THE TRANSMISSION OF VIBROACOUSTIC ENERGY FROM THE INDUSTRIAL MACHINES THROUGH THE BUILDING STRUCTURE TO THE MOUNTING BASE

Stanislav Ziaran *and* Ondrej Chlebo

Slovak University of Technology, Faculty of Mechanical Engineering, Institute of Applied Mechanics and Mechatrinics, Nam. slobody, Bratislava, Slovakia
email: stanislav.ziaran@stuba.sk

The effective vibro-isolation of machinery reduces the transmission of vibro-acoustic energy into supporting and surrounding structures. The presented paper investigates the conditions for the transmission of dynamic loading (vibration, shock and noise) from vibrating sources, industrial machines with impact processes, transmitted through the building structure of an industrial hall and its subsequent impact on the mounting base of a laser machine used for the production of sheet metal components and on the workplace, as well. The solution of this problem requires theoretical knowledge and methodology for the measurement of the structures involved. Measurements of vibration at the sources and along the path of transmission, as well as noise measurements in the working place, were performed. The results of the vibration and noise measurements were compared before and after application of vibration isolating material to determine the effects on the machines mounting base and workplace environment. Measurements were performed also on an experimental sample to compare real results at the mounting base with experimental results on the sample under the same conditions. Sound pressure level (noise) and mechanical vibration were measured and FFT analysis was used for the detection of vibro-acoustic energy. In the first case, low-frequency vibration is analysed in the mounting base and the transmission of structurally-borne waves are investigated. Finally, the paper suggests necessary conditions and measures which may have an impact on the reduction of unwanted vibro-acoustic energy acting on the laser tool machine and its working environment.

Keywords: vibration, measurement, vibro-isolation, machines, plants

1. Introduction

Passive vibration isolators of various kinds are used to reduce the transmission of vibrations and shocks. One particular example are resilient supports used as mounting bases for machinery installed in an industrial hall. The passive vibration isolators can be made of different resilient materials, which are mainly used to prevent low frequency vibration problems and/or shock loads from machines and technological processes which are a sources of dynamic loading. Passive vibration isolators of various kinds are also used to reduce the transmission of vibrations and shocks from the sources through the supporting structures. The application of resilient materials (elements) are used to reduce the transmission of audio frequency vibrations (structure-borne sound, 16 Hz to 16 kHz) through a structure which may, for example, radiate fluid-borne sound (airborne, fluid borne, or other), and the transmission of low frequency vibrations (typically 1 Hz to 100 Hz), which may, for example, affect sensitive machinery, persons within the vicinity, or even cause damage to structures when the vibration and/or shocks are severe.

Isolation and structural damping constitute the two most widely applicable means for the control of vibration and mitigation of structurally-borne sound, particularly in the audio frequency range. In essence, vibration isolation involves the use of a resilient connection between a source of vibration and an object to be protected, such that the object vibrates less than it would, if a rigid connection were to be used. In a typical situation, the source consists of a vibrating machine or structure and the object to be protected is e.g. persons, environment, instruments, and machines. Many salient features of vibration isolation can be analysed in terms of a simple model consisting of a rigid mass that is connected to a support via an isolator of vibration (resilient element) that is constrained to translate along a single axis [1, 2, 3, 4]. More complex models are needed to address situations where the magnitude of excitation depends on the motions, where an additional vibro-isolator mass system is inserted between the primary object and the support, and/or at comparatively high frequencies where the vibro-isolator mass plays a significant role or where the isolated items do not behave as rigid masses. Other complications arise due to combined motions and nonlinearities [5].

