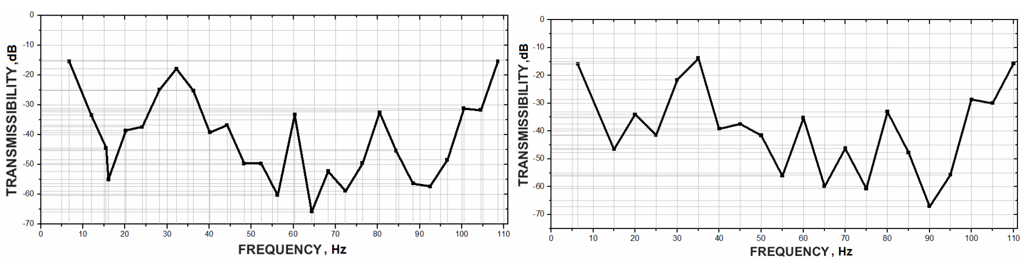

Fig. 13. Transmissibility curve of the „Minus K 500BM-1" table. Excitation vibration with 4 Hz (a) and 5 Hz (b) frequencies

Test results indicate that the best damping is obtained at a frequency range of 50-90 Hz. The worst damping is observed till 10 Hz and in a range of 30-35 Hz.

Conclusions

1. A methodology and equipment were developed for vibration table testing. An experimental research was conducted on a negative-stiffness vibroisolation table "Minus K 500BM-1". The paper presented the obtained excitation and inhibition signals and their spectrums.

2. Vibration transmissibility characteristics were determined for the negative-stiffness vibroisolating table in the frequency range of 2-110 Hz of vertical vibrations.

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