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# Elevator Vibrations and Ways to Reduce Them

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**The article deals with the measurement of the vibrations in passenger elevators. The introduction of an article briefly discusses machine vibrations and their impact on machine design and the surrounding environment. The basic equations from which the equations of motion are derived are listed here. The importance of analyzing machine vibrations in their design, or rather proposing solutions to reduce vibrations during machine reconstruction, is emphasized. Specifically, attention is paid to vibrations generated during an elevator operation in the elevator shaft. This is an elevator for transporting people in a newly constructed 5-story building. Vibration values generated by an elevator operation were measured in order to assess the suitability of simple anti-vibration modifications. Vibration measurements were taken on an existing elevator without modifications, and after the initial measurements, modifications were made to attach the guides to the bracket and attach the bracket to the elevator shaft wall. After the adjustment, the vibration measurement was performed again and both measurements were compared with each other.**

**Keywords:** Vibration, Motion Equations, Machine Dynamics

## 1 Introduction

Machines are subject to ever-increasing demands. In addition to high performance and low energy consumption, low noise and vibration levels are also required. Vibration is a common phenomenon during the operation of machines and technical equipment in general. High vibration intensity can significantly reduce their service life and operability, and can even cause accidents or, in more serious cases, complete destruction not only of the machine itself, but also of building structures (e.g., turbine accidents). Vibrations are also accompanied by increased noise in the vicinity of vibrating machines. Vibrations are of great importance in means of transport, where humans are part of the system and vibrations are transmitted to them. These vibrations affect not only the comfort of travel, but also the safety of the vehicle and can even lead to accidents, where, especially in the case of vertical vibrations, the wheels can lose contact with the track (both in road and rail vehicles) [1].

The development of new machines is usually associated with increased performance and, not infrequently, larger dimensions. The requirement to save material and reduce the weight of new machines during construction leads to an increase in their dynamic flexibility. This results in a greater number of natural vibration modes and frequencies, as well as an increase in the occurrence of resonance states, which can lead to damage or even destruction.

All these phenomena lead to the need to introduce new calculation methods into the design or pre-production stages of machine development, which will enable not only the analysis of vibrations, but also the design of modifications to suppress them. In practice, it is important not only to effectively control the dynamic properties of existing machines (diagnostics), but also to implement measures to reduce them when designing new machines or reconstructing existing ones. Therefore, machines are dynamically balanced, multi-cylinder machines are built, and other design modifications are made to reduce the disruptive forces and moments arising during their operation.

Flexible mounting of machines also plays an important role in reducing the impact of vibrations. This ensures that the natural frequencies in all directions, especially in the directions of the main axes of inertia, are sufficiently distant from the frequencies of the disturbing forces that cause these vibrations.

The design of new machines, especially the analysis of their vibrations, is significantly influenced by the development of new computational methods, particularly FEM. However, this does not diminish the importance of analytical methods, which allow for more accurate identification of vibration parameters [2, 3].

When designing machines, it is particularly important to correctly design a computational model (both mathematical and physical) of a real mechanical system (machine). Creating a mathematical model is an important prerequisite for successfully solving a mechanical system.

Oscillating mechanical systems can be classified according to a number of criteria. According to [4, 5], from the point of view of mathematical modeling, we can classify them according to:

- Degrees of freedom,
- Type of differential equations,
- Origin of oscillation,
- Nature of motion.

## 2 Methodology

Mechanical vibration is usually characterized by displacement (deflection), velocity, and acceleration. The source of vibration with a single frequency component can be, for example, rotor imbalance in rotating machines, or vibration with multiple frequency components in the case of unevenness in the path along which we are moving (e.g., vehicles on a road).

In practice, forced vibration of a flexibly mounted and bound body occurs most frequently. When deriving the equations of motion for such a body, we start from Lagrange's equations of the second kind [6 - 8].

$$\frac{d}{dt}\left(\frac{\partial E_k}{\partial \dot{q}_j}\right)-\frac{\partial E_k}{\partial q_j}+\frac{\partial E_p}{\partial q_j}+\frac{\partial D}{\partial \dot{q}_j}=Q_j \qquad \text{for } j = 1, 2, \ldots, 6 \tag{1}$$

Where:
$E_k$…Kinetic energy,
$E_p$…Potential energy,
$D$…Dissipative energy,
$q_j$…Generalized coordinate,
$Q_j$…Exciting force.

Kinetic energy is determined from the equation (Fig. 1).

$$\begin{aligned}E_k= \frac{1}{2}\Big\{&m\left(\dot{u}_0^2+\dot{v}_0^2+\dot{w}_0^2\right)+J_x\dot{\varphi}_x^2+J_y\dot{\varphi}_y^2+J_z\dot{\varphi}_z^2-2\left(\dot{\varphi}_x\dot{\varphi}_y D_{xy}+\dot{\varphi}_x\dot{\varphi}_z D_{xz}+\dot{\varphi}_y\dot{\varphi}_z D_{yz}\right)+\\&+2\left[S_x\left(\dot{w}_0\dot{\varphi}_y-\dot{v}_0\dot{\varphi}_z\right)-S_y\left(\dot{u}_0\dot{\varphi}_z-\dot{w}_0\dot{\varphi}_x\right)-S_z\left(\dot{v}_0\dot{\varphi}_x-\dot{u}_0\dot{\varphi}_y\right)\right]\end{aligned} \tag{2}$$

Where the moments of inertia of body $J$ about the $x$, $y$ and $z$-axes are:

$$J_x=\int_m (z^2+y^2)\,dm \qquad J_y=\int_m (z^2+x^2)\,dm \qquad J_z=\int_m (y^2+x^2)\,dm \tag{3}$$

The deviation momentsDxz, $D_{xy}$, $D_{yz}$ are:

$$D_{xz}=\int_m xz\,dm \qquad D_{xy}=\int_m xy\,dm \qquad D_{yz}=\int_m yz\,dm \tag{4}$$

And the static moments $S$ relative to the individual axes ($x$, $y$, $z$) are:

$$S_x=\int_m x\,dm \qquad S_y=\int_m y\,dm \qquad S_z=\int_m z\,dm \tag{5}$$

Potential energy is determined by the formula:

$$E_p=\frac{1}{2}\left[\sum_{i=1}^{n_x}k_{xi}u_i^2+\sum_{i=1}^{n_y}k_{yi}v_i^2+\sum_{i=1}^{n_z}k_{zi}w_i^2+\sum_{i=1}^{n_x}\varkappa_{xi}\left(v_i^2+w_i^2\right)+\sum_{i=1}^{n_y}\varkappa_{yi}\left(u_i^2+w_i^2\right)+\sum_{i=1}^{n_z}\varkappa_{zi}\left(u_i^2+v_i^2\right)\right] \tag{6}$$

After substitution and adjustments, the equation of motion takes the form:

$$\mathrm{M}\ddot{\mathrm{q}}_j+\mathrm{B}\dot{\mathrm{q}}_j+\mathrm{K}\mathrm{q}_j=\mathrm{F}_j(\mathrm{t}) \tag{7}$$

Where:
M…Mass (inertia) matrix,
B…Damping matrix,
K…Stiffness matrix,
$q_j$…Generalized coordinates,
$F_j$…Vector of excitation functions.