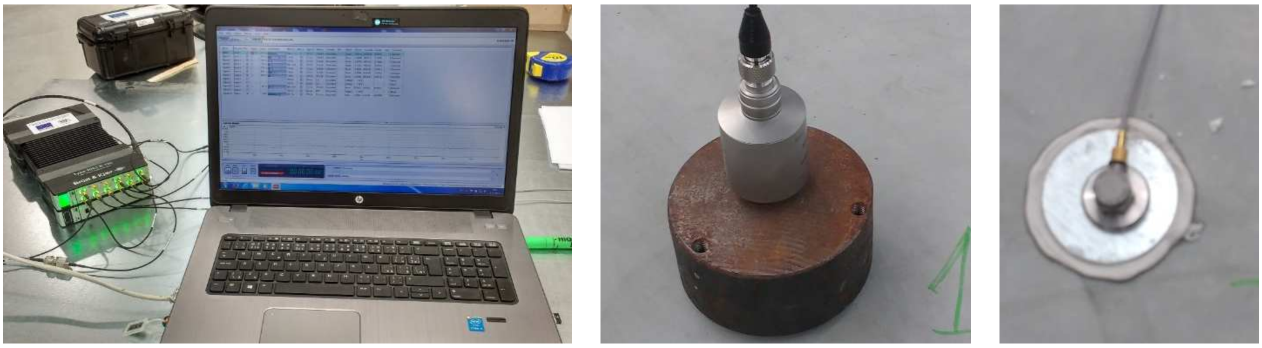
2. Goals, instrumentation and methodology

2.1 Goals of the study

The goals of the study were to investigate and analyse the transmission of structure-borne vibro-acoustic energy flow of an impact excitation generated by the technological process of tool machines. The study shows that the low frequency excitation waves generated by these processes can have a negative effect on the precession of nearby laser processes for the production of components from sheet metal. Frequency spectrum of the measured vibration signals, before and after the production of a concrete mounting base, were compared and analysed. The paper also investigates the energy flow throughout an experimental sample.

2.2 Instrumentation

The transient signal of short duration was generated by the technological process, mechanical shock and the consequent response on the planned mounting base, and was measured using the FFT analyser PULSE Bruel & Kjaer platform (Fig. 1a). The system consists of a piezoelectric accelerometer with a frequency range from 1 Hz to 10 kHz (amplitude ±10 %), seismic accelerometer for low frequency vibrations from 0.1 Hz to 1.5 kHz, and display and memory module. To identify the energy dominant low frequencies more precisely, the seismic sensor was attached to a steel cylinder (Fig. 1b) and the fast Fourier transform (FFT) analysis was carried out using the FFT analyser PULSE. Sensors mounted on the structural elements were attached to the building structure (concrete floor) by means of a circular steel plate, which was glued in position (Fig. 1c). Measurement of the investigated objects coincided with ISO 5348 guidelines for accelerometers and with respect to past experiences [6, 7, 8]. The goal was to ensure that the sensor correctly reproduces the motion of the analysed components without interfering with the response. The sound pressure level was analysed using Sound Level Meters 2250 Bruel & Kjaer platform. Other than the frequency range, for the type of signal, it is also very important to select the appropriate type of averaging as well and number of averages per unit time as well as a suitable time window [7, 9]. The methodology presented in the paper can also be applied to other excitation sources of low frequency vibration.

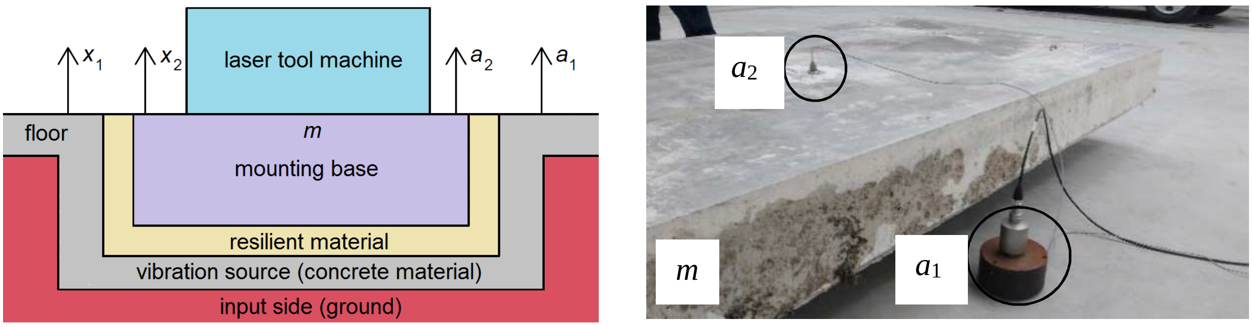


a) b) c)

