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Research Paper

The Primary Noise Control in the Work Environment by Increasing the Quality of Bearings and Effective Mounting of Machines

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Implementation of European directives is closely related to the quality of production and the associated operational safety, maintenance of machines and mechanical systems, both mobile or stationary, in order to reduce the dynamic load (vibration and noise) on the working environment, not only during their operating state but also during their design, production, and setting of the vibration isolation components. Reducing the dynamic load of mechanical systems and their components is reflected in the working environment by reduced emissions and immissions of noise and vibration. The presented paper investigates the methods and conditions for noise and vibration control, focusing mainly on increasing the quality of rotating machine components, such as bearings by means of effective vibration isolation of the machines. The solution of this problem requires theoretical analysis and methodology for the measurement of the mechanical systems involved. The results of the vibroacoustic measurements were analysed in terms of the low frequency vibration and noise level (quality) of bearings and conditions for effective vibration isolation of the machines using vibroacoustic diagnostic method. Furthermore, the impact on the working environment was also analysed. Finally, the paper suggests some actions to be taken to effectively reduce the unwanted vibrosound energy in working places, also using recycled material as a vibration isolation element.

Keywords: noise control; vibroacoustics; vibroisolation; human; measures.

1. Introduction

Reducing the dynamic load of machines with respect to the surrounding environment and humans leads to their increased safety, reliability, and lifetime, as well as to reducing emissions and immissions of mechanical vibration and noise, thus improving the working environment. The application of noise and vibration control principles in the work environment is essential for the production of machine components, its mounting, and designing of new workplaces in terms of minimising the dynamic impact with respect to the surrounding environment, human factor, structures, and machine systems themselves. Optimal machine designs and well made components are characterised by low vibration levels, and hence levels of noise. In such cases, vibration is an undesirable phenomenon that initiates the sources of damage and accelerates their progress. Vibroacoustic diagnostics is one of the most effective methods of reducing unwanted vibration and noise (Randal, 2011; Kreidl, Šmid, 2006; Tirinda, 2009; Galovič, 2006). It is used both for detecting faults in machine components such as bearings, and for designing and evaluating high quality vibration isolation.

Bearing diagnostics involves extracting the properties of vibration signals (time history and frequency spectra) of bearing components operating in good, as well as faulty, conditions. Under both conditions, the properties of signals generated by the individual components are recovered from the overall dynamic response. In addition to energy levels, phase information can be used effectively to diagnose bearing condition in many instances. Statistical descriptors of the wave or spectral forms are useful to identify faults generated by impulsive events. The vibration and noise of primary sources is also significantly influenced by manufacturing precision, technology of assembly, appropriate lay out and location of the constituent parts, used material components, load, and process technology. Vibration generated during the test quality and a service component of machinery contains a lot of useful information about their technical conditions and also reflects the nature of the working processes and the intensity of the load of the component or the machine itself.

Vibration isolation and structural damping constitute the two most widely applicable means for the control of vibration and structure borne sound, particularly in the audio frequency range. Data on the damping inherent in materials cover a wide range, extending from comparatively low damping for high strength structural materials to a very high damping for some viscoelastic and resilient materials (typically, plastics or elastomers) with a limited strength. The structural components that are sufficiently strong and that also exhibit a relatively high damping may be obtained by combining high damping viscoelastic materials with structural materials in the form of added layers, or in sandwich arrangements. The article is therefore focused on primary vibration and noise reduction processes for machine systems to reduce noise in the working environment (Flimel, 2017; Smagowska, Pleban, 2018; Žiaran, Chlebo, 2016; 2017).

2. Goals, tool design, and methodology

An undesired dynamic loading generated by bearings is presented as an increased vibration severity and increased noise, both of which affect the machine itself and human surroundings. The first scientific aim of the article is to design vibroacoustic parameters for the objective methodology for the assessment of bearings in terms of their dynamic load, where the acceleration of vibration is measured and evaluated from the time history and autospectrum. Furthermore, the generated bearing noise is simultaneously subjectively assessed by the operator's auditory skills (see Fig. 2a). The proposed methodology is based only on objectively measurable parameters that characterise the vibration intensity and nature of the noise generated by the bearing and cannot be affected by the residual noise detected in the location of the quality assessment of the bearing. The subject of testing was a statistically representative sample of double row ball-ball bearings. The second aim of the article is to investigate and analyse the transmission of structure borne vibrosound energy flow of the machines mounting by the vibration isolation materials. The study shows that low frequency excitation waves generated by different machines can have a negative effect on the accuracy of production, working places (human), and the machine itself. Frequency spectrum of the measured vibration signals after application of the vibration isolation mounting base (Fig. 1, top) and through an experimental sample (Fig. 1, bottom), where is unambiguously no flanking transmission, were compared and analysed.