The solution to this equation is well known, e.g., from [3, 8, 9].

The oscillation of a mechanical system is a source of vibration and noise in the vicinity of this vibrating system. To assess the suitability of a simple modification to reduce vibration transmission during the operation of a passenger elevator in a 5-story building, vibration and noise measurements were taken before and after the modification. The measurements were taken on new equipment before the building was put into operation, thus eliminating the influence of wear and tear due to operation. However, there was a risk that the equipment had not been properly run in.

An elevator for 6 people is located in a concrete shaft. The guides were attached to the wall using chemical anchors. The elevator machine is located in the elevator shaft above the elevator structure. The cage is of classic construction with a steel frame and sandwich panel infill.

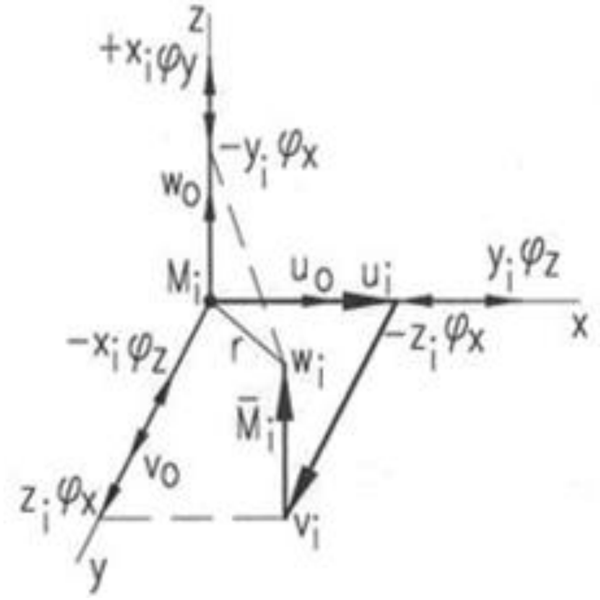


***Fig. 1*** *Geometry of displacement and rotation of a body Mi*

The subject of the measurement was the transmission of vibrations from elevator operation to the walls of the elevator shaft. The measurements were performed:

- On an unmodified elevator,
- On a modified elevator, i.e., a 1 cm thick rubber block was inserted between the guide and the bracket and between the mounting foot and the wall - see Fig. 2c.

The measurement was performed on the wall of the elevator shaft, which is crucial for the transmission of vibrations into the building structure [10] (Fig. 2a). The location of the acceleration sensor for measurement on the console is shown in Fig. 2b.

The measurements were performed using two Brüel & Kjær (hereinafter BK) type 4331 triaxial piezoelectric accelerometers with a BK 2647B converter. A BK 4291 mechanical acceleration sensor calibrator was used.

The measurement and evaluation of the effective values of vibration acceleration was performed in accordance with ČSN EN 2631-2. The maximum acceleration values were measured in octave bands of 63, 125, 250, and 500 Hz, i.e., in the frequency range significant for the transmission of structural noise into the building structure. The frequency components of vibrations were determined in third-octave bands from 63 to 10,000 Hz in three main directions, *x*, *y*, *z* (see Fig. 2a,b). The results were processed using BZ 5503 software. All instruments used are in accuracy class 1. Expanded measurement uncertainty $\varepsilon = 3$ (technical measurements).

The acceleration sensors were located between the third and fourth floors in the elevator shaft.

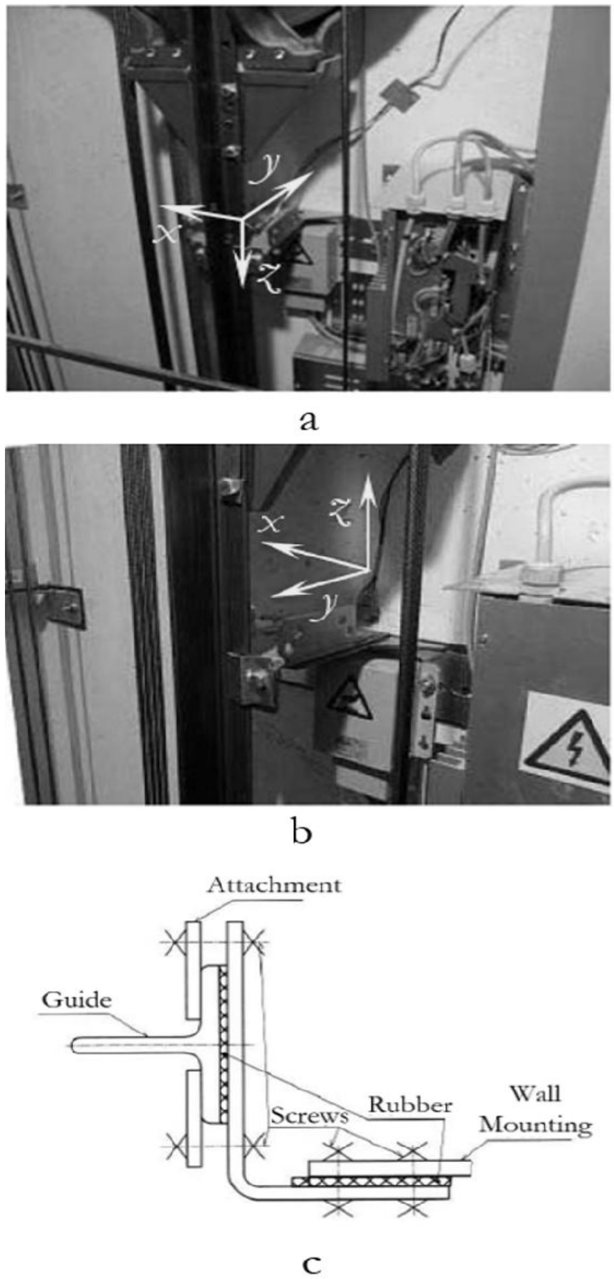


***Fig. 2*** *Acceleration measurement in an elevator shaft (a – mounting the acceleration sensor on a bracket, b – mounting the sensor on the wall, c – diagram of guide mounting modifications)*

## 3 Solutions and discussion

As already mentioned, the measurement was performed before and after adjusting the attachment of the guide to the bracket and the bracket to the wall. In all cases, the measurement time ranged from 115 to 250 seconds. Each measurement consisted of 16 cycles, including opening and closing the doors at the destination station. The results were averaged.