Figure 1: Measurement instrumentation

2.3 Methodology of the measurement and determination of the transmission loss

The measurement of the transmission loss assumes that the resilient material behaves in a linear fashion and that it has negligible mass compared with the mass loading (Fig. 2). The method determines the impedance of the material when loaded by a mass providing a compression force equivalent to that found when the resilient material is pressed between the floor, as a vibration source (input side), and receiving system (output side). Hence, the system consists of three blocks, which represent the vibration sources of surrounding tool machines through the floor, isolator (resilient material) and the receiving system (which is laser tool machine and workplace situated on the mounting base) respectively (Fig. 2). A fixed contact is assumed at each connection between the source and isolator and between the isolator and receiver. This is done by measuring the transfer function of the mass-loaded material at all the required frequencies [3, 9, 10]. The method uses a vibration excitation system above which the resilient material (element) is placed with the loading mass m (mounting base plus tool machine – Fig. 2a, or concrete block – Fig. 2b, as a receiver system) on top of the resilient element.



a) b)

Figure 2: Theoretical model (a) and experimental setup (b) for determination of the transfer function and/or transmission loss represented by block diagram of vibration source – isolator of vibration – receiver system

Two accelerometers measure the vibration of the source (on the floor as a vibration source), a_1 ($\ddot{x}_1$), and the vibration of the mass m (on the mounting base and concrete block), a_2 ($\ddot{x}_2$) (Fig. 2). Assuming the resilient material tested has negligible mass, the equation of motion is

$$m\ddot{x}_2 = -Z_{\mathrm{m}}(\dot{x}_2 - \dot{x}_1) \tag{1}$$

from which the mechanical impedance is

$$Z_{\mathrm{m}} = \frac{j\omega m}{\left[\dfrac{A_1}{A_2}\right] - 1} \tag{2}$$

at a single frequency it follows that [5]

$$\frac{A_1(\mathrm{j}\omega)}{A_2(\mathrm{j}\omega)} = \text{magnitude} \times [\cos(\textit{phase}) + \mathrm{j}\sin(\textit{phase})]$$

where

- $A_i(\mathrm{j}\omega)$ is Fourier's transformation of acceleration a_i ($\ddot{x}_i$);
- *magnitude* – the magnitude of the ratio denoted by A_1/A_2;
- *phase* – the phase difference between A_1 and A_2.

If only the mounting base (receiver system) of mechanical impedance Z_{b} is under consideration, the equation of motion can be written as

$$Z_{\mathrm{b}}\dot{x}_2 = -Z_{\mathrm{m}}(\dot{x}_2 - \dot{x}_1) \tag{3}$$

from which the required vibration transmissibility or transfer function when the resilient material is loaded by the mounting base system or concrete block (see Fig. 2) can be expressed as

$$T = \left|\frac{\dot{x}_2}{\dot{x}_1}\right| = \left|\frac{Z_{\mathrm{m}}}{Z_{\mathrm{b}} + Z_{\mathrm{m}}}\right| \tag{4}$$

It is more suitable to directly determine the transmission loss D (in dB) of the resilient material which can be calculated by the formula

$$D = 10\lg\frac{\ddot{x}_1^2}{\ddot{x}_2^2} \quad \text{or} \quad D = 10\lg\frac{\dot{x}_1^2}{\dot{x}_2^2} \tag{5}$$

The methodology presented here can also be applied to other sources of very low frequency acoustical vibration, as e.g. air-conditioning systems, boiler systems, cogeneration units, etc. [5, 7, 11, 12].

2.4 Conditions for transfer of vibration through resilient material

The vibro-acoustic energy from the surrounding floor (input side) to the mounting base (output side) transferred through the resilient material (vibro-isolation layer) depends on their loading and implementation (mounting) (see Fig. 2). The dynamic transfer stiffness is most appropriate to characterize the vibro-acoustic transfer properties of the vibro-isolators for many practical applications. At low frequencies only elastic and damping forces are important, the low frequency dynamic stiffness is only weakly dependent on frequency due to material properties. In principle the dynamic transfer stiffness of vibro-acoustic isolators is dependent on static preload and temperature.