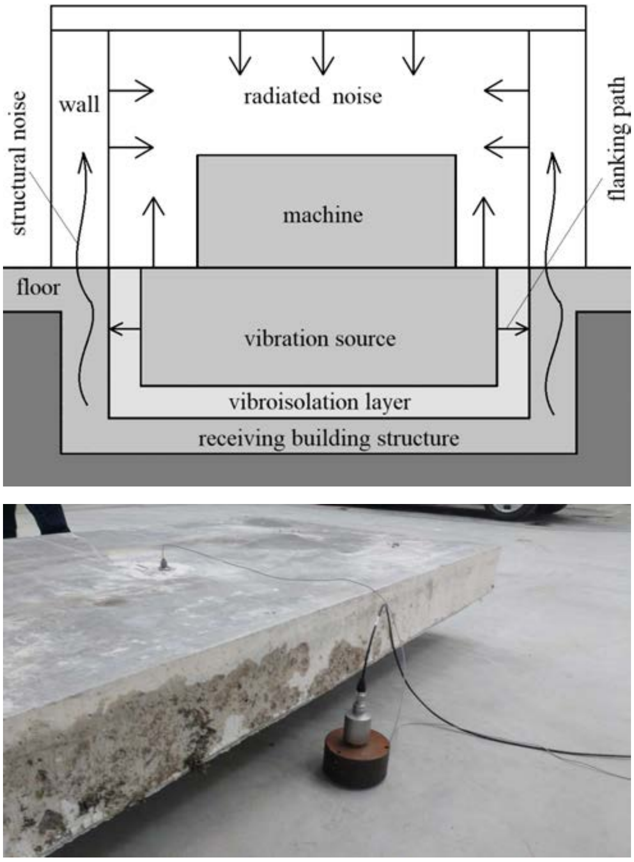

Fig. 1. Real mounting of the machine (top) and experimental set up (bottom) for determination of the transmission loss represented by block diagram (see Fig. 3) of vibration source – vibration isolation – receiving building structure.

The stationary and non-stationary excitation signals of the bearings and the signal of the machines and their technological process in an industrial plant was generated and measured using the frequency analyser. The apparatus consists of a piezoelectric accelerometers with a frequency range from 1 Hz to 10 kHz, seismic accelerometer for low frequency vibrations from 0.1 Hz to 1.5 kHz (Fig. 2c), impact hammer, and display and memory module. To identify the energetically dominant Eigen frequencies of the bearings more precisely, the fast Fourier transform (FFT) analysis was carried out using the frequency analyzer PULSE. The microphone was used for the acoustic measurements of the signal generated by bearings and was compared to the vibration signal. The hand held precision sound analyser Type 2250 (classes 1) was used to do FFT transformation of the time signal into the frequency domain. The 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. 2b). The measurement of the investigated objects corresponded to ISO 5348 and ISO

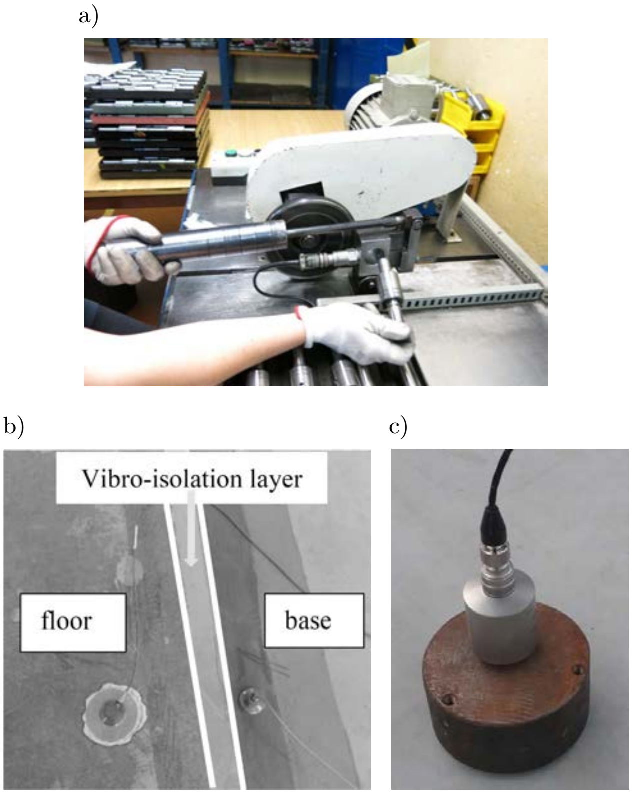

Fig. 2. Bearing test equipment and subjective evaluation (a); attaching the sensors to the sides of the vibroisolation layer (b); attachment of a seismic sensor (c).