### 3.1 Measurement results before adjustment

The acceleration curves on the elevator shaft wall in the *x*, *y*, and *z*-axes are shown on Figs. 3 to Fig. 8.

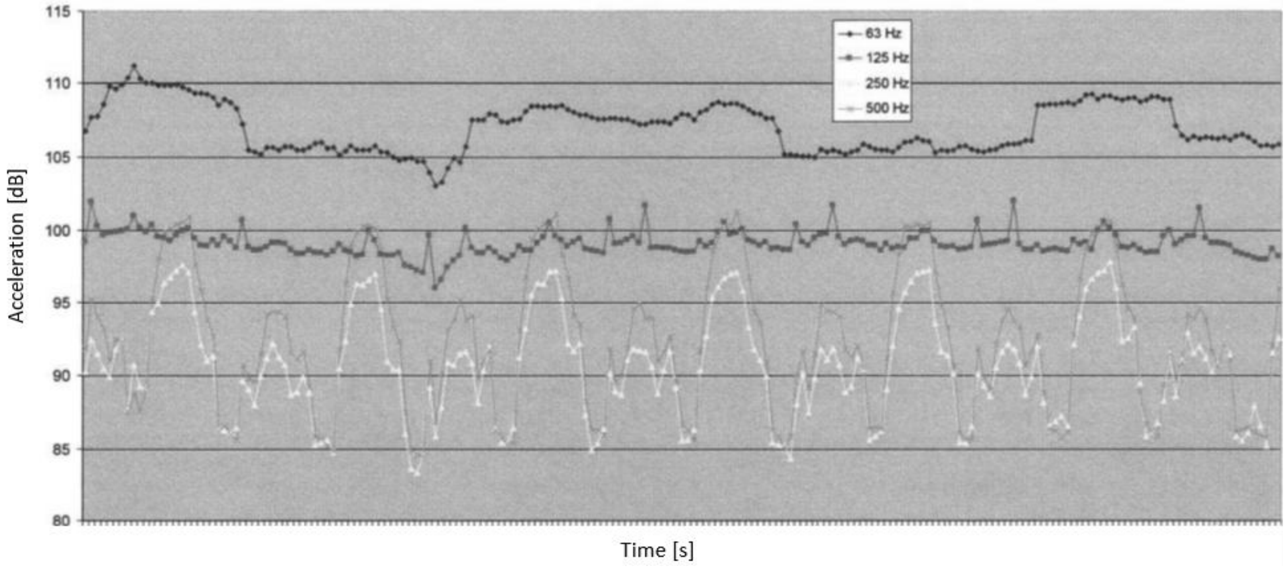


***Fig. 3*** *Acceleration curve on the wall in the x-axis direction when traveling between the 3rd and 4th floors*

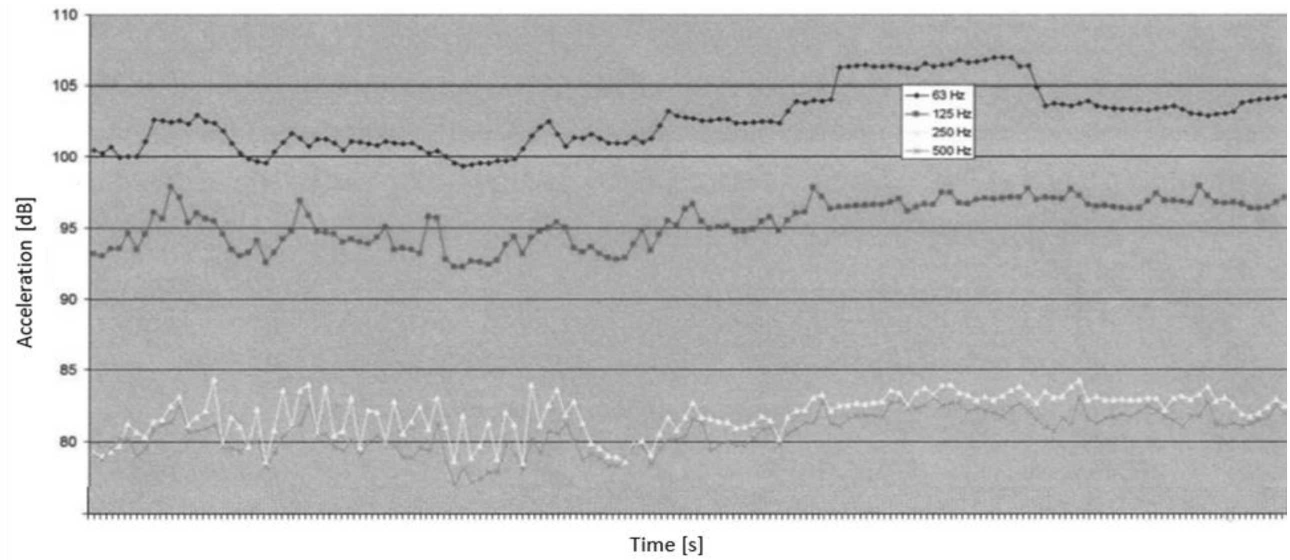


***Fig. 4*** *Acceleration curve on the wall in the x-axis direction during repeated closing and opening of the door on the 4th floor*