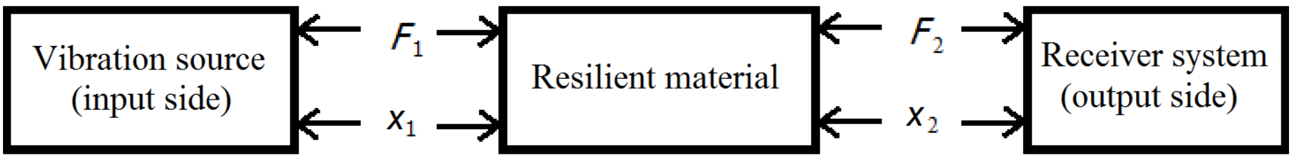


Figure 3: Simplified corresponding block diagram of the vibration source – isolators – receiver system

The system consists of three blocks, which represent the vibration source (floor and ground excited by tool machines), isolator of vibration (resilient material) and the receiving system (laser tool machine and workplace on the mounting base) respectively as is shown in Figs. 2 and 3. The blocked transfer stiffness is suitable for isolator characterization in many practical cases and for this one the damping force is not needed. For unidirectional vibration of a single vibration isolator, the isolator equilibrium may be expressed by the following stiffness equations

$$F_1 = k_{1,1}x_1 + k_{1,2}x_2 \quad \text{and} \quad F_2 = k_{2,1}x_1 + k_{2,2}x_2 \tag{6}$$

where $k_{1,1}$ and $k_{2,2}$ are driving point stiffness's when the isolator is blocked at the opposite side (i.e. $x_2 = 0$, $x_1 = 0$, respectively) and $k_{1,2}$ and $k_{2,1}$ are blocked transfer stiffness's, i.e. they denote the ratio between the force on the blocked side and the displacement on the driven side. For passive isolators is $k_{1,2} = k_{2,1}$, because passive linear isolators are reciprocal. The matrix form of equations (6) is

$$\boldsymbol{F} = [k]\, \boldsymbol{x} \tag{7}$$

with the dynamic stiffness matrices
$$[k] = \begin{bmatrix} k_{1,1} k_{1,2} \\ k_{2,1} k_{2,2} \end{bmatrix} \tag{8}$$

For excitation of the receiving structure via the isolator

$$k_\mathrm{r} = \frac{F_2}{x_2} \tag{9}$$

where k_r, denotes the dynamic driving point stiffness of the receiver. From Eq. (6) and Eq. (9) it follows that

$$F_2 = \frac{k_{2,1}}{1 + \frac{k_{2,2}}{k_\mathrm{r}}} x_1 \tag{10}$$

Therefore, for a given source displacement x_1 the force F_2 depends both on the isolator driving point dynamic stiffness and on the receiver driving point dynamic stiffness.

However, if $\left|k_{2,2}\right| \le 0{,}1\left|k_\mathrm{r}\right|$, then F_2 approximates the so-called blocking force to within 10 %, i.e.

$$F_2 \approx F_{2,\,\mathrm{blocking}} = k_{2,1}\, x_1 \tag{11}$$

Because vibration isolators are only effective between structures of relatively large dynamic stiffness' on both sides of the isolator of vibration (resilient material), Eq. (11) represents the intended situation at the receiving end, therefore these conditions have to be respected when setting up the vibration isolator.

3. Frequency analysis of the results and discussion

3.1 Frequency analysis before the installation of the mounting base

In Fig. 4 the position of the vibration sources, denoted by 1, 2 and 3, with respect to the mounting base and the corresponding frequency spectrums of the vibrations measured on the floor, in close proximity to the mounting base, at different measurement points is shown. The prolonged effects of the vibro-acoustic waves emitted from these sources can lead to damage in the technological and building structures (e.g. cracks in walls and floor), decreased accuracy of products, as well as health risks to persons in the vicinity [13, 14, 15, 16]. With respect to the dynamic loading of the mounting base, the amplitudes at low frequencies are affected by the chosen measuring points on the surrounding floor structures. Design of the mounting base vibro-isolation must incorporate the amplitudes of acceleration in the frequency range from 1 Hz up to approximately 100 Hz. The distances of the sources of vibration and shock loading from the mounting base of a machine are important variables in calculating the maximum amplitude of waves that propagate from these sources [11, 15, 17].