7919-1 guidelines for accelerometers and were made with respect to past experiences (Darula, Žiaran, 2011; Žiaran, Darula, 2013). The goal was to ensure that the sensor would correctly reproduce the motion of the analysed components without interfering with the response. For the signal type, besides the frequency range, it is also very important to select the appropriate type of averaging, as well and number of averages per unit time and a suitable time window (Tuma, 2009). The methodology presented in the article can be also applied for other sources of vibration.

3. Conditions for transmission of vibration through vibration isolation element

This part of the paper explains why the dynamic transfer stiffness is the most appropriate to characterise the vibroacoustic transfer properties of isolators and/or resilient material for many practical applications. The vibrosound energy from the mounting base of the vibrating machine (input side) to its receiving structure (output side) transferred through a vibration isolation element depends on the loading and mounting strategy. Dynamic transfer stiffness is the most appropriate parameter to characterise the vibroacoustic transfer properties of the vibroisolation element for many practical applications. At low frequencies only elastic and damping forces are important; low frequency dynamic stiffness is only weakly dependent on frequency due to the material properties. In principle, the dynamic transfer stiffness of vibroisolation elements is mainly dependent on the static preload and its perfect isolation from the source and the receiving structure (Izrael *et al.*, 2011; Žiaran, Chlebo, 2016; 2017). In other words, the following theory only applies if there is no flanking transmission between the vibration source and the receiver system.

The system consists of three blocks which represent the vibration source (machine and mounting base as a source(s) of vibration), vibration isolation element(s) (isolator), and the receiving structure(s) (building and technological structures) respectively, as is shown in Figs 1 and 3. The blocked transfer stiffness is suitable for isolation element characterisation in many practical cases. For the presented case, the damping force was not necessary to be considered. Assuming unidirectional vibration of a vibration isolation element (perpendicular to the contact surface), the isolation element equilibrium may be expressed by the following stiffness equations:

$$F_1 = k_{1,1}x_1 + k_{1,2}x_2 \quad \text{and} \quad F_2 = k_{2,1}x_1 + k_{2,2}x_2, \qquad (1)$$

where $k_{1,1}$ and $k_{2,2}$ are driving contact stiffnesses occurring 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 stiffnesses, i.e., they denote the ratio between the force on the blocked side and the displacement on the driven side. For passive isolators $k_{1,2} = k_{2,1}$, because passive linear isolators are reciprocal. The matrix form of Eqs (1) is

$$\mathbf{F} = [k]\mathbf{x} \qquad (2)$$

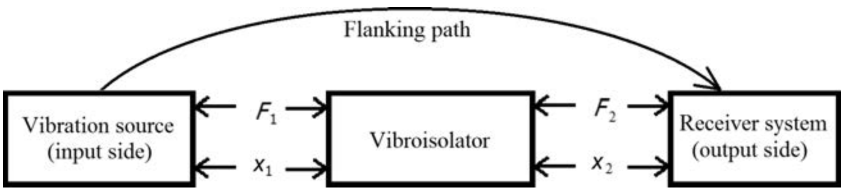

Fig. 3. Simplified block diagram of the vibration source – vibration isolation element – receiving structure and flanking path.

with the dynamic stiffness matrices

$$[k] = \begin{bmatrix} k_{1,1} & k_{1,2} \\ k_{2,1} & k_{2,2} \end{bmatrix}. \tag{3}$$

For excitation of the receiving structure via isolator

$$k_{\mathrm{r}} = \frac{F_2}{x_2}, \tag{4}$$

where k_{r} denotes the dynamic driving contact stiffness of the receiver. From Eqs (1) and (4) it follows that

$$F_2 = \frac{k_{2,1}}{1 + \frac{k_{2,2}}{k_{\mathrm{r}}}} x_1. \tag{5}$$

Therefore, for a given source of displacement x_1 the force F_2 depends both on the isolator driving contact dynamic stiffness and on the receiver driving contact dynamic stiffness. However, if $|k_{2,2}| \leq 0,1\,|k_{\mathrm{r}}|$, then F_2 approximates the so called blocking force to within 10%, i.e.,

$$F_2 \approx F_{2,\mathrm{blocking}} = k_{2,1} x_1. \tag{6}$$

Because vibration isolators are only effective between structures of relatively large dynamic stiffness on both sides of the vibration isolation element, Eq. (6) represents the desirable situation at the receiving end; therefore these conditions have to be met when setting up the vibration isolation elements.