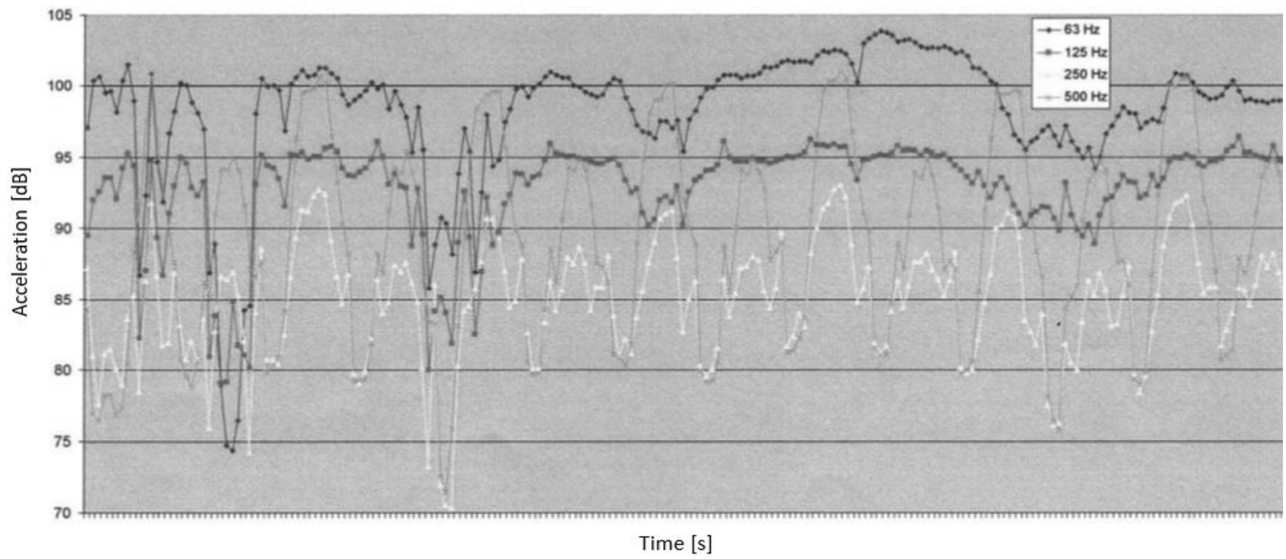


***Fig. 5*** *Acceleration curve on the wall in the y-axis direction when traveling between the 3rd and 4th floors*

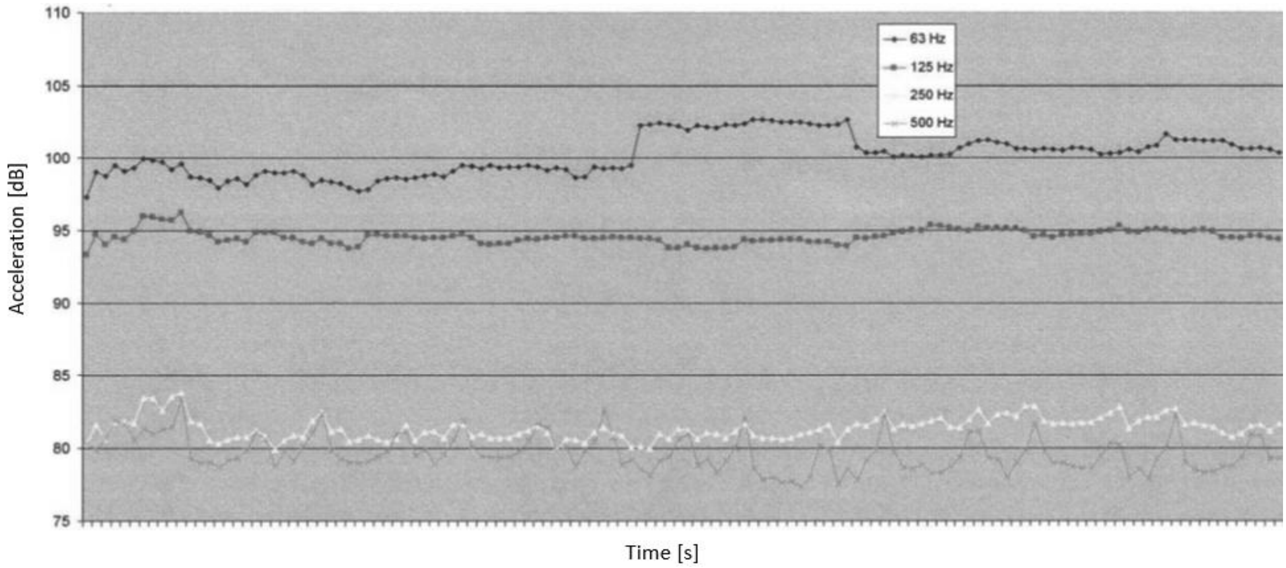


***Fig. 6*** *Acceleration curve on the wall in the y-axis direction during repeated closing and opening of the door on the 4th floor*