From the frequency spectra in Fig. 4 the peak value acceleration amplitudes can be determined. The maximal values of the dynamic loading occurred in the frequency range from 30 Hz up to 100 Hz and at a frequency of 224 Hz. It should be noted that the frequency of 224 Hz is significant during technological idle times, as well. The source of this frequency with strong amplitude is ventilating system and have not effects on the foundation. From the frequency spectrums an obvious dependence of the amplitude of the vibration acceleration on the frequency, and the amplitude difference between the background and the individual measuring points near the installation of the mounting base, were observed.

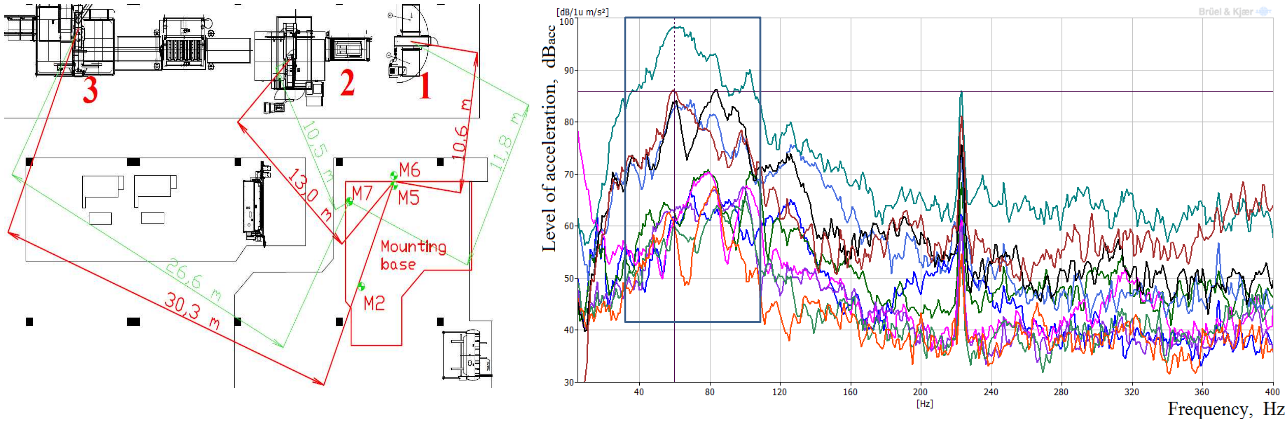


Figure 4: Position of the dynamic sources with reference to the analysed space and frequency vibration spectrum of the floor exited by machine tool (dynamic sources) at different measurement points

3.2 Frequency analysis after the installation of the mounting base

The frequency response of the mounting base surroundings and their transfer to the vibro-isolated foundation (mounting base) is shown in Fig. 5. By comparing frequency-amplitude characteristics for the measured mounting base, significant differences in the transmission loss of vibration energy can be observed. The input represented by measurement point M6 and output represented by chosen measurement points M2, M5 and M7 (see Fig. 4) were analysed. The transmission loss at the 63 Hz (maximal amplitude) was different and depended on the flanking transmission and theory of wave propagation [5, 7]. The attenuation of the transferred vibration energy between point M6 and M2 was 12.7 dB, between points M6 and M5 it was 14.7 dB and between points M6 and M7 it was only 6.6 dB. Based on the measured maximum vibration acceleration values shown in Fig. 5, for various measurement points, exposure to the analysed vibration and shock of the vibro-isolation mounting base, returned higher values of acceleration than was expected [9]. It was based on a comparison of the requirements of the manufacturer with refer to maximum acceleration and maximum speed values of the vibration and shock and the measured results of dynamic loading on the mounting base of the laser tool machinery, and on the comparison of the amplitudes of acceleration before and after installation of the mounting base. The amplitude difference between the background and excitation from external sources considering vibro-isolation with flanking transmission was analysed, as well. The flanking transmission was confirmed by experimental measurements performed on a sample near the mounting base for the same excitation, resilient material and measurement conditions.