4. Results and discussion

The methodology for assessing the vibroacoustic response of bearings based on the measurement of acceleration and auditory identification in comparison to etalon is subjective and cannot always reliably identify the specific bearing noise near the permissible noise limit. The subjective method is also dependent on residual noise in the test room and, in particular, discrete frequencies limit the identification of discrete components representing specific bearing damage (Fig. 4). It is clear from Fig. 4 that the vibration sensing within a specified frequency range (see Fig. 6) is much more transparent than the noise sensing per-

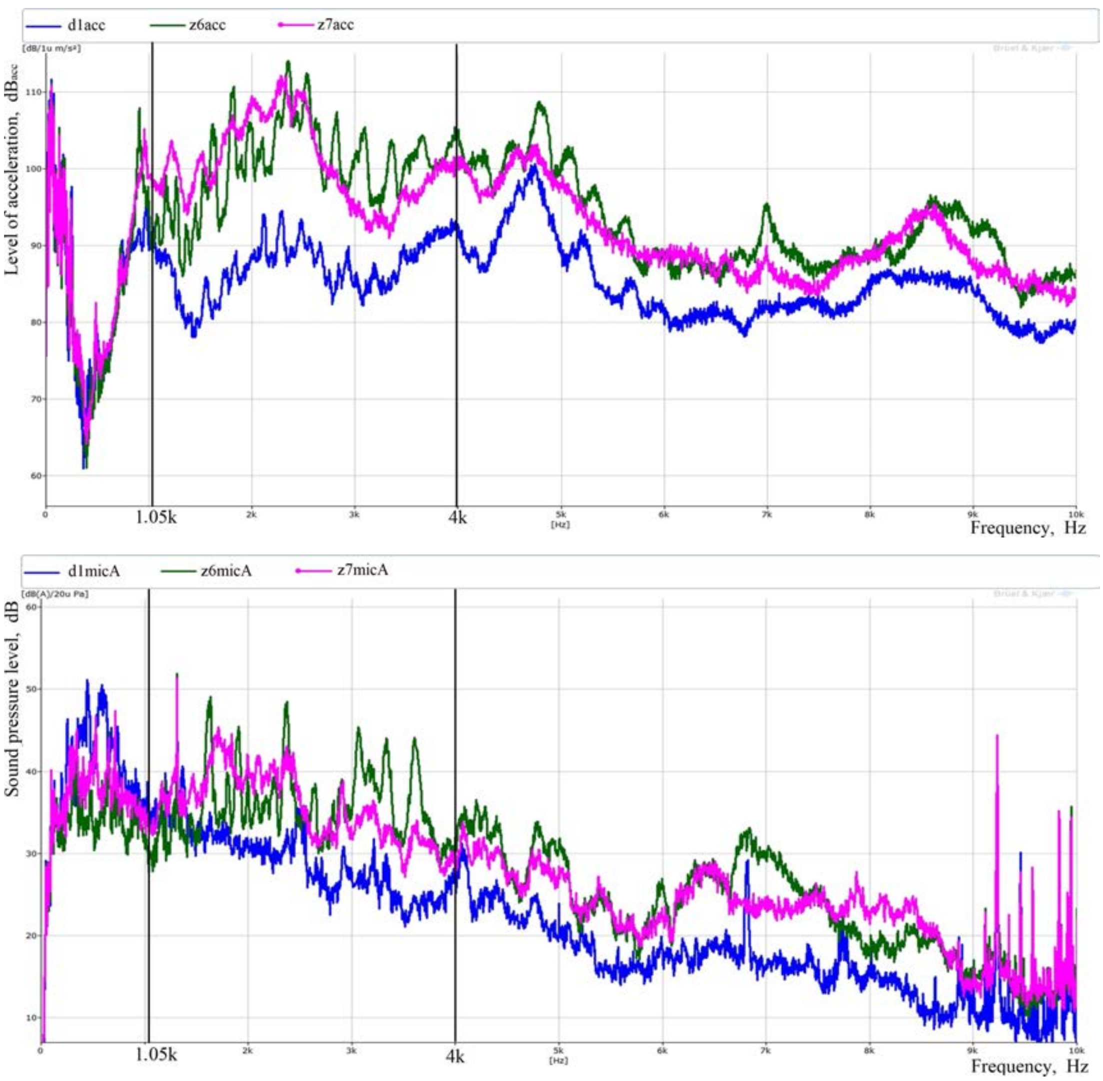

Fig. 4. Bearing frequency spectrum of vibration (top) and radiated noise frequency spectrum with filter A (bottom).

formed either by the microphone, or the auditory organ. The differences in the perception of satisfactory and unsatisfactory sound in the selected frequency interval are minimal, which may lead to an incorrect evaluation of the bearing quality.

Vibrosound energy generated by machine components. Rotational frequency and bearing loading , as well as those generated stationary and non-stationary vibration signals of variable intensity which were dependent on the quality (vibration and noise level according to the subjective methodology) and type of bearing faults (damage) were analysed under specified test conditions. The amplitudes of vibration and sound at rotational frequencies depend on the accuracy of the production of individual components and on the assembly technology of bearings. A total of 45 good, bad, and unclassified bearings were diagnosed and classified according to the original methodology. For the design of the objective methodology, maximum acceleration values read from the time history of the vibration signal generated by the bearing were recorded. Moreover, effective decibel values of the acceleration were measured by means of an FFT analyser within the selected frequency range, based on the experimental results of a representative sample of that type of bearings (Žiaran *et al.*, 2015).