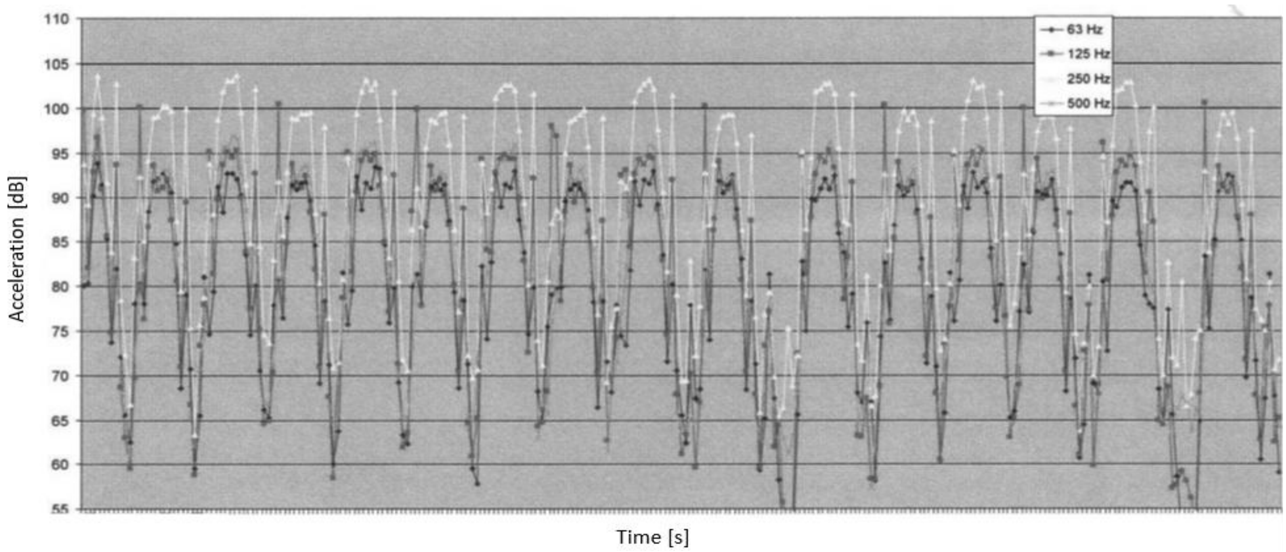


***Fig. 7*** *Acceleration curve on the wall in the z-axis direction when traveling between the 3rd and 4th floors*

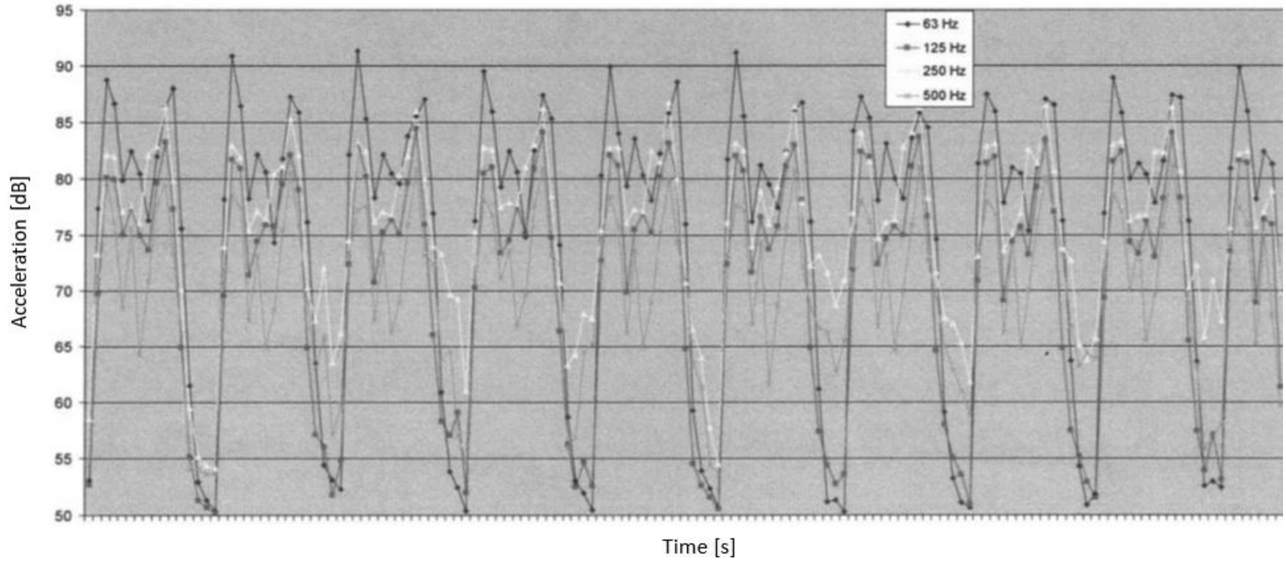


***Fig. 8*** *Acceleration curve on the wall in the z-axis direction during repeated closing and opening of the door on the 4th floor*

The figures clearly show that the acceleration of vibrations on the elevator shaft wall at a frequency of 63 Hz is mostly above 100 dB (in the *x* and *y*-axes). The acceleration values for individual octave bands in the *x*, *y*, and *z*-axes are listed in Tab 1.

***Tab. 1** Vibration values on the wall without adjustments to the guide attachment in [dB]*

| Direction of acceleration | Mode | Maximum level rating | Octave band | | | |
|---|---|---|---|---|---|---|
| | | | 63 Hz | 125 Hz | 250 Hz | 500 Hz |
| x | drive | Absolute max. | 111.2 | 102.0 | 97.8 | 101.3 |
| | | $L_{95}$ | 109.8 | 100.6 | 97.1 | 100.4 |
| | | $L_{90}$ | 109.1 | 100.1 | 96.3 | 100.1 |
| | door | Absolute max. | 107.0 | 97.9 | 84.3 | 83.1 |
| | | $L_{95}$ | 106.5 | 97.4 | 83.9 | 82.7 |
| | | $L_{90}$ | 106.4 | 97.1 | 83.6 | 82.5 |
| y | drive | Absolute max. | 103.9 | 96.4 | 93.0 | 101.0 |
| | | $L_{95}$ | 102.8 | 95.7 | 91.6 | 100.1 |
| | | $L_{90}$ | 102.3 | 95.3 | 90.7 | 99.6 |
| | door | Absolute max. | 102.7 | 96.2 | 83.8 | 83.4 |
| | | $L_{95}$ | 102.5 | 95.3 | 82.8 | 81.9 |
| | | $L_{90}$ | 102.3 | 95.1 | 81.4 | 81.3 |
| z | drive | Absolute max. | 93.8 | 100.5 | 103.7 | 97.7 |
| | | $L_{95}$ | 92.4 | 95.3 | 102.7 | 95.5 |
| | | $L_{90}$ | 91.7 | 94.6 | 102.0 | 94.3 |
| | door | Absolute max. | 91.3 | 84.3 | 86.7 | 81.2 |
| | | $L_{95}$ | 88.7 | 83.1 | 85.9 | 79.9 |
| | | $L_{90}$ | 87.2 | 81.9 | 83.1 | 78.1 |
| Measurement uncertainty | | | 3 | | | |
| Limit according to VDI 2566-2-2004 | | | 90 | 90 | 85 | 85 |

From the above acceleration values (in [dB]), it is clear that:

- The most significant components for the propagation of noise from elevator operation are those perpendicular to the wall, i.e., in the direction of the *y*-axis,
- The acceleration components in the direction of the *x* and *z*-axes are less significant,
- The elevator activities that are relevant are those corresponding to the highest measured values (travel causes greater vibrations transmitted to the wall than opening or closing the doors),
- The maximum acceleration values given are not the absolute maximum measured levels due to statistical evaluation, which excludes outliers.

### 3.2 Measurement results after adjustment

The measurements were performed to the same extent as in the previous case. In this case, 1 cm thick hard rubber pads were inserted into the structure connecting the guides to the bracket and the bracket to the wall.