In general, the vibration isolators are effective only between the structures relatively high dynamic transfer stiffness on both sides of the vibration isolator and without flanking transmission as is shown hereinafter.

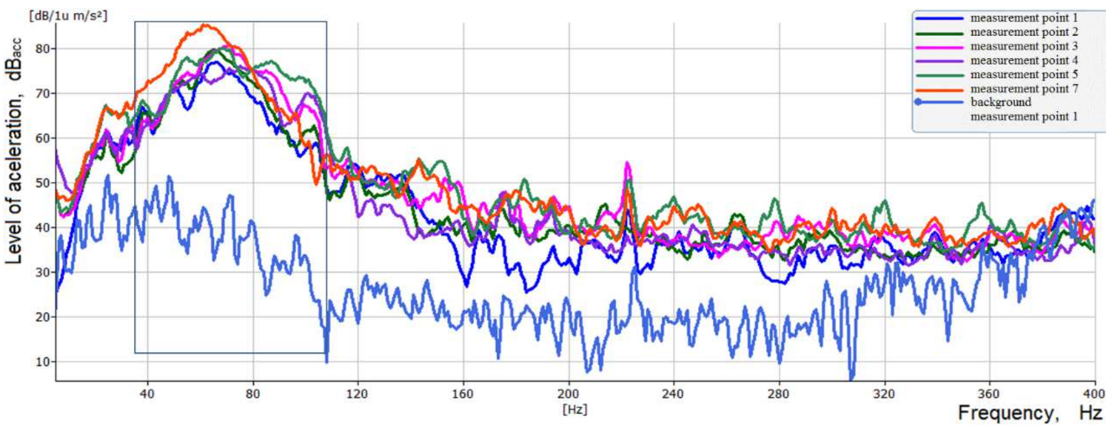


Figure 5: Frequency spectra of measurement points on the mounting base and background

3.3 Determination of the vibro-isolation efficiency

The model shown in Fig. 1 utilizing Eqs. (1) to (11) is correct under the assumption that the resilient elements form the only transfer path between the vibration source and the receiving structure (mounting base). In practice there may be mechanical or acoustical parallel transmission paths which cause flanking transmissions. For any measurement method of isolator properties, the possible interference of such flanking with proper measurements has to be minimized.

The results of frequency analysis on the mounting base showed that the transmission attenuation of vibration isolation material does not reach the calculated value. The measurement results indicated the formation of flanking transmission that was flowing back through the fine-grained concrete into the gaps between the mounting base and the surrounding environment (floor, ground).

To confirm this, an experimental sample was setup (see Fig. 2b) using the same isolation material and the same conditions of excitation. It is clear, that there is no flanking transmission and the contact of vibration isolation material is only with the floor (excitation) on the input side and with the concrete block on the output side (receiver). The results are confirmed by the frequency spectra in Fig. 6, where transmission loss at the frequency of 63 Hz reaches a value of 24.5 dB for the experimental sample (Fig. 6b) and the transmission loss between the measuring points M6 and M2 situated on the mounting base is only half, i.e. 12.7 dB (Fig. 6a).