The time response of the vibroacoustic signal generated by the bearings unambiguously contributed to the determination of the quality parameters of the bearing assessment. From the recorded time responses, the time variation of noise can be assessed, and the character (nature) of the noise can be determined.

The comparison of the time history of the dynamic response of the tested bearings shows significant differences in the amplitudes and their time distribution, where the dynamic response of satisfactory bearings, bearings with significant noise, and pulse rattling bearings are shown (Fig. 5). In Figs 5a,b time histories

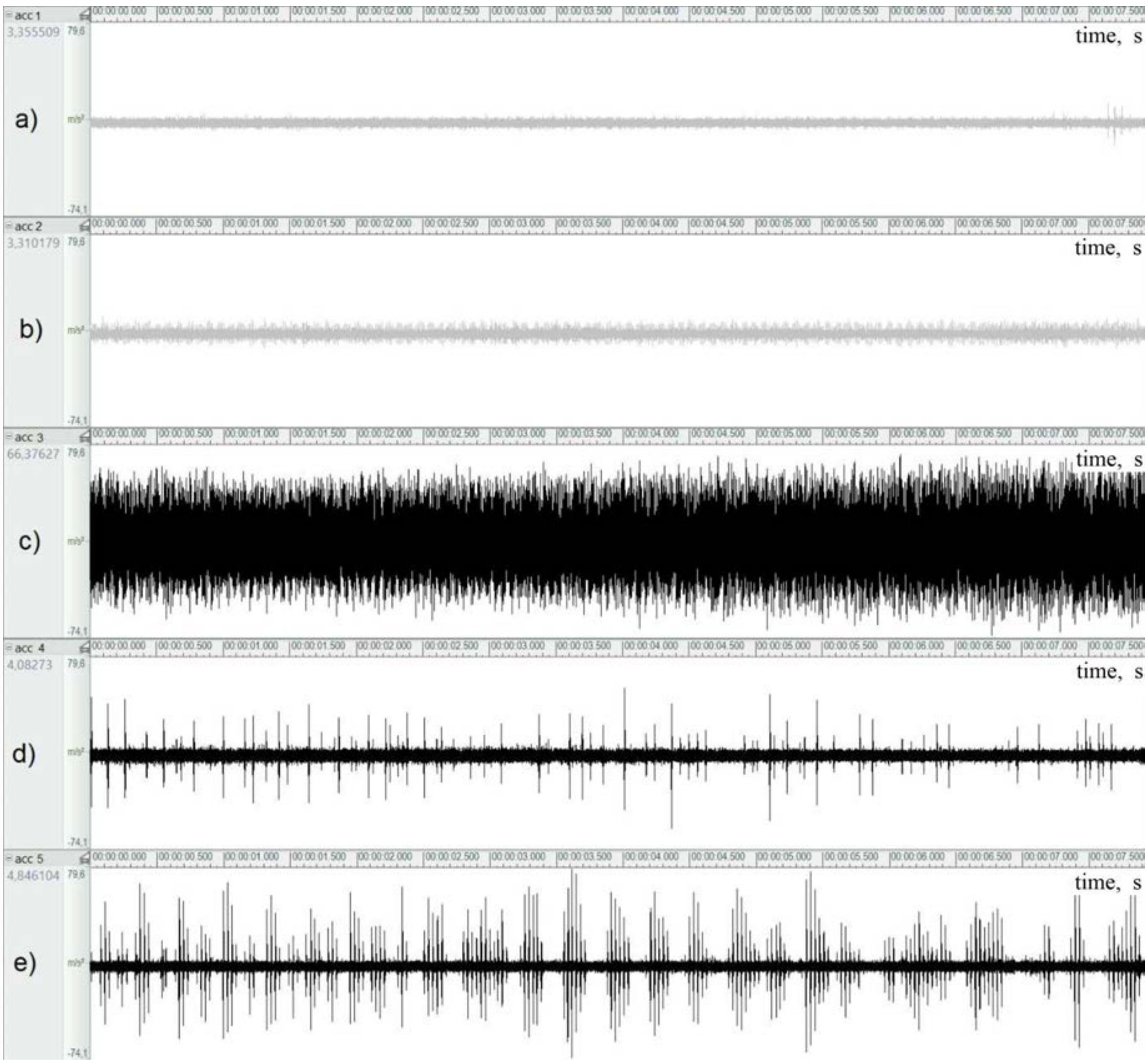

Fig. 5. Time histories of the dynamic response qualified as satisfactory bearings: a) and b) and qualified as unsatisfactory: c), d) and e).

measured vibration acceleration generated by bearings, qualified as satisfactory (good), are specifically selected. The amplitudes of the time histories of these bearings are steady and their sound response is very weak. Figure 5c,d,e presents the specifically selected time histories of the vibration acceleration measured of three bearings classified as unsatisfactory for the given type. The selection of these time histories of tested samples was carried out on the basis of their acoustic responses, so that they represent the different nature of noise, such as loud noise, rattling, low frequency noise, impulse noise, and tonal noise. The dynamic noise is less pronounced and smoother in high quality bearings than in low quality ones, where the amplitude of vibration is, on average, greater, with more pronounced regular and irregular maximum amplitudes achieving high acceleration values.