The Figs. 9 to 14 show the acceleration curves on the elevator shaft wall in the *x*, *y*, and *z* directions after the structural modification. The numerical values in the individual octave bands for the *x*, *y*, and *z* axes are given in Tab. 2.

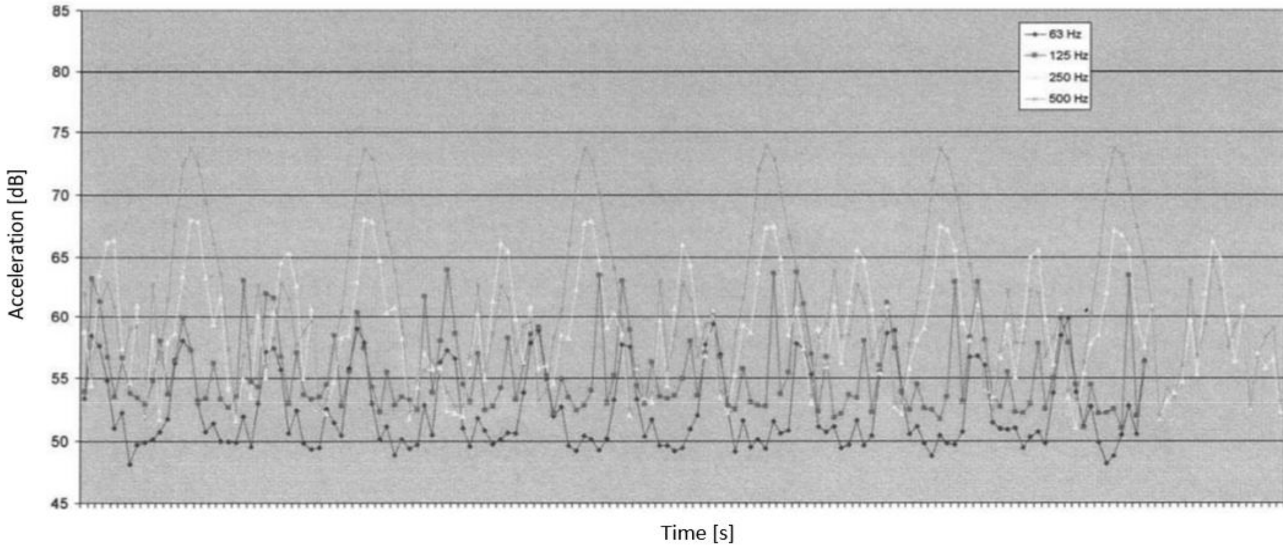


***Fig. 9*** *Acceleration curve on the wall in the x-axis direction when traveling between the 3rd and 4th floors – after adjustment*

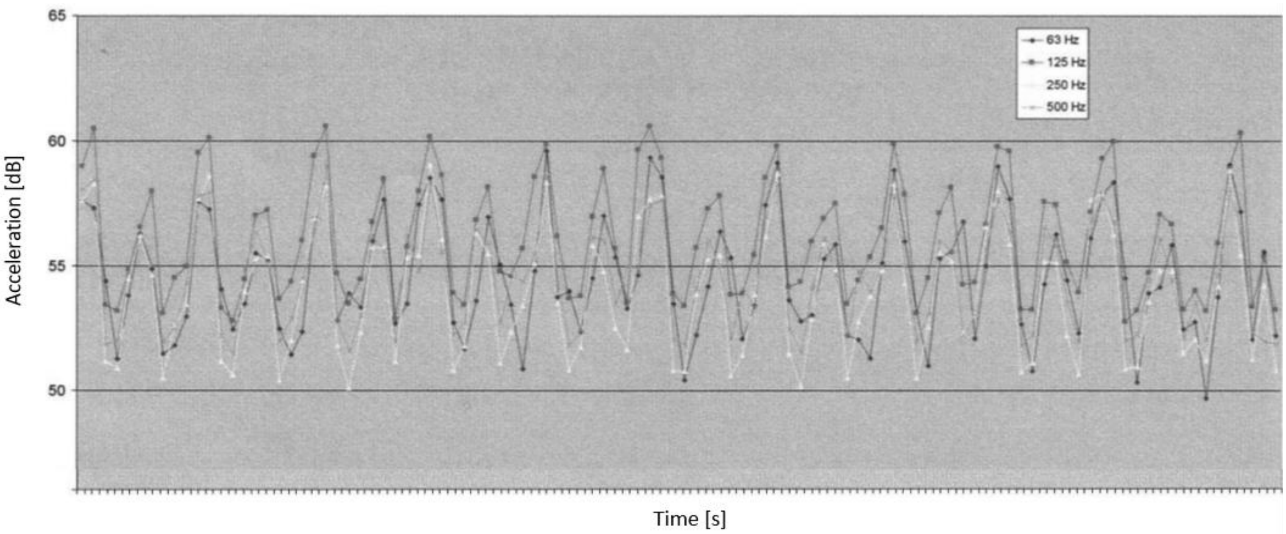


***Fig. 10*** *Acceleration curve on the wall in the x-axis direction during repeated closing and opening of the door on the 4th floor – after adjustment*