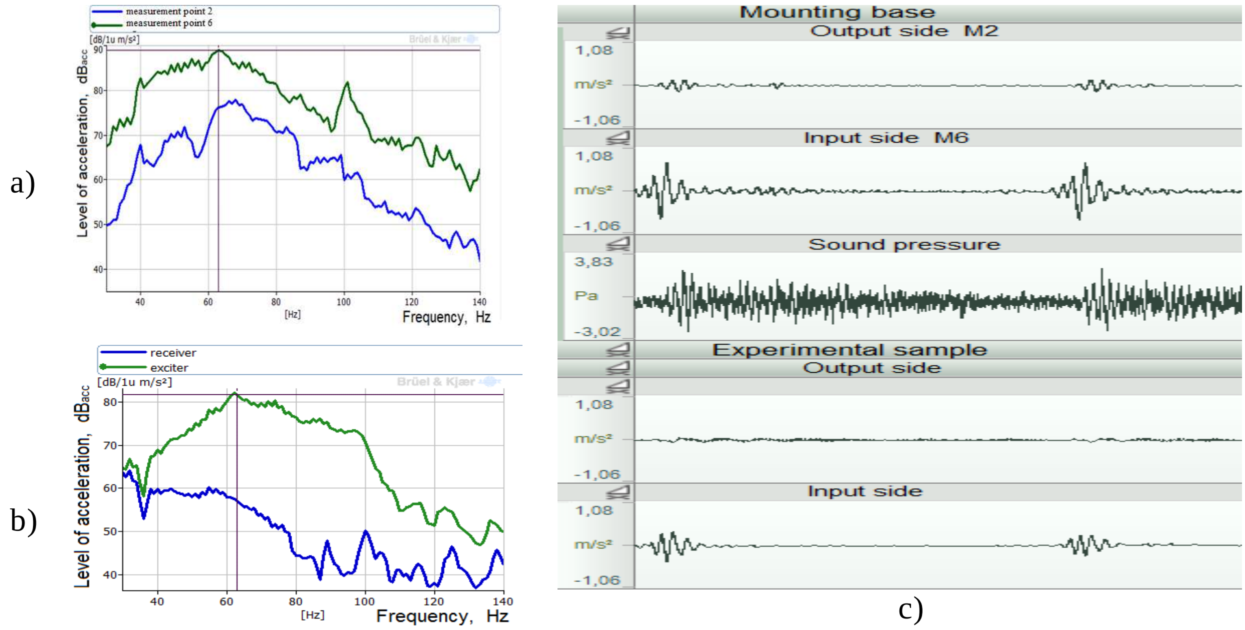


Figure 6: The frequency spectra representing the transmission loss of the excitation input M6 and mounting base output M2 (a) and on the experimental sample (b), and time record representing the transmission loss in the mounting base and in the experimental sample (c)

It is notable that the value of the maximum acceleration measured using experimental sample (near the mounting base) on the concrete block which represented the mounting base was several times less than the actual mounting base at the same excitation, and attenuation between input and output is more than 10-fold, in contrast to the 3-fold attenuation in the measuring points M6 (input) and M2 (output) on the mounting base. Also, a transmission loss of 24.5 dB on the experimental sample, is substantially greater than the transmission loss between points M6 (input) and M5 (output) on the mounting base at a value of 14.7 dB and between measuring points M6 (input) and M2 (output) with just 12.7 dB, which is half of the value of the measured experimental sample. The values of maximum acceleration and transmission loss for the mounting base and the experimental sample confirm the presence of flanking transmission between the mounting base and the ambient environment vibration.

4. Conclusions

Vibration and structure damping constitutes the two most widely applicable means for the control of vibration and structure-born sound, particularly in the audio frequency range. Vibration isolation in essence involves use of a resilient connection between a source of vibration and a space to be protected. For the effective insulation, some necessary parameters must be fulfilled and it is important to create specific conditions [5, 8, 12]. It is very important to obtain the frequency spectrum of the source of vibration. Using the theory introduced above the transmission loss of the material which requires isolation can be calculated.

The problem of low frequency vibration sources, transmission and their influence on machines, structures and humans is currently of great interest. Especially, their effects on persons in proximity to these sources which can affect physical health and create mental problems when an exposed to these frequencies for long periods of time as is shown in Refs. [11, 13, 14, 16]. Increasing noise due to insufficient vibro-isolation has an effect on the comfort of employees and affects their safety and productivity.

Figure 6c contains the time record of the impulse noise with corresponding time record of the mechanical vibration. Decreasing noise after the impact of the tool machines is more gradual than the vibration. It is causes by transformation of structural-borne noise from the surrounding building structures on the air-borne noise. Lowering the intensity of structural-borne noise by effective vibro-isolation will also reduce noise levels at the workplaces.

Acknowledgement

This work was supported by the Slovak Research and Development Agency under the contract No. APVV-15-0630 and with the support from the Scientific Grant Agency of the Ministry of Education of the Slovak Republic and the Slovak Academy of Sciences (VEGA) 1/0544/16.

Európska únia

This contribution was made possible by the project “Improving the safety of nuclear installations in seismic events” (ITMS project code: 26220220171), on the basis of support from the operational program for research and development financed by the European fund for regional development.

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