Abnormal time dynamic response of the bearings over the maximum value of acceleration value is, according to the proposed methodology, one of the three criteria (parameters) for the objective evaluation of bearing quality, whose upper limit, for the given type of bearing, is set at 30 m/s^2.

The comparison of the vibration acceleration of time histories for the tested bearings showed significant differences in the size of the amplitude and their time distribution. The dynamic noise at high quality (satisfactory) bearings is not so pronounced (analysing the amplitude of vibration) and smoother (Fig. 5a,b) than in the case of low quality (unsatisfactory) bearings, where the vibration amplitude is significantly pronounced with regular and irregular amplitude peaks, reaching high acceleration values (Fig. 5c,d,e). These stationary and/or non stationary signals and their maximum amplitudes represent a response of the size and type of bearing damage, and/or component during the production and/or indicative of an insufficient technology and/or inadequate installation and environmental influences. A more detailed identification is possible using frequency analysis which allows detecting the causes and types of faults (Žiaran *et al.*, 2015). However, that is not the purpose of this article.

The frequency analysis, as stated above, does not serve only to determine syndromes and symptoms of bearing faults, but also to determine the statistically significant frequency interval, as well as the equivalent decibel acceleration values in this frequency interval. Frequency spectra of thirty bearings (15 satisfactory and 15 unsatisfactory) in certain parts of the frequency interval form a significant boundary between the quality of satisfactory and unsatisfactory bearings (Fig. 6). Thus, from the effective acceleration value the quality of bearings (satisfactory/unsatisfactory) can be determined. It should be emphasised that the quality (satisfactory) bearing will always have some external dynamic response. This dynamic response may not reduce the quality of the bearing itself. Based on the plots of frequency analysis, using probability theory and experience of investigators in determining the limiting equivalent acceleration value for the given type of bearings, the frequency interval from 1.05 kHz to 4 kHz, to assess the quality of the bearings of the given type, was determined.

The frequency response of the bearing through the effective value of the acceleration at the determined frequency interval is, according to the proposed methodology, the second of the three criteria (parameters) for the objective evaluation of bearing quality.

The frequency response of the bearings through effective acceleration values expressed in decibels

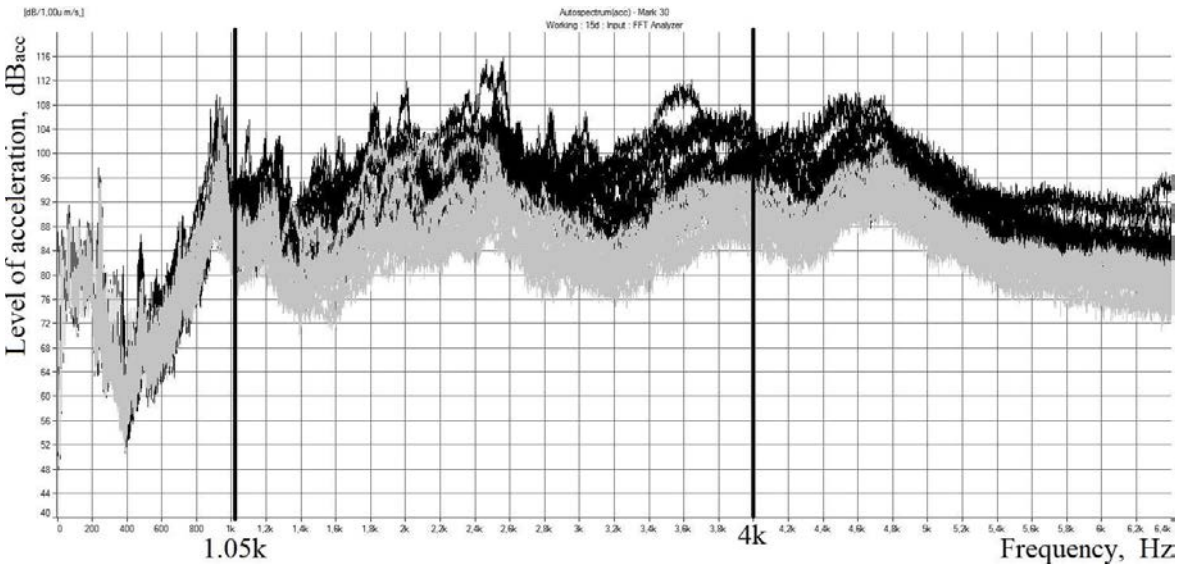

Fig. 6. Frequency spectra of bearings with different dynamic response; spectra highlighted in black represent unsatisfactory bearings and the grey ones show satisfactory bearings.