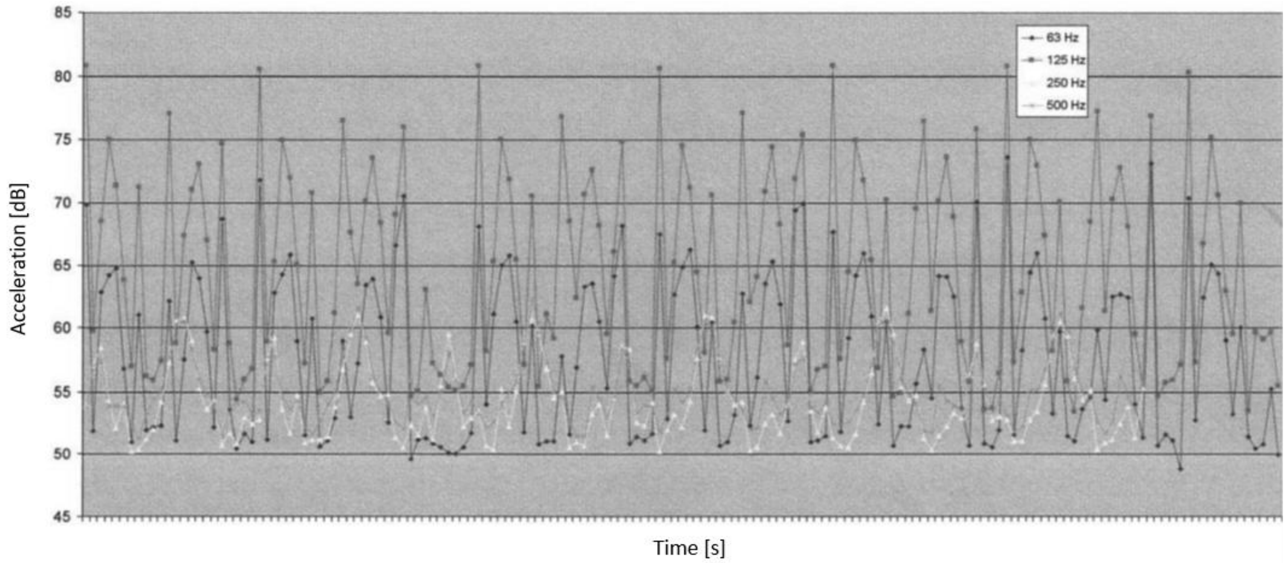


***Fig. 11*** *Acceleration curve on the wall in the y-axis direction when traveling between the 3rd and 4th floors – after adjustment*

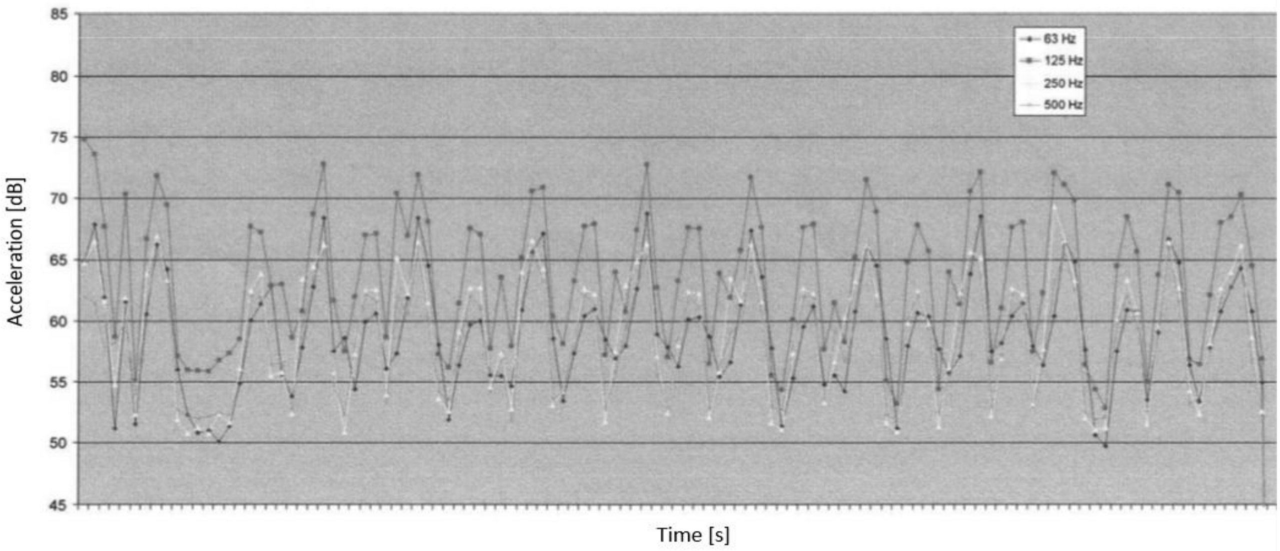


***Fig. 12*** *Acceleration curve on the wall in the y-axis direction during repeated closing and opening of the door on the 4th floor – after adjustment*

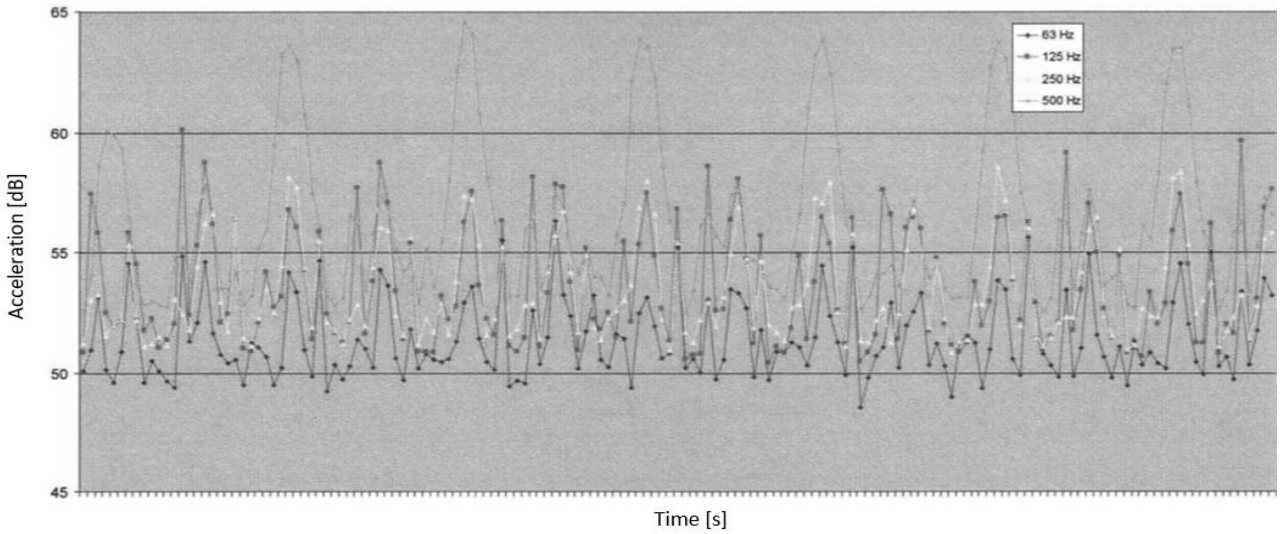


***Fig. 13*** *Acceleration curve on the wall in the z-axis direction when traveling between the 3rd and 4th floors – after adjustment*

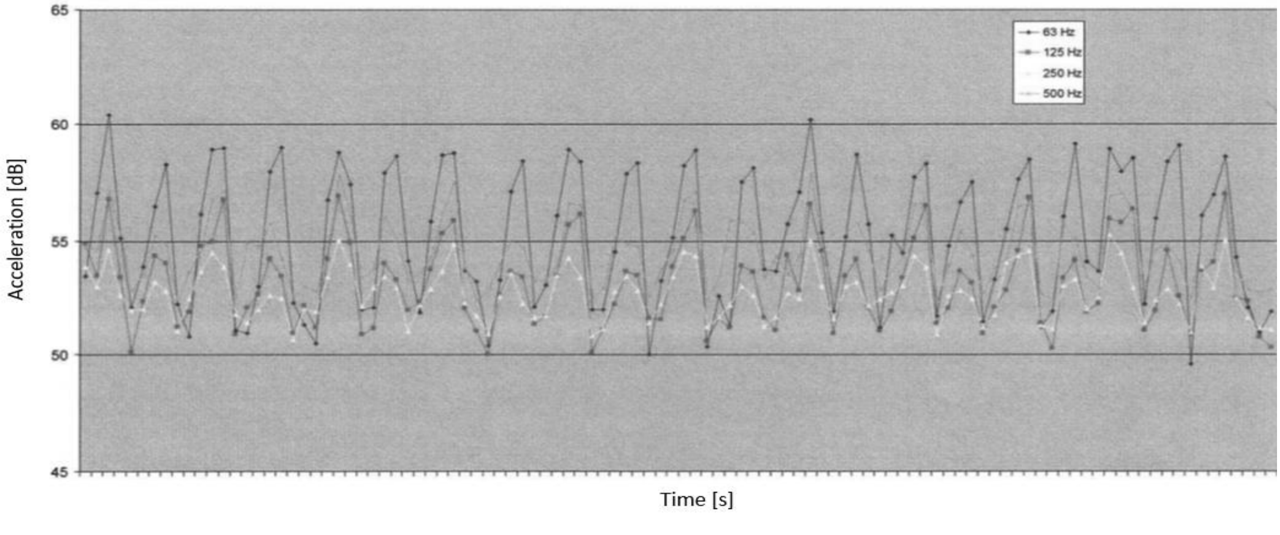


***Fig. 14*** *Acceleration curve on the wall in the z-axis direction during repeated closing and opening of the door on the 4th floor -after adjustment*

**Tab. 2** *Vibration values on the wall after adjustments to the guide attachment in [dB]*

| Direction of acceleration | Mode | Maximum level rating | Octave band | | | |
|---|---|---|---|---|---|---|
| | | | 63 Hz | 125 Hz | 250 Hz | 500 Hz |
| x | drive | Absolute max. | 59.9 | 63.9 | 61.6 | 62.4 |
| | | $L_{95}$ | 58.4 | 63.0 | 60.5 | 60.2 |
| | | $L_{90}$ | 57.6 | 61.6 | 59.4 | 59.0 |
| | door | Absolute max. | 59.6 | 60.6 | 59.0 | 59.0 |
| | | $L_{95}$ | 58.8 | 60.1 | 58.3 | 58.1 |
| | | $L_{90}$ | 57.8 | 59.7 | 57.8 | 57.7 |
| y | drive | Absolute max. | 73.6 | 80.9 | 68.0 | 73.9 |
| | | $L_{95}$ | 69.4 | 77.1 | 67.4 | 72.9 |
| | | $L_{90}$ | 66.0 | 75.5 | 66.1 | 71.3 |
| | door | Absolute max. | 68.7 | 74.7 | 69.3 | 66.1 |
| | | $L_{95}$ | 67.2 | 71.9 | 66.4 | 65.4 |
| | | $L_{90}$ | 65.3 | 71.0 | 65.9 | 63.7 |
| z | drive | Absolute max. | 56.3 | 60.1 | 58.6 | 64.6 |
| | | $L_{95}$ | 54.7 | 57.9 | 57.3 | 63.5 |
| | | $L_{90}$ | 53.9 | 57.4 | 56.7 | 62.3 |
| | door | Absolute max. | 60.4 | 57.0 | 55.3 | 57.9 |
| | | $L_{95}$ | 59.0 | 56.6 | 54.6 | 57.1 |
| | | $L_{90}$ | 58.7 | 55.9 | 54.4 | 56.7 |
| Measurement uncertainty | | | 3 | | | |
| Limit according to VDI 2566-2-2004 | | | 90 | 90 | 85 | 85 |

### 3.3 Evaluation

The above measurements show that inserting rubber pads into the wall to which the guide was attached had a positive effect in terms of attenuating vibrations transmitted to the elevator shaft wall in the *y*-axis direction at all frequencies assessed. The insertion of rubber pads had the greatest effect in the 63 Hz octave band, where a maximum reduction of 30.3 dB was achieved. A significant reduction was also achieved in the 250 and 500 Hz octave bands, by approximately 25 dB. The smallest decrease was recorded in the 125 Hz octave band, at only 14.5 dB.

It can be assumed that these rubber pads had a similar effect on reducing the vibration level for the surrounding walls of the elevator shaft, thereby contributing to a reduction in noise in the vicinity of the elevator shaft.

## 4 Conclusion

The article presents the results of measurements of the elevator structure vibrations and their transmission to the elevator shaft walls. It provides acceleration values at the guide attachment to the bracket and at the bracket attachment to the elevator shaft wall in the *x*, *y*, and *z*-axes. It is clear that the acceleration value in the *y*-axis direction, i.e., into the elevator shaft wall to which the guide is attached, radiates vibrational energy, which manifests itself as noise in the surrounding area. In order to reduce the impact of vibrations on the elevator structure and elevator shaft, a modification was designed in which rubber pads were inserted into the structure between the guide and the wall. This modification significantly reduced vibrations in the elevator shaft wall, so it can be expected to also reduce noise in the surrounding area. Since this modification cannot be used in practice, it will be necessary to find another solution in the future.

However, the proposed solution is suitable for flexible mounting of other machines, such as compressors, combustion engines, etc. It should be noted, however, that other types of flexible mounting are also used for flexible mounting, e.g., air springs or coil springs, massive reinforced concrete blocks often mounted in a flexible environment (e.g., in sand). However, it is always necessary to pay close attention to the flexible mounting of machines, which should be based on a detailed analysis of their vibrations.

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