at a specified frequency range is, according to the methodology developed, the second criterium for the quality assessment of the type of bearings and, to some extent, it characterises the severity of their vibration. A statistically significant frequency range for determining the effective (equivalent) acceleration values expressed in decibels ranges from 1.05 kHz to 4 kHz for the investigated type of the bearings only and is the third parameter of quality. The upper limit of the equivalent level of acceleration was set to 125.5 dB, corresponding to an acceleration of 1.88 m/s^2 of the dynamic behavior of the investigated bearing type (Žiaran *et al.*, 2015). The difference r.m.s. between satisfactory bearings and unsatisfactory bearings was approximately 10 dB. The vibration values in dB for unsatisfactory bearings ranged from 125.6 dB to 135.5 dB, and for satisfactory bearings from 117.2 dB to 125.5 dB, considering the tested, statistically representative, samples of the satisfactory and unsatisfactory bearings. By eliminating the unsatisfactory bearings, the dynamic load on the machine is greatly reduced, thus decreasing the vibration and noise impact on both the surrounding environment, and on people.

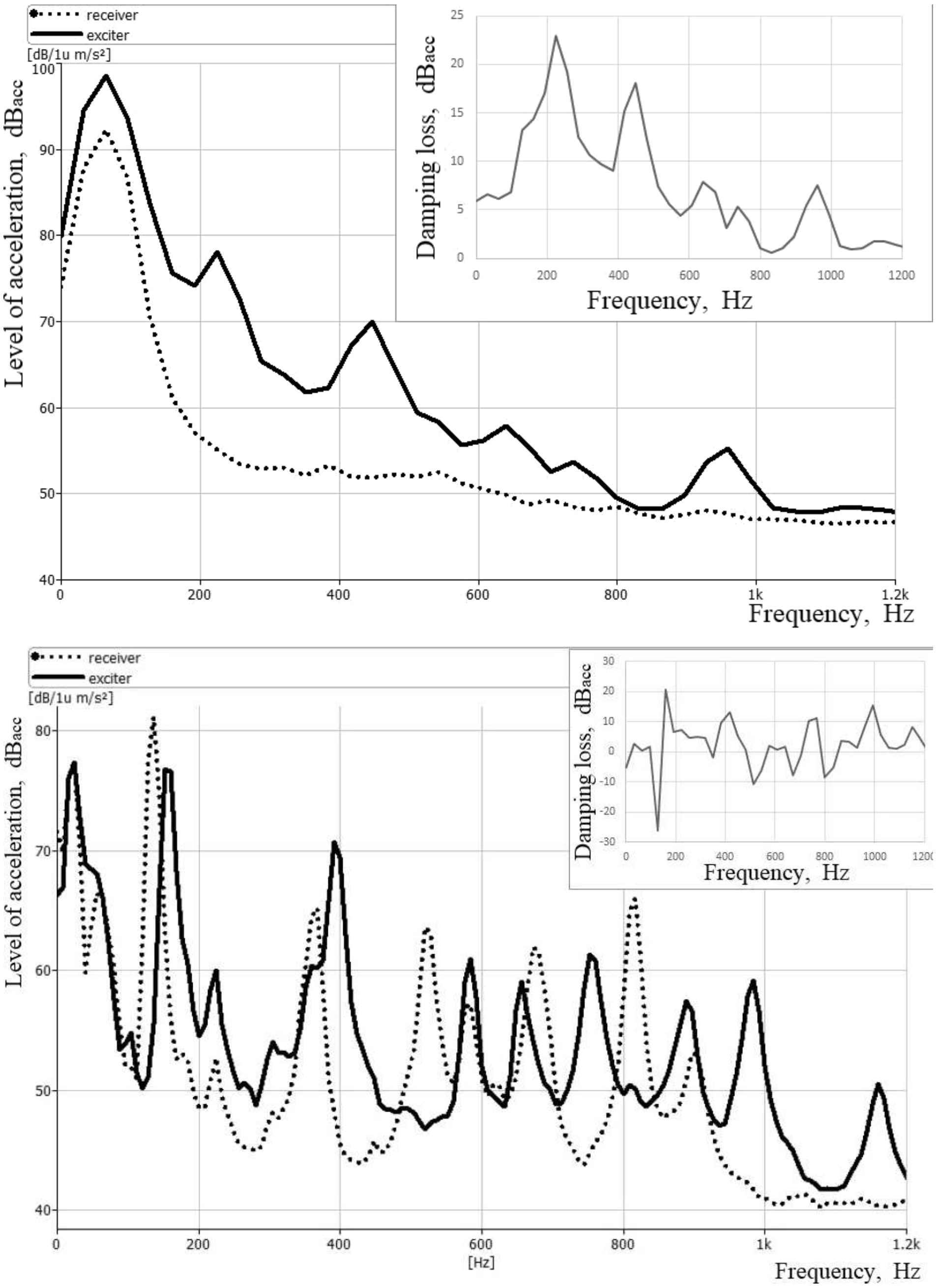

Fig. 7. Frequency spectra representing the transmission loss of the excitation input and output of the experimental sample (top) and on the mounting base (bottom).

Vibrosound energy generated by the machines. The model shown in Fig. 1 utilising Eqs (1) to (6) are correct under the assumption that the vibration isolation elements form the only transfer path between the vibration source (mounting base) and the receiving structure (building structures and human). In practice there may be mechanical and/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 minimised.

The results of frequency analysis on the mounting base showed that the transmission attenuation of vibration isolation material does not reach the calculated value (Žiaran, 2010). 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 set up (see Fig. 1, bottom) 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 (source of excitation) on the input side and with the concrete block on the output side (receiving building structure). The results are confirmed by the frequency spectra in Fig. 7, where transmission loss between the receiver and the exciter, e.g. at the frequency of 235 Hz, reaches a value of 23.5 dB for the experimental sample (Fig. 7, top) and the transmission loss between the two measuring points situated on both sides of the vibration isolation element (on the mounting base and beyond this element) is only approximately one third, i.e., 8.1 dB for the same excitation (Fig. 7, bottom) (Žiaran, Chlebo, 2017).

It is notable that the value of the maximum acceleration measured using an experimental sample (near the mounting base) on a concrete block which represented a building structure was several times less than recorded on the building structure itself at the same source of excitation, and the attenuation between the input and output is more than 10-fold, in contrast to the 3-fold attenuation in the measuring points (input) on the mounting base and (output) on the floor of the building structure (see Fig. 2b). Also, a transmission loss of 23.5 dB on the experimental sample is substantially greater than the transmission loss between different points around the perimeter (input) on the mounting base and around the perimeter (output) on the floor. These values corresponded to 14.7 dB and just 8.1 dB respectively, which is one third of the value of the measured experimental sample, as mentioned above. 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 building structures. In this case, structure borne noise is generated by the badly installed machine, and negative transmission loss values represent the vibration energy concentration in the mounting base corners (Fig. 7, bottom). Structure borne noise in industrial plants is typically generated by machines and their technological process that are not adequately isolated. A well designed isolation of the vibrating sources can effectively reduce the transmission of low frequency energy into acoustically protected spaces, such as work places.

5. Conclusions

The aim of this paper is to show that in order to decrease noise pollution in the working environment it is necessary to reduce vibrations and the noise of individual components of the machine, as well as the transmission of vibrations generated by the machines into the surrounding and technological structures. The vibrosound energy from the machines, when transmitted through the poorly applied vibration isolation elements into the surrounding structures, radiates an undesirable noise which may harm the employees of the company. Therefore, it is essential to reduce the vibrations and noise of individual components of the machines and to design an effective vibration isolation of the machinery and install it *in situ*.

The analysis of these measurements and using the results for statistically representative number of samples tested, helped determine the boundaries of the quality bearings, thus complying with the boundary between dynamically satisfactory bearing and dynamically unsatisfactory bearing in terms of noise and vibrations. From the frequency spectra the statistically significant frequency range for determining the equivalent acceleration values expressed in decibels was determined and from time dependence the maximum value of the acceleration in m/s^2 was identified. Thus unwanted noise of the bearings can be monitored through a maximum acceleration value determined from the time history and time-frequency plots; the nature and intensity of the noise generated by the bearing can be determined. The acceleration limit values were obtained from the time history and the specified frequency spectrum boundaries with the determination of the maximum acceleration level was applied to a particular bearing. The proposed methodology can be applied to all types of bearings respecting the above theory and experimentally verified criteria, namely, the maximum acceleration from the time history and the maximum acceleration level in the statistically defined frequency interval.

Vibration and structure damping constitutes the two most widely applied means for the control of vibration and structure born sound, particularly in the audio frequency range. Vibration isolation in essence involves the 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 (Žiaran, Chlebo, 2016; 2017). 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. In particular, their effects on persons in proximity to these sources, long-term exposure to these frequencies can affect physical health and cause mental problems, as is shown in (Argalášová *et al.*, 2013; Balážiková, Sinay, 2012). Increasing noise due to unsatisfactory machine components and their insufficient vibration isolation affects the comfort of employees as well as their safety and productivity. Reducing vibration intensity of each rotating machine component and lowering the intensity of structure borne noise by effective vibration isolation will also reduce noise levels at the workplace (Šooš *et al.*, 2016).

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